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A dynamical FEA fretting wear modeling taking into account the evolution of debris layer

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ABSTRACT

A new Finite Element Analysis (FEA) strategy is developed to simulate fretting wear, taking into account the evolution of the debris layer trapped in the interface, so called "third body". To validate this approach, simulations were compared to experimental results from gross slip Ti-6Al-4V cylinder/plane experiments. Adequate worn surface analyses allow the estimation of both cylinder and plane friction energy wear rates and the debris layer thickness evolution. A third body conversion factor (γ(x)), expressing the proportion of worn thickness transferred to the third body layer (i.e. debris layer) at a given position in the fretted interface is introduced. A coupled Matlab-Python-Abaqus algorithm is developed to simulate the surface wear on plane and cylinder surfaces to formalize the continuous evolution of the debris layer trapped within interface. Quantitative comparisons with experimental results confirmed the interest of this FEA approach. The maximum wear depth, which was underestimated by nearly 80% without considering the third body, is predicted with an error less than 10%. A numerical investigation demonstrates that the elastic properties of the third body do not influence the surface wear profile. Acting as a contact pressure concentrator, the third body effect appears more geometrical than rheological. This third body FEA fretting wear modeling is extended in order to consider both test duration and sliding amplitude effects. Rather good correlations with experiments confirm the interest of this approach.

1. Introduction

Fretting wear damage are observed in many industrial assemblies subjected to vibrations. Cracking or/and surface wear are occuring depending on the sliding amplitude [1,2]. Small displacement amplitudes, by promoting a partial slip contact, induce cyclic stresses which favor crack nucleation and propagation. However, this loading condition is non dissipative (i.e. very closed tangential force Q – displacement δ* hysteresis) and the surface wear is very limited. Above gross slip transition, a full sliding condition is activated promoting a quadratic dissipative fretting loop. The surface wear is then significantly increased, the contact area extended and the maximum contact pressure sharply reduced. By reducing the surface stresses and removing the top cracked surface, gross slip tends to decrease the cracking risk. Competition between cracking and wear was extensively investigated during the past decades [3–5]. This concept is currently considered in industrial components like dovetail blade-disk aeronautical turbines where sacrificial high wear rate coatings are currently applied to reduce the fretting cracking problem. However, when the wear volume and maximum wear depth are too high, over clearance and potentially cracking induced by "notch" stress concentration can occur [6]. Hence, there is a real interest to model the fretting scar profile and above all to predict the maximum wear depth [7].

Various wear models may be considered for fretting wear damage [8]. However two main strategies are usually adopted to predict the fretting wear volume (V) extension. The first one, based on the Archard's wear equation [9], expresses the total wear volume as a linear function between the product of the normal force and the total sliding distance. Due to the friction fluctuations recent developments suggest that the friction work approach (i.e. friction energy wear concept) provides a more representative description of the surface damage so that [10]:

$$V = \alpha \times \sum Ed \tag{1}$$

Where α is the friction energy wear coefficient. Both Archard and related friction energy approaches are quantitative, require a limited number of variables and are easily implemented in FEA contact modeling. However, these approaches do not explicitly consider the debris layer (third body) trapped in the interface which can drastically modify surface wear damage. Godet followed by Berthier and co-authors

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described this aspect, developing "third body theory" [11–14]. They demonstrated that the wear rate can be expressed as a function of the debris formation flow from the first bodies and the debris ejection flow from the interface. This wear modeling states that the wear rate depends on the thickness and rheological properties of the third body layer. This third body theory provides a more physical description of wear processes. However, the formulation provided by the authors requires numerous variables, and appears complex to implement in FEA simulations.

If the global wear volume prediction is still an open question, the prediction of worn profiles is even less advanced. Indeed, the prediction of local wear depth requires a local description of surface wear processes. In addition to semi-numerical methods [15], a significant effort was done during the past decades to implement surface wear modeling in finite element methods. A common approach consists in moving the surface nodes to simulate the worn profile [16]. However, a major limitation of such approaches is the fact that they do not consider the presence of a third body layer. Ding et al. [16] then Gosh et al. [17] and Basseville et al. [18] were the first to investigate the presence of wear debris in FEM fretting interfaces. They proposed to represent the third body by an individual partition in the FEM model of fretting interface. Other models involving coupled FEM and DEM (discrete element method) analysis were proposed by Haddad et al. [19] and Leonard et al. [20] to evaluate the influence of the third body on friction, contact pressure and stress fields.

However, most of these developments consider a static description of the third body and were not able to simulate simultaneously the evolution of surface wear and third body layer. Besides these, numerical investigations were barely related to experimental results.

Ding et al. [16,21] presented a debris evolution model, with consideration of micro-scale asperity-induced plasticity and included the high related effects of oxidation, for Ti-6Al-4V, to give a debris evolution model, including also reasonable comparisons with measured debris layer thickness evolution across the contact width. Done et al. [22] recently proposed a 3D cylinder/plane surface wear modeling including a third body description. Using semi analytical formulation, the fretting wear profile of the plane is simulated assuming a flat distribution of worn thickness converted to third body. The model also considers a unilateral wear process on plane. Despite such limitations, rather good correlations were observed with experiments. Simultaneously, a similar strategy using FE analysis was introduced [23]. This 2D model allows the bilateral wear of plane and cylinder surfaces. Besides, the comparison with experiments suggest that better predictions are achieved if the γ conversion factor from worn surface to third body thickness is not constant but expressed as a parabolic function of the (x) distance from the center of the contact. Indeed, the probability of wear debris to be included on the third body layer is larger in the center of the contact than on the lateral sides where it can more easily be ejected from the interface.

The purpose of this research work is to deepen such a 2D FE analysis by investigating the effect of the elastic properties of the debris layer and by taking into account the effect of test duration and sliding amplitude regarding the debris layer extension and resulting fretting wear profile predictions.

2. Experiment

2.1. Materials

The studied interface consists in a homogeneous Ti-6Al-4V/Ti-6Al-4V cylinder plane interface.

The Ti-6Al-4V titanium alloys consist in a 60% alpha and 40% beta microstructure. Its mechanical properties are listed in Table 1.

Table 1
Mechanical properties of the studied Ti-6Al-4V.

	Young's modulus, E (GPa)	Poisson's ratio, υ	Vickers hardness Hv0.3	Plastic yield $\sigma_{YO.2\%}$ (MPa)
Ti-6Al-4V	120	0.3	360	880

2.2. Plain fretting test

This analysis is focused on a single cylinder/plane contact configuration, with cylinder radius R = 80 mm, normal load P = 1066 N/mm inducing a maximal Hertzian pressure of p_{max} = 525 MPa, and constant radius a_H = 1.29 mm. The lateral width of the pad was fixed at L = 8 mm allowing a 2D plain strain hypothesis (i.e. $a_H/L < 0.16$). As shown in Fig. 1, the fretting test consists in a vertical contact where the normal force (P × L) is applied using a trolley system and the displacement is monitored using a hydraulic actuator [24]. The resulting tangential force (Q × L) is recorded allowing the plotting of Q-δ fretting loops. From the fretting loop, we deduced the tangential force and the displacement amplitude respectively Q* and δ*. Integrating the hysteresis loop, we can also assess the friction energy Ed. The displacement amplitude includes the contact displacement but also the tangential accommodation of the test apparatus and specimens. Therefore, to only consider the contact sliding, the experiments were performed monitoring the residual displacement δ_0, measured on the fretting cycle when Q = 0. Indeed when Q = 0, no tangential deformations are generated in the test system and the corresponding δ_0 displacement corresponds well to the sliding amplitude imposed in the interface.

A representative mean friction energy coefficient is considered to quantify the friction response of the interface [10].

$$\mu_e = \frac{Ed}{4\times\delta_0^*\times P\times L} \quad (2)$$

For the studied conditions it was established at μ_e = 0.65 ± 0.05

Frequency was decreased to f = 0.11 Hz in order to limit adhesive wear phenomena and to ensure almost exclusively quasi-pure three bodies abrasion wear process, as previously described [26].

Two sets of experiments were applied to calibrate the surface profile modeling (Fig. 2). The first one consists in varying the test duration from 2500 to 30,000 fretting cycles while keeping constant the sliding amplitude at δ_0 = ± 75 µm. The second set of experiment consists in keeping constant the test duration at 10,000 fretting cycles while varying the sliding amplitude from δ_0 = ± 62, ± 75, ± 100 and ± 125 µm. Then various fretting test conditions outside the N-δ_0 calibration axis were applied to establish the stability of the model (Fig. 2)

2.3. Fretting scar analysis

2.3.1. Fretting scar morphology

After the test, specimens were cleaned for 30 minutes in an ultrasonic ethanol bath to remove most of wear debris trapped in the fretting scars. This was followed by 3D surface analysis of the worn profiles. The cylinder shape on the pad was removed and wear volumes on plane (V_p) and cylinder (V_c) are measured (Fig. 3).

The corresponding plane (α_p) and cylinder (α_c) energy wear rates [10] were estimated assuming a linear evolution of wear extension and an equal distribution of friction energy toward plane and cylinder surfaces:

$$\alpha_p = \frac{V_p}{\sum Ed/2} \text{and } \alpha_c = \frac{V_c}{\sum Ed/2} \quad (3)$$

The objective of this research work is to predict the 2D fretting scar profiles. This implies to extract representative 2D experimental worn