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Is the wear coefficient dependent upon slip amplitude in fretting? Vingsbo and Söderberg revisited

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ABSTRACT

More than 25 years ago, Vingsbo and Söderberg published a seminal paper regarding the mapping of behaviour in fretting contacts (O. Vingsbo, S. Söderberg, On fretting maps, Wear, 126 (1988) 131–147). In this paper, it was proposed that in the gross-slip fretting regime, the wear coefficient increased by between one and two orders of magnitude as the fretting displacement amplitude increased from around 20 μm to 300 μm (defined as the limits of the gross-slip regime).

Since the publication of this paper, there have been many papers published in the literature regarding fretting in the gross-sliding regime where such a strong dependence of wear coefficient upon fretting displacement has not been observed, with instead, the wear coefficient being shown to be almost independent of fretting amplitude. Indeed, many researchers have demonstrated that there is a good correlation between wear volume and frictional energy dissipated in the contact for many material combinations, with the additional insight that a threshold in energy dissipated in the contact exists, below which no wear is observed (experimental data relating to fretting of a high-strength steel is presented in the current paper which supports this concept).

It is argued that in deriving a wear coefficient in fretting, there are two key considerations which have not always been addressed: (i) the far-field displacement amplitude is not an adequate substitute for the slip amplitude (the former is the sum of the latter together with any elastic deformation in the system between the contact and the point at which the displacement is measured); and (ii) there is a threshold in the fretting duration, below which no wear occurs and above which the rate of increase in wear volume with increasing duration is constant (this constant may be termed the wear coefficient, k_{true}). Not addressing these two issues results in the derivation of a nominal wear coefficient ($k_{nominal}$) which is always less than k_{true}. A simple analysis is presented which indicates that

$$\frac{k_{nominal}}{k_{true}} = 1 - A - B$$

where A is associated with erroneously utilising the far-field displacement amplitude in place of the contact slip amplitude in the calculation of the wear coefficient and B is associated with the failure to recognise that there is a threshold in fretting duration below which no wear occurs.

A and B are shown to depend upon the tractional force required to initiate sliding (itself dependent upon the applied load and coefficient of friction), the system stiffness, the applied displacement amplitude, the threshold fretting duration below which no wear occurs and the number of fretting cycles in the test. Using typical values of these parameters, the ratio of $k_{nominal}$ to k_{true} has been shown to be strongly dependent upon the applied displacement amplitude over the range addressed by Vingsbo and Söderberg (with the ratio rapidly decreasing by an order of magnitude over this range). As such, it is argued that k_{true} shows no strong dependence on slip amplitude in fretting, and that the strong dependence of $k_{nominal}$ upon displacement amplitude presented by Vingsbo and Söderberg does not imply a change in k_{true} as is often inferred.

The routine recording of force–displacement loops in fretting is a major experimental advancement which has taken place since the publication of the paper by Vingsbo and Söderberg. It is argued that this technique must be routinely used to allow the correct interpretation of wear data in terms of the actual slip amplitude (or energy dissipated); moreover, a range of conditions should be experimentally

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examined to allow the threshold fretting duration below which no wear has occurred to be evaluated and its significance assessed.

1. Introduction

1.1. Fretting maps and fretting regimes

Perhaps one of the most significant developments in the study of fretting has been the development of fretting maps, first presented by Vingsbo and Söderberg [1] in 1988. Such maps are based upon an understanding of the contact mechanics as described by Mindlin and Deresiewicz [2], the central concept of which is that when a normally loaded, non-conforming contact experiences a tangential load, there will be an outer region of the contact which exhibits slip and a central zone which is stuck (together, these are termed a partial slip condition). The stuck zone will decrease in size until the tangential force is equal to the product of the coefficient of friction and the normal load, whereupon slip will occur over the whole contact; this condition is known as gross sliding.

Vingsbo and Söderberg [1] introduced the concept of fretting maps which was based upon their own experimental observations along with a detailed review of the literature. By comparing force–displacement plots (fretting loops) recorded during fretting experiments (using a ball-on-flat geometry) and subsequent metallographic examination of the wear scars, they were able to identify three distinct fretting regimes characterised by the stick–slip behaviour of the contact:

1. Stick: Characteristic of very low displacement amplitudes. The wear scar shows no visible damage beyond limited plastic shearing of individual asperities with no indication of material damage in between.
2. Stick–slip: At higher displacement amplitudes, a central stick area with a surrounding slip-annulus is seen as would be expected from the Mindlin model. There is evidence of plastic shearing of the asperities in the central stick region as in the stick case, but in the slip annulus, there is considerable damage of the surface.
3. Gross sliding: The entire wear scar shows extensive plastic shear with visible sliding marks.

The fretting regimes described in terms of the Mindlin model and as presented by Vingsbo and Söderberg [1] have often been assumed to be stable i.e., not a function of slid distance. However, Zhou et al. [3] described a slip regime, which they termed the mixed fretting regime (MFR), where the fretting loop is unstable with periods where an elliptical fretting loop is observed and periods where an open fretting loop is observed. In a subsequent paper, Zhou and Vincent [4] described the regimes (as a function of increasing displacement amplitude) which occur at smaller displacement amplitudes than those in gross-sliding as follows:

1. Sticking regime: Associated with a nominally closed fretting loop where the contact is behaving as described by the Mindlin contact mechanics i.e., some central region of the contact is stuck while micro-slip occurs in the outer zones. The chosen terminology of "sticking" is somewhat confusing since this is normally referred to as "partial slip".
2. Mixed fretting regime (MFR): Characterised by an unstable fretting loop with periods of closed, elliptically-shaped loops and periods of fully open loops.

They ignored the stick regime suggested by Vingsbo and Söderberg [1] suggesting (in agreement with Mindlin) that there will always be some outer region of the contact experiencing slip, and therefore that no stick regime exists in reality. It was suggested that the unstable behaviour observed in the MFR is due to the evolution of the wearing surfaces and generation of loose debris within the contact. Initially, the contact may be in gross-sliding due to the lubricating effect of the surface oxide and contaminant films; however, these will rapidly disperse and as a result, the tangential force will increase as regions of metallic (adhesive) contact form. Due to accumulation of damage, wear debris will be released, a layer of which can provide a low shear interface, relieving the tangential force and allowing a period of gross sliding. This proposed mechanism was supported by observations made during the testing of a 9005 Al–Li alloy where it was found that the MFR could not be established due to the rapid generation of debris [4]. Later work by Hager et al. [5] on Ti6Al4V produced further evidence which supported the overall mechanism, suggesting that either the adhesive junctions which are formed will fatigue and rupture forming debris particles, or that a layer of material with a tribologically transformed structure (TTS) will form and breakdown to generate the debris.

The potential usefulness of fretting maps which characterise regimes of behaviour is clear: they allow tests by different workers to be quantitatively compared where the fretting regime has been identified to be the same [6]. In fact, as has been shown by correlation between the mode of surface degradation and the fretting regime [3], it is impossible to make any meaningful comparison without knowing the location of the test with respect to the regime boundaries. Ultimately, fretting maps can be a practical aid to designers and have been expanded to cover other dimensions (rather than just those of load and stroke which are most commonly employed) [6,7]. Given their importance, the measurement of fretting loops (which underpin many fretting maps) is clearly an essential requirement in fretting research.

1.2. Dependence of wear coefficient upon the fretting amplitude– a critique of Vingsbo and Söderberg's paper

In their paper, Vingsbo and Söderberg [1] presented a figure (a revised version of which is reproduced here in Fig. 1) which has been widely quoted since. Based upon a review of the literature, it shows that the wear coefficient is very small (and increases only slowly with displacement amplitude) over the stick–slip regime; it then rises

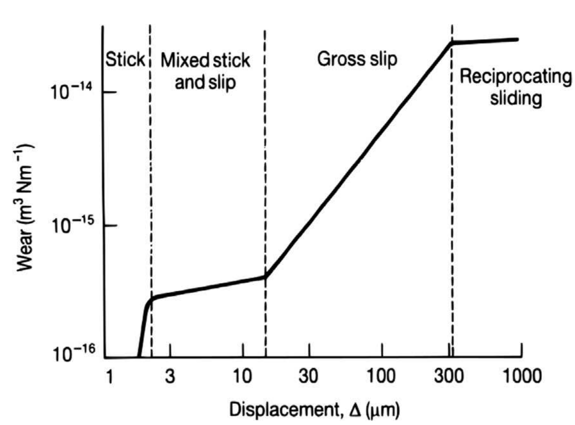

Fig. 1. Illustration of the dependence of wear coefficient with displacement amplitude in fretting with reference to the slip regime: from [14] after [1].

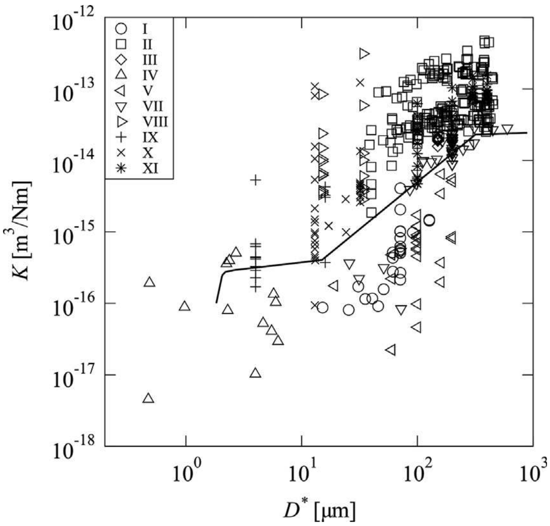

Fig. 2. Presentation of the original data used in the construction of Fig. 1 which was collected and summarised by Knudsen and Massih in their analysis of the effect of displacement amplitude on the wear coefficient in fretting [8]. The legend refers to the sources of the data, the details of which can be found in their paper.

rapidly with increasing displacement amplitude (with an increase of almost two orders of magnitude on increasing the displacement amplitude from about 15 μm to 300 μm) over the gross slip regime. In generating this figure, Vingsbo and Söderberg [1] recognised the difficulties in bringing data together where tests had been conducted under a wide range of conditions, stating: "Therefore literature data can be incomplete and difficult to interpret, and often only orders of magnitude are relevant." Despite this caveat, a review was subsequently published of the literature which had been used in the construction of Fig. 1 [8]. In this paper, Knudsen and Massih reproduced Fig. 1, but superimposed upon it the data which Vingsbo and Söderberg had used in its construction; their figure is reproduced in Fig. 2. The spread of the data is seen to be very large (note that the wear coefficient is plotted on a logarithmic axis).

Such variations in wear coefficient in a system are rarely seen elsewhere in tribological research, except where there is a transition in the underlying mechanism of material removal. However, such a transition in mechanism has not been generally reported with increasing displacement amplitude within the gross-slip regime; moreover, the experimental results presented in many recent publications where the effect of displacement amplitude is addressed are not in accord with Fig. 1 (e.g. [9,10]). It must be recognised, however, that experimental techniques have advanced considerably since the publication of the paper by Vingsbo and Söderberg [1] (and the publication of the papers upon which their proposal was based). We argue in this paper that there were two primary limitations in early work, both of which relate to some degree to the general lack of availability of systems to record fretting loops in earlier research:

1. In general, the fretting displacement amplitude was measured, but this is not the same as the amplitude of the slip in the contact itself. The recording of fretting loops (which is now a commonplace in fretting research) facilitates the derivation of the slip amplitude from the applied displacement amplitude (see Section 1.3), but this was not generally available in earlier research. The applied displacement amplitude will be higher than the slip amplitude (due to elastic deformations in the system); in the limit, the applied displacement may be taken up entirely by elastic deformation, with no slip at the contact actually occurring. This problem was specifically noted by Bryggman and Söderberg [11] (in a paper which preceded that of Vingsbo and Söderberg [1]), where they stated: "… the bulk [measured] displacement may be considerably larger than the actual slip amplitude at the interface. The value of the interfacial slip amplitude is difficult to measure experimentally, …". It should be noted that the inappropriate use of the applied displacement amplitude in place of the slip amplitude has been reported in more recent work alongside fretting loops (which could have been used to derive the slip amplitude from the measured displacement amplitudes); this indicates that there is a lack of clarity relating to the differences between the slip and displacement amplitudes which exists alongside the technical difficulties associated with measurements of the fretting loops themselves [12].
2. Whilst the wear volume may be correctly recorded, the calculation of the wear coefficient may be affected by errors in measurement and invalid assumptions. Firstly, the wear volume should be calculated using the total distance slid by the contact (rather than any measurement based upon the far-field displacement amplitude); as noted previously, the former was rarely available, so that the latter was used as a substitute. Secondly, the derivation of the wear coefficient often assumes that the wear volume is directly proportional to the duration of fretting (be that measured by accumulated slip distance or byenergy dissipated in the contact). Recent research has cast doubt upon this assumption, indicating that there may be a threshold fretting duration below which no wear occurs (although damage is being accumulated) [10,13].

These two concepts are central to this paper and will be explored in more detail in the following sections.

1.3. Description of a fretting loop

Before progressing further, it is necessary to define the terminology employed in descriptions of fretting, and in particular, the terminology used to describe fretting loops (Fig. 3). Throughout the literature, a number of terms are used to describe the displacement imposed between fretting specimens; in this work, the applied reciprocating

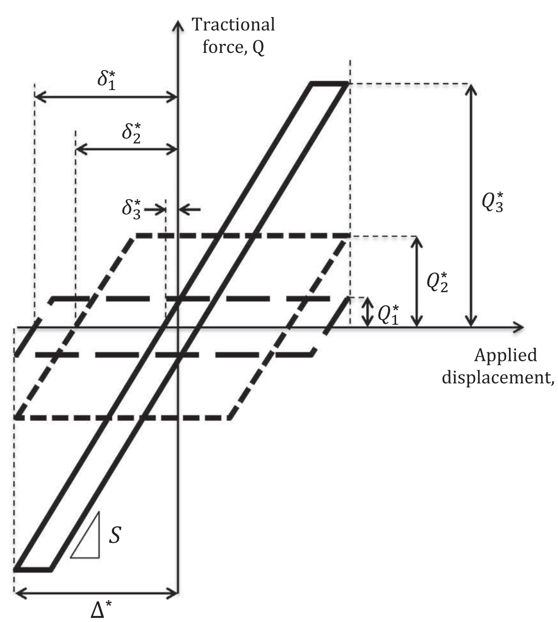

Fig. 3. Schematic diagram of three idealised fretting loops with the same applied displacement amplitude (Δ^*) and system stiffness (S) but different values of δ^* associated with different values of Q^*. The different values of Q^* can be associated with changes in normal load, P, or coefficient of friction, μ. The energy dissipated per cycle, E_d, is represented by the area enclosed by each parallelogram.

displacement at any point in the cycle is given the symbol Δ, with Δ^* referring to its amplitude (referred to as the applied displacement amplitude). It is important to recognize that Δ is measured at some point remote from the contact and (in addition to any slip at the contact itself) includes all elastic deformation in the system between the contact point and the position at which the displacement is measured i.e., the combined contact, bulk specimen, fixture and rig elastic displacements. Together, these can be described via a system stiffness, S; the role of S in determining the shape of the fretting loops is shown in Fig. 3. Some examples of experimental apparatus system stiffnesses for fretting testing reported in the literature are as follows (in order of increasing stiffness): 1.4 MN m^{-1} for a system with a 5 mm radius steel ball on both a PVD coated and uncoated steel flat [10]; 1.75 MN m^{-1} for a system with a 5 mm radius alumina ball on alumina flat [15]; 17.6 MN m^{-1} for a system with a 6 mm radius PVD coated titanium alloy cylinder on PVD coated titanium alloy flat contact [16]; 20–27 MN m^{-1} for a system with a 6 mm radius titanium alloy cylinder on titanium alloy flat contact [17–19]; 59 MN m^{-1} for a system with a 12.7 mm radius steel ball on steel flat contact (with both bodies both PVD coated and uncoated) [20]; and 57–66 MN m^{-1} for a system with a 6 mm radius steel cylinder on steel flat contact [21]. As can be seen, within these examples, the stiffness, S, varies over a factor of around 30.

The actual contact slip amplitude between the specimens, δ^*, is not directly measured, and is commonly determined by post-processing of the force and displacement data (i.e., the fretting loops). Depending on the loading conditions and design of the test apparatus, the slid distance per cycle ($4\delta^*$) may be much less than the distance moved by a remote measuring point ($4\Delta^*$). In addition, for a given applied displacement, the resultant slid distance will decrease as the tractional force for slip (Q^*) increases (be that through the application of an increased normal load or through an increase in the coefficient of friction) due to a greater proportion of Δ^* being accommodated by the system compliance (as illustrated schematically in Fig. 3 where δ^* is seen to decrease with increasing Q^* for a constant Δ^*). The area enclosed within the fretting loop is the energy dissipated in the contact per cycle (E_d) due to gross sliding of the contact.

1.4. *Derivation of wear coefficients in fretting: errors associated with assumptions related to the sliding distance or energy dissipated*

In fretting, an Archard-based wear coefficient ($k_{Archard}$) is typically quoted in terms of the volume of material lost per unit normal load per unit distance slid (with units of $m^3\, m^{-1}\, N^{-1}$) as follows:

$$k_{Archard} = \frac{V^W}{4N\delta^* P} \tag{1}$$

where V^W is the measured wear volume, N is the number of fretting cycles, and P is the applied normal load on the contact. However, more recently, a number of researchers have used a different form of wear coefficient (termed the energy-based wear coefficient, k_{energy}), which is defined as the volume of material lost per unit of energy dissipated in sliding (units of $m^3\, J^{-1}$), with the total energy dissipated over the test (E_d^{tot}) being the sum of the values of E_d for the individual fretting cycles which together make up the test,

$$k_{energy} = \frac{V^W}{E_d^{tot}}. \tag{2}$$

If the two coefficients are compared, it can be seen that they are dimensionally the same, with the Archard wear coefficient incorporating the coefficient of friction which is directly integrated within E_d in the energy method.

If a fretting loop is not measured as part of the experimental procedure, then the only measurements available to the researchers are typically those of Δ^* (the applied displacement amplitude) and either the normal load, P or the maximum tractional force, $Q*$ (it is often the former of these two which is readily available and is therefore more often used in the calculation of $k_{Archard}$).

As can be seen, there is a discrepancy between the data required and that often available in relation to the slip distance; as such, an approximation is commonly made that $\Delta^* \approx \delta^*$ which then allows estimates of the wear coefficient to be made. However, the validity of the approximation depends upon a number of parameters associated with the fretting test, such as the rig stiffness (S), the tractional load for sliding (Q^*) (which itself depends upon the normal load P and the coefficient of friction, μ) and the applied displacement amplitude itself (Δ^*); the effect of these can be understood by reference to Fig. 3. The validity of the approximation improves as S and Δ^* increase and as Q^* decreases.

Ohmae and Tsukizoe [22]—whose work was one of the primary sources used by Vingsbo and Söderberg [1] to estimate the transition to sliding wear—considered the fretting of a flat-on-flat (conforming) steel contact. A plot from their work, showing the wear volume as a function of slip amplitude is shown in Fig. 4—this is essentially a plot of wear coefficient as a function of slip amplitude as the tests were all conducted under the same applied load and with the same total overall displacement being applied. It must be noted that the value of 70 μm as the transition displacement amplitude for measurable wear is high compared to that reported in more recent work. In this case, the slip amplitude quoted on the figure is actually the applied displacement amplitude (i.e. Δ^*), measured some way distant from the contact. Thus, it is not clear whether the zero values of wear volume are associated with the slip amplitude having fallen to zero (with the applied displacements all being taken up elastically within the system) or with the threshold fretting duration (below which no wear occurs) not being reached.

Another source for the work of Vingsbo and Söderberg [1] was the work of Bill [23] who—using a steel ball-on-flat contact—reported that the fretting wear in tests with applied displacement amplitudes below 25 μm was characterised by surface damage but did not increase with sliding distance. Again, it can be surmised that below an applied displacement amplitude of 25 μm, the contact was in the stick–slip regime (which will be associated with a much smaller slip amplitude, δ^*, which was not itself evaluated). Work by Toth [24] also contributed to the findings of Vingsbo and Söderberg [1]; in this work, applied displacement amplitudes from 50 to

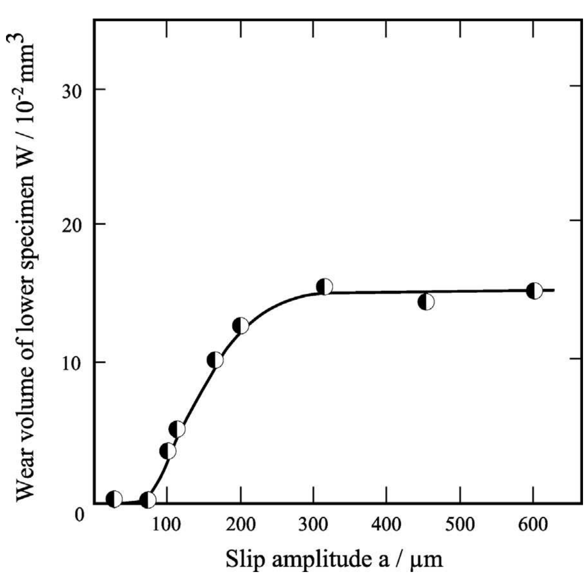

Fig. 4. Wear volume as a function of slip amplitude after fretting for a total applied displacement distance of 36 m; after [22].

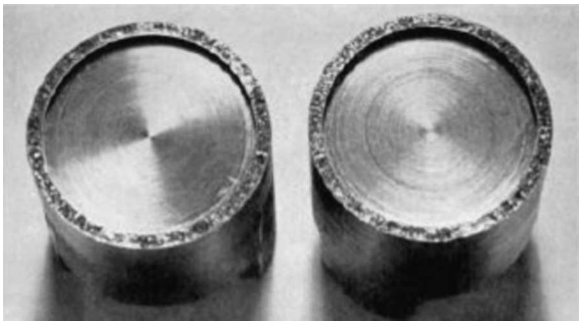

Fig. 5. A torsional fretting specimen with the centre relieved to create a contact annulus as utilised by Feng and Rightmire [26].

500 μm were considered over a range of normal loads, with the findings clearly indicating a linear relationship between wear volume and displacement amplitude. Extrapolation of the trends in the data indicates that no wear would have been observed below an applied slip displacement of ~25 μm.

In all these cases, the wear coefficient (above some applied displacement threshold) was reported to be proportional to the displacement amplitude over a significant range (up to 500 μm). There is one exception to this of which the current authors are aware, that being an extensive early study of fretting wear by Feng and Uhlig [25]. In this work, a novel fretting rig was developed which used torsional fretting between two cylinders with the centre relieved to create a narrow contact annulus (Fig. 5); this arrangement allowed them to optically observe the actual slip displacement at the contact (as opposed to the applied displacement at a distant point). It is then perhaps not surprising that they found the wear coefficient to be constant over a range of slip amplitudes, spanning the range of approximately 10–230 μm.

1.5. Derivation of wear coefficients in fretting: errors associated with assumptions related to the wear volume being proportional to the sliding distance or energy dissipated

Fouvry and co-workers have pioneered the use of the energy-based wear coefficient (see Eq. (2)). As this work developed, they examined the wear volume as a function of dissipated energy, and found that in some cases, there was a threshold energy (E_{th}), below which no wear was observed (this is a different type of threshold than that associated with the assumption that $\Delta^* \approx \delta^*$ where δ^* can become zero even for non-zero values of Δ^* as outlined previously). For a steel–alumina couple, they identified the threshold to be 13 J, whereas for a TiN–alumina couple it was found to be near zero (2.3 J) (both utilising a point contact) [27]. Similarly, Ramalho et al. [10] performed fretting tests with a 5 mm radius steel ball against coated and uncoated flat steel specimens, both in air and in vacuum; in all cases, they observed a threshold energy between 1.0 J and 3.75 J. It has been proposed that the threshold energy is related to the minimum energy density required to cause recrystallisation of the microstructure (related to the development of the TTS) which is required before the formation of wear debris occurs [13].

A more recent study [9] involved the fretting of a 40 mm diameter Ti6Al4V cylinder on a flat, with the research seeking to investigate the mixed fretting regime i.e., the transition from partial slip to gross sliding. In this work, the displacement amplitude was varied with a constant normal load; the resulting data indicated that there was a critical sliding amplitude ($\Delta^* \sim 27$ μm, $\delta^* \sim 10$ μm) below which there was no wear, and above which the wear volume was found to increase linearly with the sliding amplitude. Since the normal load was held constant, the wear volume as a function of dissipated energy showed the same trends and a threshold energy of 4 kJ was reported. This is clearly a significantly higher value than the tens of Joules reported for point contacts. If it is assumed that the energy threshold is related to the formation of the TTS, then it would be expected that the value would be higher for a contact with a larger area i.e., the threshold should be better described by an energy density. However, the evidence does not exclude the possibility that there is also a minimum displacement amplitude below which, while debris may be formed, there is insufficient relative motion to result in its ejection from the contact and hence the establishment of steady wear; indeed, this latter explanation was the conclusion of Heredia and Fouvry [9]. Moreover, Fouvry et al. [28] further suggest that for “adhesive wear contacts involving aluminium and titanium alloys” there may be an amplitude dependence of wear coefficient associated with debris removal from the contact.

(The following argument applies equally to the threshold fretting duration being measured in terms of total contact sliding distance, but (for clarity) will be presented only in terms of k_{energy} in light of the experimental work and analysis presented in subsequent sections).

For a number of material pairs, above the critical threshold energy for material removal (E_{th}), the wear volume has been found to be proportional to the dissipated energy (E_d^{tot}); consequently, it is appropriate to express the wear volume, V^w, as

$$V^w = \begin{cases} k_{energy}\,(E_d^{tot} - E_{th}), & \text{for } E_d^{tot} \geq E_{th}, \\ 0, & \text{otherwise.} \end{cases} \quad (3)$$

Significant data available in the literature indicate that there is an energy–displacement threshold above which the wear volume is proportional to the dissipated energy. The presence of a threshold becomes important when deriving a wear coefficient from any single test; the general practice is to take a wear volume and divide it by the total energy dissipated (or the product of the load and slid distance if the former is not available) to derive a wear coefficient. If the wear volume as a function of loading parameters takes the form

$$V^w = k_{energy}(E_d^{tot}), \quad (4)$$

then an accurate estimate of the wear coefficient, k_{energy}, can be found from a single measurement of V^w and E_d^{tot}; however, if V^w takes the form described in Eq. (3), then it is impossible to derive k_{energy} from a single measurement of V^w and E_d^{tot}. If the average wear coefficient ($k_{average}$) is derived from such a simple ratio of

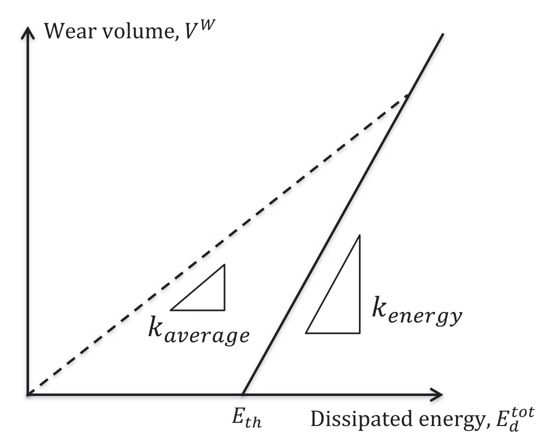

Fig. 6. Schematic illustration of the dependence of $k_{average}$ on other parameters for a wear coefficient with a linear dependence upon dissipated energy above a certain threshold value of energy, E_{th}.

V^w and E_d^{tot}, then it will exhibit a dependence upon k_{energy}, V^w and E_{th} as follows:

$$k_{average} = \begin{cases} \frac{k_{energy} V^w}{V^w + k_{energy} E_{th}}, & \text{for } E_d^{tot} \geq E_{th}, \\ 0, & \text{otherwise.} \end{cases} \quad (5)$$

This is illustrated schematically in Fig. 6. There is evidence in the literature that such an erroneous interpretation is not uncommon since the presence of an energy threshold E_{th} has not been widely recognised.

1.6. Summary and objectives

Vingsbo and Söderberg [1] have argued that the wear coefficient is a very strong function of displacement amplitude in fretting; specifically, they proposed that the wear coefficient continues to rise with displacement amplitude until the reciprocating sliding regime is reached (they estimated this to be at 300 μm), above which the wear coefficient becomes independent of displacement amplitude. This work is apparently at odds with much recent work on fretting, where the wear coefficient is observed to exhibit only a very limited dependence upon slip amplitude (noting that there is a subtle change in terminology from displacement amplitude to slip amplitude). As such, it is proposed that the conclusions of Vingsbo and Söderberg [1] are based upon errors incorporated within the interpretation of the data. The first error is associated with an invalid assumption that the applied displacement amplitude is the same as the slip amplitude and that the former can be used in the calculation of a wear coefficient; the second error is associated with the failure to recognise that there is a threshold fretting duration (best described in terms of a dissipated energy) below which wear does not occur (although the TTS is being formed). This paper will seek to show that (even with a constant true wear coefficient), the nominal wear coefficient (that calculated with these two errors of interpretation) will show a dependence upon fretting displacement; moreover, it is shown that the fretting test variables and the design of the specimens and test apparatus will affect the magnitude of the discrepancies, which perhaps itself indicates why the spread in data presented in Fig. 2 is so large. The method used involves an experimental programme, and an analysis based upon this programme of experimental work.

2. Experimental method

2.1. Materials and specimens

All specimens were manufactured from a high strength alloy steel – 3% Cr Mo V – typically employed in aero-engine transmission components [29]; the composition of the steel is presented in Table 1. The test material was first cut into blanks with a machining allowance on all dimensions. The blanks were heated to 940 °C and held for 45 min, after which they were oil quenched. Subsequently, they were tempered at 570 °C for 120 min and finally air cooled. After grinding to finished dimensions (Fig. 7), the Vickers hardness (HV20) of the surface was measured and found to be in the range 4.56–4.68 GPa, confirming that any decarburised layer had been completely removed.

2.2. Fretting tests

Fretting tests were conducted using a cylinder-on-flat arrangement (Fig. 7), generating a 10 mm line contact. The flat specimen was attached to a static lower specimen mounting block (LSMB) and the cylindrical specimen to the moving upper specimen mounting block (USMB). An oscillatory displacement (of amplitude Δ^*) was applied to the USMB at a fixed frequency of 20 Hz by an electromagnetic shaker. The relative displacement between the USMB and LSMB was measured by a linear variable differential transformer (LVDT) supplied by RDP Electronics, Wolverhampton, UK (model GT500Z with a range of ±0.5 mm). A constant normal load, P, was applied to the USMB via a dead weight and lever arm. The tangential traction forces were measured using a piezoelectric load cell between the electromagnetic shaker and the USMB. The sensing elements of the load cell are three piezoelectric load washers (Kistler type 9132BA sensors with Kistler type 5073A charge amplifiers) equispaced on a 28 mm pitch circle diameter. The charge from each sensor is summed before amplification; consequently, it can be shown that the load cell is insensitive to bending moments. Each load washer has a measuring range of 0 – 7 kN with the load washers being each preloaded in compression to 3.5 kN by individual bolts. The load washers have a quoted threshold of <0.01 N and a stiffness of 1.8 kN μm^{-1}.

Before testing, the specimens were thoroughly degreased using detergent and industrial methylated spirit. The control and data acquisition system, written in Lab-VIEW™, continuously recorded the tangential force and relative displacement at 4 kHz sampling rate (200 sampling points per fretting cycle). Post-processing of the data enabled important quantitative fretting parameters to be derived such as the contact slip amplitude, δ^*, the tangential force amplitude, Q^* and the dissipated energy, E_d (see Fig. 3). Since the data were recorded continuously, it was possible to derive these parameters for every cycle throughout the test and also to derive the total energy dissipated, E_d^{tot}. Fretting tests were conducted at combinations of four different displacement amplitudes, ranging from 10 μm to 100 μm, and three normal loads, ranging from

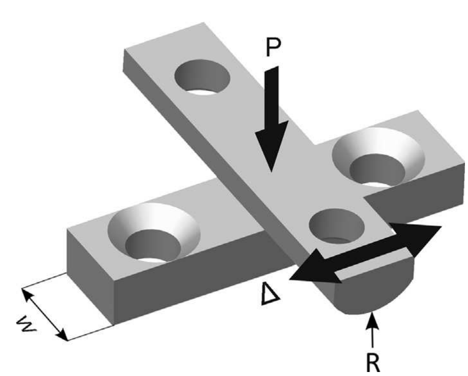

Fig. 7. Diagram of the specimens and their arrangement in the fretting test: W=10 mm, R=6 mm, P=normal load, and Δ=applied displacement.

Table 1
Composition of the test steel (wt%) [30].

C	Si	Mn	P	S	Cr	Mo	Ni	V	Fe
0.35–0.43	0.1–0.35	0.4–0.7	<0.007	<0.002	3.0–3.5	0.8–1.1	<0.3	0.15–0.25	Remainder

Table 2
Fretting test conditions.

Parameter	Values employed
Normal load, P (N)	250, 450 and 650
Applied displacement, Δ^* (μm)	10, 25, 50 and 100
Fretting cycles, N	100,000
Fretting frequency, f (Hz)	20

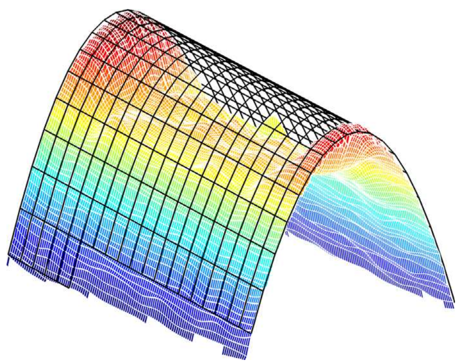

Fig. 8. Schematic diagram of a worn surface and the fitted reference surface (mesh).

250 N to 650 N; in all cases a fixed duration of 100×10^3 cycles was used. All experiments were conducted under normal laboratory conditions. The fretting conditions used in this study are summarised in Table 2.

2.3. Characterisation of wear

A tactile profilometer (Taylor-Hobson Talysurf CLI 1000) with a fine diamond stylus (90°, 2 μm radius) was used to measure the surface profile of the worn specimens. Prior to scanning, the specimens were rinsed with industrial methylated spirit to remove any debris not adhered to the surface. As the wear scars extend over the full width of the flat specimens, profiles were taken over only the central 8 mm of the scars with 0.25 mm spacing between adjacent traces. For the cylindrical specimen, similarly spaced profiles were taken over an area completely spanning the wear scar. In order to estimate the wear volume, a reference (unworn) surface must be defined. In the case of the flat specimen, the reference surface was defined as the best fit plane to all points outside of the wear scar. However, definition of the reference surface is more difficult for the cylindrical specimen. When conducting the profilometry, it was ensured that the first and last profiles were always entirely outside of the worn area. A polynomial fit was then taken for these two profiles and an estimate for the unworn surface was generated by interpolating between these two fitted profiles—illustrated by the mesh in Fig. 8. Any material build-up above the reference plane (for either the flat or cylindrical specimen) is considered to be transferred material or debris, and is defined by a positive volume V^+; any loss of material from below the reference plane is defined as a negative volume V^- (see Fig. 9). The overall wear volume (V^W) is then defined as follows:

$$V^+ = V^+_{cylinder} + V^+_{flat}$$

$$V^- = V^-_{cylinder} + V^-_{flat}$$

$$V^W = -(V^+ + V^-) \tag{6}$$

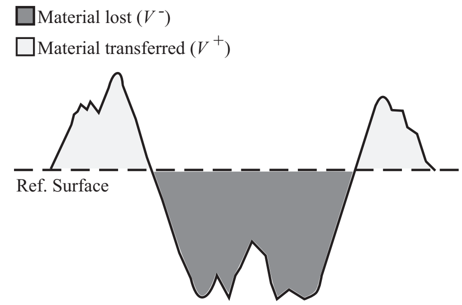

Fig. 9. Schematic diagram indicating the definitions of transfer (V^+) and wear (V^-) volumes.

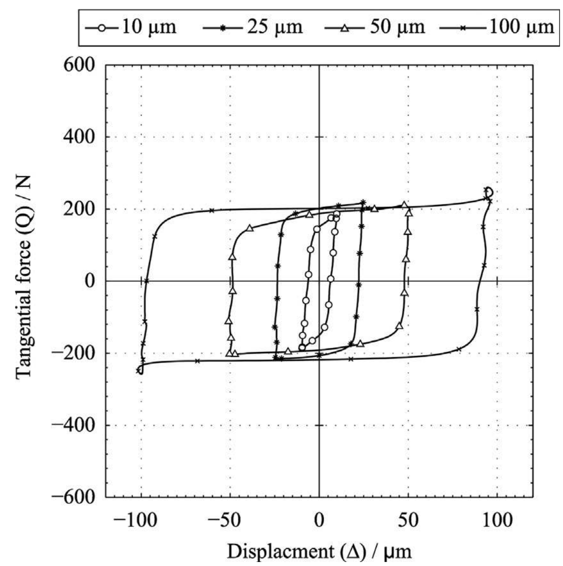

Fig. 10. Experimental fretting loops (50,000th cycle) for applied displacements (Δ^*) of 10 μm, 25 μm, 50 μm and 100 μm under a normal load of 250 N. The markers are for identification only and do not represent the 200 measurement points per cycle.

3. Results

3.1. Fretting loops, coefficient of friction and rig stiffness

Fig. 10 shows examples of fretting loops from tests with different values of applied displacement amplitude (Δ^*) with a normal load (P) of 250 N. Discounting the test where fully developed sliding of the contact had not been established ($\Delta^* = 10$ μm), it can be seen that the characteristic fretting loops share a number of common features. The general shape of the loops is that of a parallelogram: the steep sides correspond to the period of the cycle when the contact is not sliding (stuck), with the approximately horizontal portions (top and bottom of the loops) corresponding to the periods of the cycle when the contact is sliding. The gradient of the stuck portions of the cycle correspond to the system stiffness (a least squares fit shows the rig stiffness, k, to be 57.3 MN m^{-1} for the setup utilised in this work). The loop top and bottom, corresponding to the sliding period of the cycle, may be expected to be horizontal with a constant value of $Q^* = \mu P$. However, examination of Fig. 10 indicates that the tangential force in general increases throughout the sliding periods of the cycle, reaching a maximum at the end of the stroke: this is typically attributed to plasticity and the geometry of the wear scar [31–33].

The energy dissipated in fretting can be experimentally determined from the area enclosed within a fretting loop. Fig. 11 plots the total wear volume as a function of the measured dissipated energy during a test, E_d and incorporates data from all

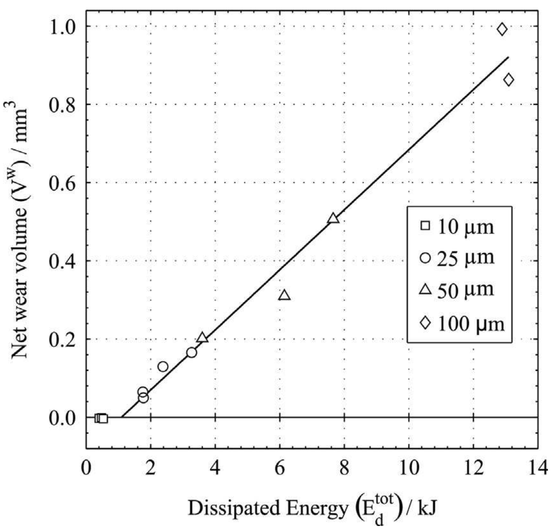

Fig. 11. Wear volume as a function of dissipated energy for SCMV specimen pairs; for combinations of P=250, 450 and 650 N and Δ^*=10, 25, 50 and 100 μm.

combinations of load and applied displacement amplitudes examined. It can be seen that, as may be expected, there is a linear relationship between the wear volume and dissipated energy [27]. Using simple linear regression (where R^2=0.98 is found), the wear volume (V^w in mm^3) as a function of the total dissipated energy (E_d^{tot} in kJ) can be shown to be

$$V^w = 7.69 \times 10^{-2} E_d^{tot} - 8.47 \times 10^{-2}. \tag{7}$$

From this relationship, E_{th} can be shown to be 1100 J; in metals, this threshold energy for wear to commence has been related to the minimum energy density required to form the TTS and hence initiate wear [13,34].

3.2. Modelling of apparent wear coefficients

For the purposes of estimating the effects of: (i) not considering the difference between δ^* and Δ^* and (ii) not considering the effect of E_{th} during analysis to derive the wear coefficient, a simple model is required. If the loop is assumed to be a true parallelogram in shape, the energy dissipated per loop can be approximated by

$$E_d = 4Q^*\delta^*. \tag{8}$$

With knowledge of the system stiffness, S, derived from the fretting loop, the slip displacement, δ^*, can be related to the measured displacement, Δ^*, by

$$\delta^* = \Delta^* - \frac{Q^*}{S}. \tag{9}$$

Therefore, the total energy dissipated throughout the test (assuming that the parallelogram-shaped fretting loops do not change in shape over the test duration of N cycles) can be defined as

$$E_d^{tot} = \begin{cases} 4Q^*\left(\Delta^* - \frac{Q^*}{S}\right) N, & \text{for } \Delta^* \geq \frac{Q^*}{S}, \\ 0, & \text{otherwise.} \end{cases} \tag{10}$$

Substituting Eq. (10) into Eq. (3) gives

$$V^W = k_{energy}\left\{4Q^*\left(\Delta^* - \frac{Q^*}{S}\right) - E_{th}\right\}. \tag{11}$$

However, in the literature, the common approach is to derive a relationship of the form

$$k_{nominal} = \frac{V^W}{4\Delta^* Q^* N} \tag{12}$$

i.e., ignoring the difference between Δ^* and δ^* and any wear threshold. Combining Eqs. (11) and (12) (shown graphically in Fig. 6) allows the nominal wear coefficient to be related to the true wear coefficient

$$k_{nominal} = k_{energy}\left(1 - \frac{Q^*}{S\,\Delta^*} - \frac{E_{th}}{4\Delta^* Q^* N}\right). \tag{13}$$

As can be seen, $k_{nominal}$ is a function of Q^*, N, E_{th} and S. By inspection, it is clear that $k_{nominal}$ is always less than k_{energy}, and increases monotonically as Δ^* increases. $k_{nominal}$ also increases monotonically as N increases, as E_{th} decreases and as S increases. However, its dependence upon Q^* is more complex and depends upon the values of the other parameters in the second and third terms in the parenthesis in Eq. (13).

Vingsbo and Söderberg [1] suggested that the wear coefficient increases with displacement amplitude in fretting up to the point when reciprocating sliding commences (this transition was defined as the point at which the nominal wear coefficient became approximately constant, although it is seen through the current analysis to be entirely arbitrary); this transition displacement amplitude was deemed to be around 300 μm. Accordingly, $k_{nominal}$ has been evaluated in this work using the experimentally derived value of k_{energy} of 7.69×10^{-14} m^3 m^{-1} N^{-1} over a range of Δ^* from 1 μm to 350 μm (to cover the range up to that defined as reciprocating sliding by Vingsbo and Söderberg [1]). Values of Q^*, N, E_{th} and S utilised are listed in Table 3 and were typical of those observed in fretting testing either in this work or in the literature; their selection was designed to illustrate the scale of the variation in $k_{nominal}$ that might be expected as a function of Δ^*.

Fig. 12 illustrates the dependence of $k_{nominal}$ on the applied displacement amplitude (Δ^*) for the conditions indicated (with both axes plotted on logarithmic scales to match Fig. 1 from Vingsbo and Söderberg's paper [1]). It can be seen that in all cases, $k_{nominal}$ tends towards the value of k_{energy} at values of Δ^* which tend towards the reciprocating sliding regime (> 300 μm) but decreases sharply from these values as Δ^* decreases. Fig. 12a shows the variation in $k_{nominal}$ with Δ^* for three selected values of Q^*. As Q^* is increased from its lowest value of 100 N to 500 N, $k_{nominal}$ is observed to increase; however, as Q^* is further increased to 2500 N, $k_{nominal}$ is observed to decrease. To further illustrate this point, Fig. 13 shows the variation in $k_{nominal}$ with Q^* for three selected values of Δ^* (typical of those used in the fretting literature). It can be seen that in each case, there is a maximum in $k_{nominal}$ at an intermediate value of Q^*, with lower values being observed at both higher and lower values of Q^*. In all cases, the reduction in $k_{nominal}$ as Q^* decreases from its value where $k_{nominal}$ was a maximum is very rapid. In contrast, the reduction in $k_{nominal}$ as Q^* increases from its value where $k_{nominal}$ was a maximum is less rapid, and the rate falls as Δ^* increases.

Fig. 12(b)–(d) illustrates the magnitude of the trends in $k_{nominal}$ with N, E_{th} and S which were identified previously. The influence of each of these three parameters on the change in $k_{nominal}$ is monotonic as indicated in Eq. (13).

All the plots in Figs. 12 and 13 show substantial reductions in $k_{nominal}$ (around an order of magnitude) on reducing Δ^* from the maximum value where it was evaluated of 350 μm down to

Table 3
Parameters used for calculations of basic wear coefficient; when the effect of variations in one parameter was being examined, the values indicated in bold for the other parameters were utilised.

Variable parameter	Values utilised
Tractional force required for sliding, Q^* (N)	100, **500**, 2500
Number of cycles, N	20,000, **100,000**, 500,000
Threshold energy, E_{th} (J)	1.1, **1100**
System stiffness, S(MN m^{-1})	11.5, **57.3**, 286.5

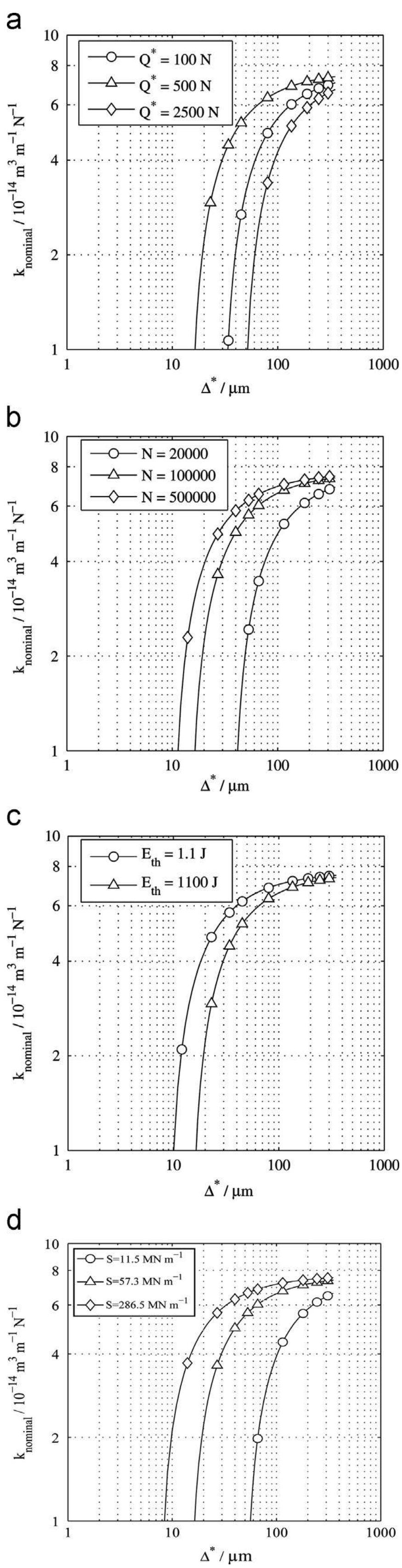

Fig. 12. Graphs of variation in $k_{nominal}$ as a function of Δ^* predicted using parameter sets with one parameter being varied in each case as follows: (a) tractional force required for sliding; (b) number of fretting cycles; (c) threshold energy for onset of wear; (d) rig stiffness. Where not a variable, the tractional force required for sliding = 500 N, the system stiffness = 57.3 MN m^{-1}, the threshold energy for onset of wear = 1100 J and the number of fretting cycles = 100,000.

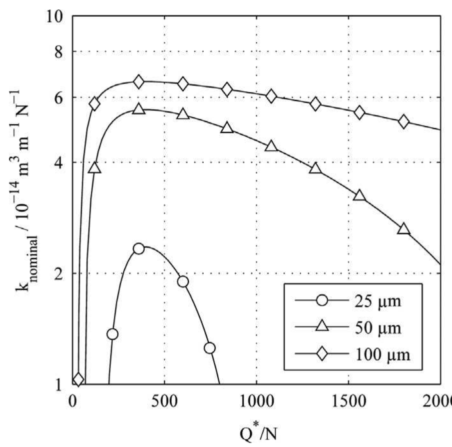

Fig. 13. Variation in predicted $k_{nominal}$ as a function of Q^* for three typical values of Δ^*.

around 10 μm. A reduction to a value 1.0×10^{-14} m^3 m^{-1} N^{-1} has been achieved in all the cases examined by the time Δ^* has been reduced to somewhere between 8 μm and 56 μm. These values span the value indicated by Vingsbo and Söderberg in Fig. 1 of around 15 μm for end of the rapid reduction in wear coefficient with slip amplitude. The values of $k_{nominal}$ predicted in Fig. 12 (for a constant value of k_{energy}) show a strong dependence upon the other parameters associated with the fretting test (N, E_{th} and S), and this may explain the very wide range of values observed in the literature for tests conducted under similar conditions (as illustrated in Fig. 2).

4. Conclusions

In this work, it has been shown that wear coefficients reported in the literature may suffer from errors in their derivation associated with the failure to taken into account differences between δ^* and Δ^*, and the failure to recognise that there may be a threshold value of fretting duration (be that measured by dissipated energy or contact slip distance) below which no wear is observed. It has been shown that these failures will result in the nominal wear coefficient rapidly decreasing from a value close to the true wear coefficient to very much lower values as the applied displacement amplitude is reduced (the rate of reduction depends upon other parameters associated with the experiments being conducted). As such, it is proposed that the results presented by Vingsbo and Söderberg [1] do not in fact indicate that the wear coefficient in fretting is dependent upon the slip amplitude (remembering that they never actually said this, since they framed their work in terms of the displacement amplitude); indeed, given that the magnitude of the change in wear coefficient presented by Vingsbo and Söderberg [1] in their analysis of the literature has been mirrored by the results presented here (which are based upon a model which assumes a constant wear coefficient), we argue that there is no clear evidence that the wear coefficient in fretting is strongly dependent upon the slip amplitude in the gross sliding regime. This main conclusion is supported by a body of more recent work (presented in the literature and in this paper) where fretting loops have been recorded, and the wear data interpreted in terms of the actual slip amplitude (or energy dissipated), taking into account a threshold fretting duration below which no wear has occurred. This work has indicated that the wear coefficient is in fact independent of slip amplitude. In addition, the influence of total sliding distance in the test (fretting duration) must also be addressed. It is therefore suggested that the recording of fretting loops and the interpretation of data to take into account of the threshold in fretting duration below which no wear is observed are essential features of modern research into fretting.

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