Wear 376-377 (2017) 1475–1488

Contents lists available at ScienceDirect

Wear

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

A numerical simulation of fretting wear profile taking account of the evolution of third body layer

P. Arnaud, S. Fouvry*, S. Garcin

Ecole Centrale de Lyon, LTDS, Ecully, France

ARTICLE INFO

Article history:
Received 30 November 2016
Received in revised form
13 January 2017
Accepted 18 January 2017

Keywords:
Ti-6Al-4V
Fretting wear
FEM simulation
Third body

ABSTRACT

In this paper we propose a new Finite Element Modeling (FEM) strategy to simulate fretting wear profiles, taking account of the presence of a third body layer evolving over time. To validate this approach, the simulations were compared to experimental results on a gross slip Ti-6Al-4V cylinder/plane interface. A simple experimental procedure based on adequate superimposition of worn surface profiles allowed the estimation of individual cylinder and plane friction energy wear rates and quantification of the thickness of the third body embedded within the interface. A third body conversion factor, expressing the proportion of worn thickness transferred to the third body layer at a given position in the fretted interface was developed. The various quantitative variables were introduced in a coupled Matlab-Python-Abaqus algorithm where the worn thickness of counterfaces was simulated using a friction energy wear approach while the third body layer was progressively established by loading a "bell" thickness profile. The constant (γ) and parabolic distribution (γ_x) of third body conversion factors were examined. Quantitative comparison with experimental results confirmed the interest of this new numerical strategy. Maximum wear depth, which was underestimated by nearly -80% using the wear model without taking account of the third body, was perfectly predicted, with error less than 10%, using the γ_x third body model. Comparison with longer tests, up to 30,000 fretting cycles, confirmed the stability of the approach which, in addition to providing very good prediction of fretting scar profiles allows more conservative prediction of fretting crack risk.

1. Introduction

Fretting is a friction solicitation occurring when small relative oscillatory movements appear between two bodies in contact. This phenomenon leads to both wear and fatigue damage, depending on the sliding conditions [1,2]. Above the transition between partial and gross slip, full sliding induces significant friction dissipation which promotes surface wear by debris formation and ejection [3]. Fretting appears in many industries and applications where vibrations occur. Fretting wear can become a severe problem, as it reduces the lifetime of structural parts (e.g. turbine engine dovetail interfaces [4], gears, steam generator tubes [5], etc.). Hence there is a crucial interest to simulate and predict fretting wear damage in order to predict surface profile evolution [6]. Many wear models can be considered for fretting wear damage [7]. Two main strategies are currently adopted. The first consists in applying the Archard wear description, which relates total wear volume to the product of normal force and fretting sliding distance [8]. This approach does not consider the coefficient of friction which was shown to be a key factor of stress-strain loading in the interface [1]. To alleviate this limitation, a friction energy wear approach can be adopted where wear volume extension is related to accumulated friction energy [9,10]. Better and more stable wear predictions are achieved for varying friction conditions. However, if constant friction conditions are observed, the Archard and friction energy approaches are equivalent. These two quantitative descriptions do not explicitly consider the debris layer entrapped in the interface. This "third body" layer can drastically modify surface wear damage. Godet followed by Berthier and co-authors, described this aspect in detail, developing "third body theory" [11–14]. They demonstrated that wear rate can be expressed as a function of debris formation flow from the first bodies and debris ejection flow from the interface. A major conclusion of this model is that 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 and appears very relevant to describing contacts subject to small fretting sliding which tends to maintain the wear debris inside the interface. However, the global formulation provided by the authors requires numerous variables, which are usually not easy to identify.

* Corresponding author.
E-mail address: siegfried.fouvry@ec-lyon.fr (S. Fouvry).

http://dx.doi.org/10.1016/j.wear.2017.01.063

If global wear volume prediction is still an open question, prediction of worn profiles is even less advanced. Indeed, prediction of local wear depth requires a local description of surface wear processes. Simple analytical formulations were first introduced to compute the local friction energy density distribution in order to predict the fretting wear endurance of thin hard coatings or solid lubricants [9]. However, this approach did not consider the evolution of worn geometry over time, and therefore it was limited to very thin layers [15]. More advanced semi-numerical simulations were successively developed to simulate deeper worn profiles [16]. In finite element modeling (FEM strategy) which is the most common approach in mechanical design, a first approach consisting in translating the surface nodes to simulate fretting surface wear was introduced by Leen and co-authors [17]. This strategy was completed by Mary et al. with a friction energy density approach, including bilateral wear on the two fretted counterparts [18]. However, a major limitation of these approaches was that they did not consider the presence of a third body layer entrapped in the interface. Gosh et al. [19] and Basseville et al. [20] 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 FEM models. Some little particles are modeled to observe the influence of the third body on stress fields and contact plasticity. Other models involving coupled FEM and DEM (discrete element method) analysis were proposed by Haddad et al. [21] and Leonard et al. [22] to evaluate the influence of the third body on friction, contact pressure and stress fields. Finally, Ding et al. first introduced a model taking account of the third body in wear simulations, by assuming a stuck layer added onto the plane surface [23]. More recently Yue et al. in a parametric investigation, showed the effect of a debris layer implemented between the fretting interfaces [24].

A major conclusion of this research was to show that the elastic properties of the third body layer do not significantly influence the pressure profile distribution. However, in assuming constant third body layer thickness over the whole fretted interfaces the authors suggested that the debris layer does not significantly modify 2D surface morphology. These results tends to contradict previous experimental comparisons showing that a surface wear modeling which does not consider the present of a debris layer tends to overestimate lateral extension and underestimate maximum wear depth [23,25,27]. To avoid this limitation, the present FEM simulation took account of the dynamic evolution of third body thickness combined with a bilateral friction energy wear simulation of cylinder and plane counterparts.

Local third body thickness was computed by iterative summation of a proportional part of the wear thickness generated on both plane and cylinder surfaces. Wear debris transferred to the third layer was quantified using a third body conversion factor, γ. Both constant and parabolic approximations of the distribution $\gamma_{(x)}$ were considered. The interest of this enriched third body friction energy wear approach is to provide a “bell” distribution of third body thickness and potentially more realistic surface wear profiles by localizing the shear work at the center of the contact, increasing the maximum wear depth compared to the lateral surface wear extension. This new numerical schema is compared to a well calibrated Ti-6Al-4V cylinder/plane experimental data.

2. Experiment

2.1. Material

The model was calibrated using a homogeneous Ti-6Al-4V/Ti-6Al-4V cylinder plane contact. This titanium alloy, made up of a 60% alpha – 40% beta microstructure, is extensively used in the aeronautics industry, especially for blades and disk components. 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 $\sigma_{Y0.2\%}$ (MPa)
Ti-6Al-4V	120	0.3	360	880

2.2. Plain fretting set-up

Plain fretting tests were carried out using a tension-compression MTS hydraulic system [25]. Normal force (F_n) was held constant while tangential force (F_t) and displacement (δ) were recorded (Fig. 1). For the present 2D cylinder/plane configuration, contact stress and damage are better described by forces per unit length [1], respectively:

$$P = F_n/L \text{ and } Q = F_t/L \tag{1}$$

with L (mm) the transverse width of the cylinder/plane contact which is adjusted to satisfy a plain strain hypothesis along the median sliding axis.

The fretting loop was plotted and corresponding values for F_t^*, Q_t^* and δ^* were extracted. The friction energy Ed (J) was determined by integrating the δ- Q loop.

After the test, the mean values of δ^* and Q^* were computed, as well as the accumulated friction energy ΣEd integrating the dissipated friction over the entire test duration. This analysis focused on a single test condition, 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. $a_H/L < 0.16$). Sliding amplitude was monitored at $\delta_g^* = \pm 75$ μm. Note that the δ_g^* value corresponds to the residual displacement measured when Q = 0 N, to avoid the artefacts related to test-system compliance. An averaged description of friction behavior during the fretting cycle was achieved using a “friction energy coefficient” [9].

$$\mu_e = \frac{Ed}{4 \times \delta_g^* \times P} \tag{2}$$

For the studied loading condition, the friction energy coefficient was μ_e = 0.56. Frequency was decreased to f = 0.11 Hz in order to limit adhesive wear phenomena and to ensure almost exclusively quasi-pure third body abrasion wear process, as previously described [26].

Four test durations of 5000, 7500, 10,000 and 30,000 fretting cycles were implemented. The shortest test was used to calibrate the model, and the longest ones were considered to establish the stability of the model.

2.3. Fretting wear analysis

After the test, specimens were cleaned for 30 min in an ultrasonic ethanol bath to remove most of the non-adhering debris contained 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 were measured on the plane (V_p) and on the cylinder (V_c) (Fig. 2).

The corresponding energy wear rates [9] were estimated assuming linear evolution of wear extension and equal distribution of the friction energy toward plane and cylinder surfaces, so that: