Tribology International 119 (2018) 795–808

Contents lists available at ScienceDirect

Tribology International

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

Thermomechanical modelling of dry friction at high velocity applied to a Ti6Al4V-Ti6Al4V tribopair

M. Coulibaly [a,*], G. Chassaing [a,b]

[a] Laboratoire de mécanique Biomécanique, Polymère, Structures (LaBPS), EA4632, Ecole Nationale d'Ingénieurs de Metz (ENIM) de l'Université de Lorraine, 1 route d'Ars Laquenexy, CS65820, 57078 Metz Cedex 3, France

[b] Safran Aircraft Engines, groupe SAFRAN, Réau, Rond-Point René Ravaud, 77551 Moissy-Cramayel, France

ARTICLE INFO

Keywords:
Semi-analytical model
Dry friction
High velocities
Green's function

ABSTRACT

A semi-analytical model is developed to predict thermal and mechanical responses of the contact surface during the dry friction of Ti6Al4V tribopair at high velocities. This model is achieved by solving thermal and mechanical problems separately, before coupling both interdependent solutions using the dissipation of mechanical work into heat. The thermal problem is solved by Green's function techniques, while the mechanical one is treated by considering the adiabatic shearing process of contact asperities. The accuracy of this new semi-analytical model is first evaluated by comparison with the results of a two-dimensional numerical finite element simulation. Then, the proposed model is adapted to the specific dry friction process considered in this study by means of experimental testing with post-mortem microstructural analysis.

1. Introduction

Friction is a natural process occurring in all physical events involving a relative motion or any mechanical interaction between at least two systems in contact. Given the multitude of physical cases involving friction, knowledge and mastery of friction process are essential and have always been considered as one of the major issues in mechanics since the beginning of fifteenth century. It is actually at this period that Leonardo da Vinci set the fundamental laws of friction that mean the starting point of tribology. Two centuries later, Amontons (1699) established the friction force opposite to the relative motion of two solids in contact, while Coulomb (1781) introduced the ratio between friction and normal forces as the coefficient of friction, which can be static or kinetic [1]. However, the three fundamental laws established by the pioneers cannot be applicable to any material, surrounding conditions and type of loading [2]. In fact, the friction process turns out to be much more complex and depends, not only on the normal pressure, but also on the surface roughness, the sliding velocity and the temperature.

Several theories have been formulated to demonstrate the surface roughness influence on friction. This latter aspect of friction has been widely taken into consideration in the literature. Bowden and Tabor [3] interpret the friction as the formation then the failure of mechanical junctions, acting as weld joints, at the contact points between the concerned sliding surfaces. This assumption implies a distinction between the "apparent contact surface" and the real integral area of contacts, so-called "true contact surface" [4–8]. Furthermore, several experimental studies showed that the coefficient of friction could change for a constant value of normal force due to a change of the apparent contact surface, thus pointing out the normal pressure effects. Montgomery [9] clearly emphasised the friction dependency on the normal pressure through its work on the wear of cannon stocks. Thereafter, many contributions [10–13] addressed the normal pressure influence by examining the variation of the true contact surface ratio to the apparent one. Through the last centuries following Coulomb's law stipulating that coefficient of friction does not depend on the sliding velocity, experimental investigations, however, revealed the opposite. Indeed, according to the tribological literature [7,9,14–19], the coefficient of friction increases with velocity up to a critical value, then decreases with speed. In addition, the mechanical work of friction force is almost entirely converted into heat [20]. Moreover, some studies as Kennedy's one [21] showed that about 95% of this dissipated energy are concentrated within a layer of 5 μm depth from the contact surface. This latter conclusion confirms the analysis of Bowden and Ridler [22] who observed that two metallic solids rubbing against each other remain macroscopically cool, while their local temperatures at the contact can reach very high values (sometimes over 1000 °C). Therefore, this inevitable temperature rise may cause a multitude of phenomena that directly impact on the friction process: mechanical softening [23], oxidation, phase transformation,

* Corresponding author.
E-mail address: mamadou.coulibaly@univ-lorraine.fr (M. Coulibaly).

https://doi.org/10.1016/j.triboint.2017.12.004
Received 16 July 2017; Received in revised form 14 November 2017; Accepted 3 December 2017
Available online 7 December 2017
0301-679X/© 2017 Elsevier Ltd. All rights reserved.

wear, etc.

Nowadays, the prediction of the friction force and local temperature reached at the contact interface still remains an industrial and academic challenge on the road to the mastery of friction. To this end, modelling of dry friction requires a thorough knowledge of the major parameters influences previously mentioned. Furthermore, for high sliding velocities (over 1 m.s^{-1}), the modelling must be carried out by combining mechanical and thermal approaches because of the strong interdependence occurring between stress distribution and temperature profile at sliding contact. However, most of the contributions in this field study the friction process by considering only the thermal aspect, assuming that the heat flow produced by mechanical energy dissipation during friction is previously known. In this way, Ettles [24] uses Blok's theory [25] to propose a calculation method of the friction coefficient as function of the decomposition temperature. Comparison between calculated values and experimental measurements reveals that the model of Ettles is limited by considering a time-constant and uniform generated heat flow. Jaeger [26] and Carslaw and Jaeger [27] introduce the Green's techniques to determine the temperature rise at any point surrounding the contact due to moving heat sources located on the contact surface. From these latter reference works, Lim et al. [7] present a model using an empirical law of friction – coefficient of friction – which depends only on the sliding velocity. Assuming steady state temperatures, unidirectional heat diffusion and mean asperity radius in the order of 10^{-5} m allows the authors to evaluate the mean and flash temperatures on the circular contact surface. In addition, Vick and Furey [28] also use Green's techniques [26,27] to examine the sliding contact between two semi-infinite solids. Through their model, they emphasise the temperature rise concentration within the contact asperities and the heat accumulation at the contact rear relative to the sliding motion. These thermal approaches certainly provide a deeper understanding of dry friction at high velocities, but are restricted to the resulting heat generation and transfer without taking into account the mechanical position.

For a complete prediction of the friction mechanism, it is necessary to connect the thermal process with the mechanical friction behaviour. In order to deal with the complexity of thermomechanical coupling, only a few attempts at modelling have been achieved to date. Thus, Molinari et al. [8] propose an analytical model based on the adiabatic shear banding of the contact junctions (or asperities) at high sliding velocities. In this work, the thermal problem is solved by assuming a case of unidirectional heat diffusion across a depth equivalent to the distance covered by the thermal front, during the passing of an asperity. This model has the main advantage of being applicable to any couple of solids by modifying the shear constitutive relation. However, determination of some influencing parameters, such as coefficients of heat distribution and fractional area of adhesive junctions, still needs to be achieved in order to obtain more relevant results. Then, by running Molinari et al.'s solving scheme [8] with a thermo-viscoplastic constitutive relation on numerical simulation software “Abaqus”, Voyiadjis et al. [29] notice a higher level of shear stress than Molinari et al.'s one [8] for the same couple of steels CRS 1018 vs CRS 1018. More recently, Coulibaly et al. [30] propose an analytical model by combining the Green's techniques of Carslaw and Jaeger [27] to express the thermal response due to friction, and the adiabatic shear banding condition of Molinari et al. [8] to describe the frictional shear behaviour. The thermomechanical coupling is achieved by using a homogenisation approach and leads to the prediction of the heat flux density distribution, temperature profile of both solids in contact and local shear stress field for any friction time.

The aim of this present work is to adapt the model previously developed by Coulibaly et al. [30] to the extreme conditions of dry friction of a Ti6Al4V tribopair occurring at blade-disc mechanical bond in turbojet engines. The first part is devoted to the description of the tests and measurements carried out to provide the experimental results of the study. In the second part, the semi-analytical modelling is proposed by first examining sliding conditions and thermal process, then presenting the thermomechanical coupling, and at last performing the adaptation of Coulibaly et al.'s model [30] to the Ti6Al4V tribopair. In the last part, some conclusive results of the applied model are depicted through comparisons and uses with experimental data in order to assess its predictive capacity, reliability and accuracy.

2. Experimental measurements

In order to reproduce sliding process at high velocity with a quasi-instantaneous loading, a specific tribometer has been used [31–34]. The friction is generated by the impact of a projectile on a specimen. It allows the investigation of the friction for a large range of normal pressure (8 to 280 *MPa*) and sliding velocity (quasi-static to 130 $m.s^{-1}$).

Although its working principle was precisely detailed in previous study [34,35], a short description of this present tribometer is proposed here to guarantee greater clarity. The tribometer is composed of two parts: a dynamometer ring used to apply and measure normal force N, and a load sensor dedicated to measure tangential force T. Fig. 1 illustrates the sliding configuration: a central specimen and two pads are inserted in the instrumented dynamometer ring (not shown here). Because dynamometer clamping causes the pressure application, sample dimensions (width of the specimen and height of the pads) accordingly define the normal force level. A projectile then hits the central moving specimen whereas two pads remain fixed to the ring and stationary. Once the impact occurs, sliding begins and progresses until the specimen completely pass the pads. An instrumented load sensor, that supports the pads, measures the tangential force T (i.e. friction force). The travels of moving specimen and projectile are stopped in a receiving tank. Sample dimensions are defined in Fig. 1. The sliding length is equal to 60 *mm* and the apparent contact surface to 30 mm^2.

In this research work, the tribometer has been adapted to be installed on a ballistic bench and a drop tower. In both cases, signals are recorded sample rate of 500 *kHz* with a DEWETRON device and only a low-pass filter at 10 *kHz* is applied.

The ballistic bench has also been presented in previous paper [34]. Cylindrical projectile with hemispherical tip is propelled by means of high-pressure gas gun. Due to mechanical work, specimen velocity is continuously decreasing during interaction. An ultra-high speed camera (Shimadzu HPV2) is used to monitor specimen position all along friction process so that the actual velocity can be estimated at any time. In studied configurations, initial velocity V_i ranges from 40 to 68 $m.s^{-1}$. Fig. 2 presents typical signals acquisition for initial velocities V_i higher than 40 $m.s^{-1}$. Despite severe conditions, the normal force (i.e. black curve) remains relatively constant until its drop (at around 1750 μs, that indicates the last 10 *mm* of sliding [34]. The friction forces measured at each side of the load sensor (i.e. blue and red curves) are firstly rapidly increasing due to impact, then decreasing to stabilize for most of the test duration, and finally reaching zero (end of interaction). Following classical Coulomb's law, friction coefficient was plotted in green line during stationary time span (i.e. from 415 to 1505 μs) in order to calculate the mean friction coefficient $\overline{\mu}$ (COF) of the interaction.

The second equipment is a drop tower from INSTRON manufacturer. It has been used to perform tests at 8 $m.s^{-1}$ constant velocity. Holding a

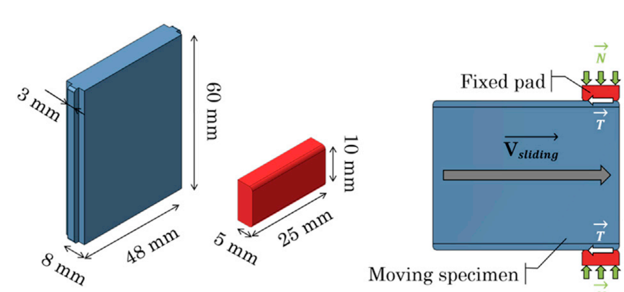

Fig. 1. Parts dimensions and sliding configuration.

Download English Version:

https://daneshyari.com/en/article/7002365

Download Persian Version:

https://daneshyari.com/article/7002365

Daneshyari.com