Application of an energy wear approach to quantify fretting contact durability: Introduction of a wear energy capacity concept

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Abstract

A friction energy formalism is considered and adapted to formalize the fretting wear responses of adhesive wear and non-adhesive wear interfaces. It is shown that for non-adhesive wear tribocouples like hard coatings (TiN, TiC, etc.) the wear kinetics can be formalized using the accumulated friction dissipated energy. By contrast, adhesive wear contacts involving aluminium and titanium alloys display a critical dependance regarding the applied sliding amplitude. The wear kinetics of such systems is captured by considering a sliding reduced energy wear formulation. A combined composite energy wear formulation is then introduced to formalize the fretting wear response whatever the tribocouple behaviour. It is shown that a local approach, focusing on wear depth analysis, is required to predict interface durability. A FEM investigation demonstrates that the wear depth kinetics can be predicted by considering the accumulated energy density. It concludes that interface durability can be related to a single energy density capacity variable ($\chi$) defined as the maximum accumulated energy density which can be dissipated in the interface before contact failure.

Keywords: Fretting; Wear; Ti–Al–4V; TiC; Coating durability; Wear modelling

1. Introduction

The term fretting denotes a small oscillatory movement between contacting surfaces which invariably occurs in engineering assemblies subjected to vibrations. Depending on the loading conditions (relative displacement amplitude, normal loading), fretting may cause damage by surface fatigue involving crack nucleation and crack propagation and/or wear induced by debris formation [1,2]. Indeed, it has been shown that the fretting damage is directly connected to the stabilized fretting sliding condition (Fig. 1).

For the smallest displacement amplitudes the contact stabilizes at the so-called partial slip condition. The contact then displays a composite structure of sticking and sliding zones [3,4]. The fretting loop defined by the tangential force evolution ($Q$) versus the applied displacement ($\delta$) is characterized by a closed elliptic shape. This sliding condition mainly favours crack nucleation and crack propagation.

For higher displacement amplitudes, the sticking zone no longer exists and the entire contact is subjected to sliding. The friction dissipation activates wear mechanisms involving debris formation and debris ejection. The fretting loop is then characterized by a rectangular shape with a maximum tangential force amplitude ($Q^*$) verifying Amonton’s law of friction ($Q^* = \mu P$). A Fretting map strategy has been introduced to format this fretting damage evolution [5–8].

Due to its dramatic impact on structural integrity, fretting cracking has been extensively investigated during the past decades [8–14]. Fretting crack nucleation and crack propagation can now be predicted, while fretting fatigue endurance can be formalized. The situation is less advanced as regards fretting wear processes [15]. This is revealed by the great number of wear models and the difficulty in predicting the evolution of wear with loading parameters like pressure, sliding or friction coefficient. There is nevertheless an increasing interest in predicting...
fretting wear better [16–18]. Indeed, one strategy used to reduce cracking risk is to apply low friction surface treatment and impose gross slip conditions. As material is progressively removed by wear, however, the palliative will be worn away and the loading will shift to the dangerous partial slip regime. The cracking phenomenon is then only delayed by the wear of the surface treatment [7]. Hence the cracking response for such a specific condition depends on the fretting wear process. In the present work, global and local approaches have been developed to quantify the fretting wear processes. To rationalize the analysis, a single friction energy approach, where the wear material removal is related to the friction energy dissipated through the interface, has been considered.

A sequential approach is then adopted focusing on the following three aspects:

- How can the energy wear approach formalize the general wear response of tribocouples displaying either adhesive or non-adhesive wear behaviours?
- How can the local wear depth kinetics be predicted and which parameter controls the surface profile evolution?
- How can the surface durability be predicted and how can it be formalized through a single energy wear description?

2. Experimental procedure

2.1. Test procedure

To investigate fretting wear phenomena, a complete and detailed analysis of the loading parameters involved in the damage process must be made. Fig. 2a illustrates the principle of the LTDS fretting rigs developed at different scales to analyse micro-meso- and macro- contacts (i.e. for normal loading comprised between 0.5 and 30 000 N) [19–21]. Various contact geometries like cylinder/plane, sphere/plane and plane/plane configurations can be adjusted to investigate the contact size effect on the wear law stability. The normal force ($P$) is kept constant while the tangential force ($Q$) and displacement ($\delta$) are recorded. A reciprocating movement with a constant speed is...
imposed. Thus the fretting loop $Q-\delta$ (Fig. 2b) can be drawn to extract quantitative variables, including the dissipated energy ($E_d$) (i.e. the area of the hysteresis), which is also related to the friction work dissipated per fretting cycle, the sliding amplitude ($\delta_g$) (i.e. the residual displacement when $Q = 0$), the tangential force amplitude ($Q^*$) and the displacement amplitude ($\delta^*$).

As illustrated in Fig. 2b, due to ploughing effects the conventional friction coefficient (i.e. $\mu = Q^*/P$) can overestimate the global friction behaviour. An energy friction coefficient has been introduced to average the friction response [16]:

$$
\mu_e = \frac{E_d}{4P\delta_g}.
$$

(1)

During a test, constant but also variable displacement amplitude sequences can be imposed to investigate the wear model's stability [20,21]. To obtain a dynamic overview of the fretting test, the fretting cycles are superimposed, resulting in the fretting log (Fig. 2c). Note that in the present study only gross slip conditions have been applied. Ambient conditions and more particularly the relative humidity play a critical part in wear processes. Specific precautions involving closed chamber, RH measurements and a controlled humidity system were systemically taken.

2.2. Materials

To evaluate the stability of the proposed energy wear approach, two different contact situations involving adhesive wear and non-adhesive wear tribocouples have been investigated.

- To investigate the fretting wear response of non-adhesive wear tribocouples, TiC, VC, TiN, TiCN monolayer and TiC/VC, (TiC/VC)$_{x}$ multilayer CVD coatings deposited on a high speed steel VANADIS 23 have been tested against a 12.7 mm radius polycrystalline alumina ball. The mechanical and surface properties of the materials are summarized in Table 1.

- To analyse the wear response of adhesive wear tribocouples, the homogenous Ti–Al–4V/Ti–Al–4V in-

interfaces, well known to induce severe transfer and seizure phenomena, were investigated. Macro-cylinder/plane configurations with a cylinder radius of 10 mm were tested. The mechanical and surface properties of the studied Ti–Al–4V contact are compiled in Table 2.

2.3. Test methodology and damage analysis

The test procedure as well as the wear damage quantifications significantly impact on the quantitative analysis. The following methodology was applied. Prior to testing, specimens were cleaned with acetone and ethanol to remove dust and hydrocarbons from the surfaces. After the test, specimens were ultrasonically cleaned with alcohol for at least 10 min to remove the debris trapped in the fretting scars. The wear volumes of the fretting scars, denoted as ($V$), were estimated from axial and cross 2D surface profiles combined with suitable formulations to define the wear volumes [20,21]. Due to the very slight wear of the ball and the absence of native transfer of the coating, only the wear volume of the hard coating (i.e. the plane fretting scar) was considered.

Concerning the Ti–Al–4V interface, both wear volumes on the plane ($V_P$) and the cylinder ($V_C$) were identified and added together to estimate the total wear volume variable ($V_T$) which was considered in the present investigation [21].

3. Development of a unified energy wear approach

3.1. Fretting wear analysis of non-adhesive wear tribocouples

The various hard coatings were tested under constant 100 N normal loading (i.e. initial Hertzian pressures above
1 GPa), \( \pm 50 \) and \( \pm 100 \mu m \) displacement amplitudes and test durations from 5000 up to 50 000 cycles with constant 5 Hz frequency and 50% relative humidity. Constant but also variable displacement amplitudes were applied, in which test durations, characterized by a total number of fretting cycles \( (N) \), were divided into one, two and four blocks of \( \pm 50 \) and \( \pm 100 \mu m \) displacement amplitudes defining the studied \( (50/100)_{\mu m} \times 1 \), \( (50/100)_{\mu m} \times 2 \) and \( (50/100)_{\mu m} \times 4 \) sequences, respectively \([20]\). Fig. 3 confirms the linear evolution of the wear volume \( (V) \) with the accumulated dissipated friction energy \( (\Sigma Ed) \). The accumulated dissipated energy corresponds to the sum over the whole test duration \([16]\):

\[
\Sigma Ed = \sum_{i=1}^{N} Ed_{(i)},
\]

with \( Ed_{(i)} \), the friction energy dissipated at \( i \)th fretting cycle.

The coating wear rate can then be expressed by the following equations:

if \( \Sigma Ed < Edth \) then \( V = 0 \),

if \( \Sigma Ed > Edth \) then \( V = \alpha (\Sigma Ed - Edth) \),

with \( Edth \) being the threshold transition energy required to first transform the material before generating wear particles and \( \alpha \) the energy wear volume coefficient. It has been shown that the \( \Sigma Ed \) values for a ceramic tribocouple, which does not involve plastic deformations, can be neglected. The wear equation is then simplified to

\[
V = \alpha \Sigma Ed.
\]

For the tested TiC/alumina tribocouple, we deduced \( \alpha = 415 \mu m^3/J \) \([20]\) with a linear coefficient near 0.90. The energy wear coefficients extracted for the other studied coatings are compiled in Table 3. All the correlation parameters \( (R^2) \) obtained for the Ti–V–C systems are above 0.9. The Ti–C–N systems display slightly lower linear correlation values. The conclusion is that the proposed energy wear approach appears to be a pertinent and reliable approach to rationalize the fretting wear kinetics of non-adhesive wear contacts.

### 3.2. Fretting wear analysis of adhesive wear tribocouples

The Ti–Al–4V/Ti–Al–4V interface was investigated considering a key point reference loading condition defined by a \( P = 133 \) N/mm linear normal force (equivalent to a 525 MPa maximum Hertzian pressure), a \( \pm 75 \mu m \) displacement amplitude and a 25 000 fretting cycles test duration. Based on this reference loading condition the impact of pressure has been evaluated by ranging the normal loading from 66 to 333 N/mm, the sliding by ranging the displacement amplitude from \( \pm 25 \) to \( \pm 150 \mu m \) and the test duration by ranging the fretting cycles from 10 000 to 50 000. Variable sliding conditions combining 25 and 75 \( \mu m \) homogeneous and heterogeneous sequences have also been studied.

Fig. 4 clearly outlines the non-linear dependance of the wear volume of the Ti–Al–4V/Ti–Al–4V interface versus the accumulated dissipated energy \([22]\).

A significant influence of the applied displacement amplitude is observed. Unlike hard coatings–alumina interfaces, metal–metal interactions like titanium or aluminium alloys will favour seizure and adherent metal transfer phenomena. It must be outlined that the fretting wear kinetics is controlled by debris formation but also by debris flow ejection. In the case of non-adhesive wear tribocouples, the debris are easily ejected from the interface and a direct correlation with the dissipated energy can be considered. For adhesive wear tribocouples, a significant quantity of the dissipated energy is consumed by the debris ejection process. An increase of the applied sliding amplitude will promote an increase of the debris ejection rate and consequently of the energy wear rate. To capture this effect, a modified sliding reduced energy wear formulation is introduced \([21,22]\):

\[
Ed_{(i)} = \frac{\delta_{(i)}}{\delta_{(ref)}} Ed_{(i)}.
\]
In this formulation, the dissipated energy of the $i$th cycle ($Ed_{i(j)}$) is factored by the applied sliding amplitude ($\delta_{gref}^i$). In other words, the higher the sliding amplitude, the better the debris flow ejection and consequently the better the energy wear efficiency. However, the formulation must also confirm the physical relationships linking the dissipated energy to a quantity of damaged material. One solution consists in normalizing the imposed sliding amplitude by a reference value associated, for instance, to a representative key point loading condition ($\delta_{gref}^i$). Like for the plain energy variable, an accumulated sliding reduced energy is deduced:

$$\Sigma Ed_i = \sum_{j=1}^{N} \delta_{gref}^i Ed_{i(j)}.$$  \hspace{1cm} (6)$$

Fig. 5 compares the total wear volume of the studied Ti–Al–4V/Ti–Al–4V interface versus the corresponding accumulated sliding reduced energy.

Although constant but also complex variable sliding conditions have been integrated, a stable linear evolution (i.e. $R^2 = 0.95$) can be observed. It therefore justifies the following energy wear formulation to formalize the fretting wear behaviour of adhesive wear tribocouples:

$$V_T = \alpha \Sigma Ed_i.$$  \hspace{1cm} (7)$$

3.3. Synthesis of the energy wear formulation

The previous paragraphs show that the wear kinetics of a non-adhesive wear tribocouple can be quantified through a plain friction energy formulation, whereas adhesive wear tribocouple behaviours are better formulated by a sliding reduced energy description. This suggests that, with some minor optimizations, the energy wear formulation is able to capture the wear behaviour of very different tribosystems. Most tribological applications are in fact between these two configurations. Hence, it could be interesting to consider a general formulation to express the wear response whatever the adhesive wear properties of the interface. The following parametric relations could be introduced:

$$Ed_{i(j)}^{(\lambda)} = Ed_{i(j)} G \left( \frac{\delta_{gref}^i}{\delta_{gref}^0}, \lambda \right).$$  \hspace{1cm} (8)$$

with $Ed_{i(j)}^{(\lambda)}$, the composite energy dissipated at the $i$th cycle, and $G$, the function expressing the relative impact of the sliding amplitude on the energy formulation as a function of the non-dimensional variable $\lambda$ (i.e. $\lambda \in [0,1]$) which indexes the adhesive wear property of the contact. The two extreme conditions also need to be verified which implies that

if $\lambda = 1$ (full adhesive wear contact):

$$G \left( \frac{\delta_{gref}^i}{\delta_{gref}^0}, \lambda \right) = \frac{\delta_{gref}^i}{\delta_{gref}^0} \Rightarrow Ed_{i(j)}^{(1)} = Ed_i,$$  \hspace{1cm} (9)$$

if $\lambda = 0$ (non-adhesive wear contact):

$$G \left( \frac{\delta_{gref}^i}{\delta_{gref}^0}, \lambda \right) = 1 \Rightarrow Ed_{i(j)}^{(0)} = Ed.$$  \hspace{1cm} (10)$$

Assuming a weighed contribution of the energy and sliding reduced energy components, a first linear expression can be introduced:

$$G \left( \frac{\delta_{gref}^i}{\delta_{gref}^0}, \lambda \right) = 1 - \lambda + \lambda \frac{\delta_{gref}^i}{\delta_{gref}^0}.$$  \hspace{1cm} (11)$$

Triboiogical processes are complex, and alternative formulations should be considered. However, this conjuncture, still
under construction, highlights the important parameters, like, the adhesive properties of the interface, the interface tenacity, etc. which must be considered to achieve a unified energy wear formulation of fretting wear.

4. Surface durability

As illustrated in Fig. 6, predicting surface durability requires a local wear description based on a wear depth formulation [7]. Assuming a macro–micro passage of the energy wear approach, our strategy will therefore consist in making a first approximation to relate the extension of local wear depth to the corresponding dissipated energy density. Alternatively, if the maximum wear depth is located at the centre of the fretting scar, our analysis will plot the maximum wear depth as a function of the accumulated friction energy density dissipated at the centre of the interface.

One major difficulty of tribology is that the damage process (i.e. the surface wear) directly affects the loading distributions. The progressive wear of the interface will alter the contact geometry and consequently modify the pressure, shear and finally friction energy distributions. A coupled iterative approach is therefore required to have a pertinent description of the local wear and surface durability [17].

Our strategy is first to develop a complete FEM simulation of the surface wear profile evolutions, and then, focusing on the maximum density value located at the centre of the contact, to derive simplified analytical formulations in order to apply the “wear depth—accumulated energy density” correlation to the experiments.

4.1. Application of an iterative FEM wear simulation

Different analytical approaches, based on complex mathematical formalisms, could be considered to capture the surface wear evolution [23]. The FEM method appears to be a simple and convenient approach which can also integrate plasticity effects. Extending previous developments introduced by McColl and co-authors [17], a specific FEM wear box combining Matlab and Abaqus codes has been designed [24]. Fig. 7b illustrates the 2D cylinder/plan meshing of the elastic plane strain model. It is defined by a $R = 10\, \text{mm}$ radius half-cylinder shape pressed and a $10\, \text{mm}$ cubic edged plane. Both plane and cylinder display similar elastic properties, and no coatings are considered. The contact modelling is driven by a master (cylinder)/slave (plane) algorithm controlled by a Lagrange multiplier contact/friction formulation. A detailed description of the model is provided in reference [24]. An iterative procedure was implemented to remesh the interface after each computed fretting cycle, thus providing a representative evolution of the contact geometry. This FEM model does not integrate the presence of the third body, which implies that it is currently restricted to non-adhesive wear interfaces. The plain energy wear formulation was derived and implemented to monitor the surface profile evolution:

$$h(x) = \pi \Sigma \text{Ed}(x),$$

with $h(x)$, the local wear depth at the $x$ position of the interface.

Note that a specific surface refining of the model up to a $5\, \mu\text{m}$ long mesh and involving at least 60 surface elements in the interface, was considered to better integrate the surface’s evolution. As simulation would take too long, the experimental energy wear factor was multiplied by an acceleration factor of 100 to simulate in 200 computed fretting cycles the wear profile obtained after 20 000 experimental cycles. The given acceleration factor has been optimized through an adequate parametric study [17]. At this stage of the wear box development only the plane surface meshing could be modified which implies that the wear surface profile modification is totally applied on the plane surface (i.e. the cylinder surface is not modified). The numerical investigation was performed for a $\delta_{el}/a_{el} = 0.4$ gross slip sliding condition, a 0.8 coefficient of friction and an energy wear factor arbitrarily fixed at $\sigma = 10^2 \mu\text{m}^3/\text{J}$. To rationalize the investigation and simplify the application for other contact configurations, a normalized representation based on the undamaged interface was adopted: the half width of the contact $a_{el}$ is normalised by the initial Hertzian value $a_{el} = a_{el}(0)$; the pressure profile $P_{el}(x)$ by the Hertzian pressure $P_{el}(0)$; and the energy density profile $\text{Ed}(x)$ by the maximum energy density value.
dissipated during the first cycle at the centre of the interface $E_d(0) = E_d(1)(0)$.

Fig. 7 plots the normalized evolution of pressure and surface profiles at different stages. A significant increase of the contact area related to a sharp decrease of the maximum pressure is observed. The very fast transition from the initial Hertzian distribution towards a quasi-squared homogeneous pressure field can be noticed.

The simulated fretting wear profile displays a typical smooth U wear shape whose maximum wear depth is always located at the centre of the interface ($x = 0$). This is consistent with experimental fretting scars for which the third body has a minor impact on the interface morphology (Fig. 8).

Fig. 9 shows the simulated wear depth kinetics ($h(x)$), and the corresponding energy density ($E_d(x)$) profiles at different stages of the interface wear.

The applied numerical analysis infers the following incremental formulation:

$$\dot{h}(x) \approx \frac{\Delta h}{\Delta N} = \frac{h_{i+1}(x) - h_{i}(N-\Delta N)(x)}{\Delta N},$$

with $\Delta N$ equal to one numerical fretting cycle or presently equivalent to 100 experimental fretting cycles. As expected, a significant decrease of the local wear kinetics is observed. The wear depth kinetics is presently reduced by a factor of 10 from the beginning of the test to the 20 000th fretting cycle.

Another interesting aspect is the surface wear kinetic profile evolution. After a short transition (i.e. here less than 100 cycles), the wear kinetics converges toward a quasi-homogeneous distribution. Except for the external borders, influenced by lateral ploughing effects and typical fluctuations induced by the contact exposure, the central part of the fretted interface is characterized by homogeneous (i.e. flat) fretting wear kinetics and consequently homogeneous (i.e. flat) distributions of the energy density.
This assumption is consistent with the idea that a stabilized wear regime will invariably converge to a homogeneous wear kinetic profile and consequently a continuous wear profile. Indeed, assuming a heterogeneous surface profile, the highest local wear kinetics will be activated at the maximum pressure peaks and the lowest at the minimum pressure peaks. Hence the initial surface discontinuities will be progressively erased and the interface will progressively converge toward a continuous wear profile and homogeneous wear kinetic profile.

4.2. Development of a simplified energy density formulation

One consequence of this evolution is the flattening of the pressure field from an elliptical Hertzian distribution toward a quasi-square profile. By focusing on the pressure field from an elliptical Hertzian distribution (Fig. 10a), one can observe an activated transition from an elliptical to a square distribution of the pressure profile, a two-stage evolution can be considered (Eq. 10).

(a) Elliptical approximation of the pressure profile ($a(i)<a(i_0)$): During the initial period the surface degradation is taken to be small enough to consider an elliptical hypothesis of the pressure distribution. The energy density dissipated at the centre of the interface under gross slip condition (i.e. $\delta_{E_0}<a(i)$) is then expressed by the explicit formulation [16] (Fig. 10a):

$$ Ed(i)(0) = 4\mu\epsilon(i)p(i)(0)\delta_{E_0} \left( \frac{1 - \epsilon^2}{4} + \frac{1}{2}\arcsin(\epsilon(i)) \right). $$

(b) Flat approximation of the pressure profile: Assuming a flat distribution of the pressure profile, the local energy density dissipated at the centre of the interface is then expressed by (Fig. 10b)

$$ a(i)p(i)(0) = a(i)p_0, $$

where

$$ e(i) = \delta_{E_0}/a(i). $$

with $Ed_{E_0}(0)$, the elliptical approximation of the maximum energy density dissipated at the centre of the interface during the $i$th fretting cycle; $a(i)$, the half width of the contact at the $i$th cycle and; $\epsilon(i)$, the exposure ratio defining the sliding behaviour of the interface. Note that the given formulation verifies the Hertzian situation, replacing $a(i)$ by $a_1$ and $p(i)(0)$ by $p_0$, respectively. The conservation of the total dissipated energy must be verified which implies that

$$ a_0 p_0 = a_1 p_{0(i)}/a_{0(i)}.$$

Fig. 10. Illustration of the energy density distribution defined from pressure field approximation: (a) elliptical assumption when the surface wear is limited ($a(i)<a(i_0)$); (b) flat pressure approximation when the surface is sufficiently worn to promote a homogeneous wear kinetics within the interface ($a(i)>a(i_0)$).
neglected and the accumulated energy density can be estimated by directly integrating the flat approximation:

$$\Sigma Ed(0) = \sum_{i=0}^{N} \left[4 \mu_{\rho(0)} \rho_{\rho(0)} \delta_{\rho(0)} \right] = \sum_{i=0}^{N} \left[ \frac{Ed_{0,i}}{S_{0,i}} \right]. \quad (22)$$

4.3. Prediction of the wear depth extension of non-adhesive wear tribocouples

Using the above simplified formulation (Eq. (22)), given for a sphere/plane configuration, a local investigation of the TiC/alumina wear depth extension was performed. The studied coating thickness, less than 2 µm, is near 100 times smaller than the contact radius, which justifies the application of the previous analytical formulations defined from a homogeneous contact hypothesis. Moreover, the final simplification, which considers the mean pressure approximation (Eq. (22)), can be applied whatever the contact configuration, assuming that the contact area is correctly estimated. The contact area $S$ was extrapolated by fitting the measured experimental fretting scar areas and combining adequate energy wear volume formulations [20]:

$$S = A (\Sigma Ed_{0,i})^{1/2} + B. \quad (23)$$

A geometric analysis shows that the square root energy factor is related to the surface extension induced by the wear of the spherical cap shape, whereas the $B$ parameter takes into account the initial Hertzian contact area. The parameter $A$ is a function of the contact geometry ($R$) and the wear kinetics ($x$). The following values were deduced for the studied TiC/alumina ($R = 12.7$ mm) tribocouple:

$A = 5300 \mu m^2 J^{1/2}$ and $B = 98000 \mu m^2$. A rather good correlation between the proposed formulation and the experiments is observed (Fig. 12).

Fig. 13 plots the evolution of the maximum wear depth versus the computed accumulated dissipated energy density. A two-stage evolution is observed. Below a critical wear depth, related to a so-called effective worn thickness ($t_e$), a linear evolution is defined. The wear depth kinetics can be expressed thus as

$$h = x_h \Sigma Ed(0) \quad h < t_e, \quad (24)$$

with $x_h$, the energy wear coefficient defined from the wear depth analysis.

The local energy parameter $x_h = 474 \mu m^3 J$ obtained is close to the volumetric variable $x = 415 \mu m^3 J$. Above the effective worn thickness, spalling phenomena associated to coating decohesions are observed. This quasi-instantaneous failure of the coating is controlled by critical overstressing of the substrate interface.

The results lead us to draw the following important conclusions:

- The very good correlation observed between $x$ and $x_h$ variables confirms (at least for the studied non-adhesive wear tribocouple) that it is possible to develop and formalize a unified global–local wear description (i.e. $x_h = x$).
- It has been shown that coating lifetime is not only related to the progressive energy wear process but is also a function of more severe interface failures. The coating-interface adherence strength must be considered. This suggests that specific precautions must be taken regarding the coating endurances defined from nominal coating thickness ($t_n$) hypotheses. It shows that a
prediction based on the introduced effective worn thickness (i.e. $t_e \leq t_n$) is basically more pertinent and conservative.

Finally, by relating the effective worn coating thickness ($t_e$) to the interface endurance ($N_e$), the energy capacity variable ($\chi$) which rationalizes through a single variable the durability of a given tribocouple can be introduced:

$$\chi \approx \sum_{i=0}^{N_e} Ed(0) = \left[ \left( \frac{Ed(0)}{m^2} \right) \right],$$

(25)

which can be approximated by

$$\chi \approx \frac{t_e}{a}.$$  

(26)

This energy capacity can be defined as the critical energy density which must be dissipated at a given point to induce coating failure. Because it considers a local description, based on the energy density variable, this predictive methodology is less dependent on contact geometry than the conventional wear volume description. It can theoretically be transposed from any contact configuration if energy density is correctly estimated. Future works are, however, required to establish the validity of this assumption, focusing on the contact size effects at global and local scales. Assuming the stability of the introduced global–local approach, this analysis has been extensively applied for different loading conditions involving variable sliding amplitudes and different pressures. The critical contact endurance $N_e$ and the related accumulated energy density variable $\Sigma_{(N_e)} Ed(0)$ are reported in Table 4.

By averaging the obtained values, a reliable estimation of the studied TiC/alumina energy capacity is extrapolated: $\chi = 2.7 \times 10^{-3}$ J/µm² (standard deviation: $0.34 \times 10^{-3}$ J/µm²).

The very small dispersion confirms the stability of the proposed energy density approach.

The principal mechanical parameters like pressure, friction coefficient and sliding amplitude are integrated through a single variable. This single energy density variable, representative of the contact durability, not only integrates the progressive energy wear process through the $\chi$ coefficient but also the substrate/coating cohesion through the effective worn coating thickness ($t_e$). The stronger the substrate-coating strength (i.e. $t_e \rightarrow t_n$), the higher the energy capacity.

4.4. Development of a $Ed-N$ wear endurance master curve description

A major advantage of this energy density approach concept is that it provides, by analogy to any stress analysis regarding cracking phenomenon, a single and synthetic description of wear phenomena as well as the possibility to formalize the accumulation of damages. To illustrate this aspect, an equivalent fatigue $\sigma-N$ representation was introduced.

The $N$ variable, connected in fatigue to a cracking failure, is here related to the wear endurance of the contact, whereas the $\sigma$ variable, associated in fatigue to stress, is related to the representative energy density dissipated per fretting cycle (Fig. 14).

Unlike fatigue problems, the continuous surface evolution imposes a variation of the loading parameter (i.e. energy density). However, by considering the additive property of the energy variable (at the global and local scale), the equivalent average energy density dissipated per

<table>
<thead>
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<th>$\delta^s$ (µm)</th>
<th>100</th>
<th>50</th>
<th>(50/100) x 1</th>
<th>(50/100) x 2</th>
<th>(50/100) x 4</th>
<th>150</th>
<th>200</th>
<th>(150/200) x 1</th>
</tr>
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<tbody>
<tr>
<td>$N_e$</td>
<td>24 500</td>
<td>41 000</td>
<td>31 000</td>
<td>30 000</td>
<td>27 000</td>
<td>20 500</td>
<td>14 500</td>
<td>16 500</td>
</tr>
<tr>
<td>$\Sigma_{(N_e)} Ed(0)$ ($\times 10^3$ J/µm²)</td>
<td>2.6</td>
<td>2.44</td>
<td>2.66</td>
<td>2.52</td>
<td>2.2</td>
<td>3.23</td>
<td>3.03</td>
<td>2.95</td>
</tr>
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Fig. 13. Wear depth evolution versus the maximum accumulated dissipated energy density ($\Sigma Ed(0)$; loading conditions defined in Fig. 12); phase I: smooth energy wear activated process, phase II: instantaneous coating spalling ($\chi$: Energy wear capacity) (TiC/alumina, plane/sphere ($R = 12.7$ mm) contact, $P = 100$ N, $f = 5$ Hz, RH = 50%, $\delta^s$ (± µm): ■ 50 µm, □ 100 µm, ◆ (50/100) µm x 1, ○ (50/100) µm x 2, ● (50/100) µm x 4).
A very good correlation, again confirms the stability of the approach. The energy capacity concept suggests that the contact durability can simply be expressed as a function of the critical number of fretting cycles \( N_c \) and the associated accumulated energy density \( \Sigma \overline{E_d(N_c)}(0) \). Such a simplified methodology does not require any surface treatment or specific surface preparation. The following expression:

\[
N_c = \frac{\chi}{\overline{E_d(N_c)}(0)}.
\]

Fig. 14 plots the evolution of the coating endurance \( N_c \) versus the corresponding average energy density \( \overline{E_d(N_c)}(0) \) for the studied loading conditions. All the experimental results are aligned along a so-called energy wear endurance master curve displaying asymptotic evolutions. As expected, the given master curve can be formalized by the following expression:

\[
\overline{E_d(N_c)}(0) = \left( \frac{1}{E_d(0)} \right)^{1/N_c^{2/3}}.
\]

The energy capacity concept suggests that the contact durability can simply be expressed as a function of the critical number of fretting cycles \( N_c \) and the associated accumulated energy density \( \Sigma \overline{E_d(N_c)}(0) \). Such a simplified methodology does not require any surface treatment or specific surface preparation. The following expression:

\[
N_c = \frac{\chi}{\overline{E_d(N_c)}(0)}.
\]
The given energy wear capacities have, respectively, identified under fretting wear. The following conclusions have been drawn.

Considering such advantages, a reverse identification approach, which consists in plotting the experimental Ed–N master chart and adjusting the value of the energy density capacity in Eq. (28) to fit the experimental curve, can be considered.

This methodology has been applied for two different tribosystems displaying a significant friction variation at the coating failure: a TiN/alumina (R = 12.7 mm) contact and a polymer bonded MoS2 solid lubricant film (Fig. 16). The given energy wear capacities have, respectively, identified: \( \gamma_{TiN/alumina} = 0.4 \times 10^{-3} \text{ J/\mu m}^2 \) and \( \gamma_{MoS2/Ti-Al6-4V} = 2 \times 10^{-3} \text{ J/\mu m}^2 \). Endurance master curves are obtained on which all the experimental results can be superimposed and the different coating endurance compared. The good correlation observed with the theoretical curves infers that this approach could be generalized for many contact configurations. It also supports the reliability of the proposed global–local energy wear description.

5. Conclusion

A global–local approach has been developed to rationalize the fretting wear response of adhesive wear and non-adhesive wear interfaces and to predict interface durability under fretting wear. The following conclusions have been drawn.

- The non-adhesive wear ceramic tribosystems (TiC, TiN, etc.) can be quantified by applying a plain friction energy description.
- Adhesive wear tribocouples (i.e. low weight metals like Titanium and aluminium alloys), imply to consider the debris flow ejection process. A sliding reduced energy wear formulation has been introduced which, by normalizing the dissipated energy with the applied sliding amplitude, permits the wear kinetics of such specific contacts to be rationalized.
- Predicting the surface durability implies a coupled local energy density description to formalize the surface shape and pressure field evolutions.
- A FEM wear simulation, including an iterative evolution of the interface has been implemented. The conclusion is that the interface evolution is controlled by the transition from an elliptical to a homogeneous distribution of the wear depth kinetics in the fretting scar.
- Simplified analytical formulations based on elliptical and flat contact pressure distributions were introduced to approximate the maximum friction energy density dissipated in the interface. It is shown that after very slight wear, the flat approximation provides a very good estimation of the FEM simulations. A simplified formulation was then derived to estimate the accumulated energy density dissipated at the centre of the interface.
- The application of the energy density approach to the ceramic interface TiC/alumina demonstrates that the wear depth kinetics can be predicted as a function of the local accumulated dissipated energy density. It supports the idea that a global–local energy approach is able to formalize the interface durability at least for tribosystems displaying a weak influence of third body.
- It has been shown that the coating (TiC/alumina) lifetime, is not only a function of the energy wear kinetic but also depends on the substrate interface strength. Above a critical “effective worn thickness” an instantaneous coating spalling failure is observed. The weaker the interface, the smaller the effective worn thickness and the shorter the coating lifetime.
- Combining the energy density description and the introduced “effective worn thickness variable”, it has shown that the interface durability can be related to a single energy density capacity variable (\( \gamma \)). This variable, representative of a given tribocouple, is defined as the maximum accumulated energy density which can be dissipated in the interface before coating failure.
- It is finally shown how the energy density capacity variable (\( \gamma \)), independent of the contact geometry, can be applied to compare the palliative durability and to predict contact durability under complex industrial loading conditions.

These important results suggest that for non-adhesive wear tribocouples involving a third body minor effect, a global–local energy wear description is able to capture the contact durability. However, fretting wear, like any tribology process, is a complex problem which cannot be restricted to the introduced energy formulation. The present synthesis outlines the crucial impact of the third body on the wear process particularly for adhesive wear tribocouples. Hence, specific attention must be paid to better consider the dynamical evolution of the visco-elasto-plastic properties of the resulting tribo films, and the dynamics of their evacuation from the interface. This also implies that the current FEM surface modelling should be extended to include the presence of a third body.

References