Wear 328-329 (2015) 229–240

Contents lists available at ScienceDirect

Wear

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

A new scuffing test using contra-rotation

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ARTICLE INFO

Article history:
Received 24 September 2014
Received in revised form
29 January 2015
Accepted 30 January 2015
Available online 19 February 2015

Keywords:
Lubricated wear including scuffing
Gears
Lubricant additives
Boundary lubrication

ABSTRACT

The mode of lubricant failure known as scuffing provides a significant design constraint in high sliding gears, cams and metal cutting and forming processes. It is therefore important to have an effective test method to measure the scuffing resistance of lubricant formulations. In most existing scuffing bench tests, a moving surface is rubbed against a stationary one at a fixed sliding speed and the load at which scuffing occurs is determined. This approach has two disadvantages. One is that wear of the stationary surface can lead to a large decrease in effective contact pressure during a test. The second is that viscous lubricants often generate significant elastohydrodynamic films at the sliding speeds employed. This means that the scuffing tests measure a complex combination of the influence of the fluid and boundary film-forming properties of the lubricant on scuffing rather than reflecting solely the influence of lubricant formulation.

This paper describes a new scuffing test method in which the two metal surfaces are rubbed together in mixed rolling–sliding with the two surfaces moving in opposite directions with respect to the contact, i.e. in contra-rotation. This enables the sliding speed to be decoupled from the entrainment speed so that the scuffing properties of a lubricant can be determined in boundary lubrication conditions over a wide range of sliding speeds. Also, because both surfaces move relative to the contact, wear is distributed and this minimises changes in contact pressure during a test.

1. Background

Scuffing (often termed scoring in the United States) occurs in lubricated contacts when the lubricating film present in the contact suddenly collapses, resulting in solid–solid adhesion with a consequent very rapid increase in friction and extensive surface damage. It occurs in contacts operating at high pressures and sliding speeds and was first documented in the 1920s when automotive hypoid gears were introduced. This episode led to the development of extreme pressure additives, initially based on lead, sulphur and chlorine, specifically designed to prevent scuffing [1]. Scuffing is still a design barrier in many high-sliding gear configurations, in sliding cam-follower systems and in metal-cutting and forming processes.

Numerous test methods have been designed to determine the conditions at which scuffing occurs and thus to measure the scuffing resistance of lubricants and materials and to explore the mechanisms of scuffing. The first scuffing tests such as the four ball method originated in the 1930s and were developed to address the hypoid gear scuffing problem and to assist in the development of extreme pressure additives [2].

One important limitation of the four ball test [3] and the more modern Timken test [4] is that they involve a moving surface rubbing against a stationary counterpart. This generally results in considerable wear on the stationary surface within the contact, which leads to a large increase in effective contact area and thus to a considerable reduction in contact pressure during a test. This means that systems which suffer high wear often require higher loads to scuff than those which have low wear, a factor that can obscure their intrinsic scuffing resistance. This limitation was recognised in the 1930s, where it was noted that the SAE Extreme Pressure Machine, a disc machine in which both surfaces move relative to the contact, gave more useful scuffing information than tests with one stationary surface [1,5].

It is important to appreciate that scuffing only takes place when all of the protective lubricant films that separate the lubricated rubbing surfaces are destroyed by the rubbing action, i.e. both elastohydrodynamic films due to liquid entrainment and boundary films resulting from lubricant–surface interactions. The penultimate step before scuffing is the removal of the metal oxide film, to expose the nascent metal surface. Since scuffing normally occurs at high sliding speeds, an elastohydrodynamic (EHD) film is often present and the process of scuffing thus involves first the collapse of this fluid

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http://dx.doi.org/10.1016/j.wear.2015.01.080
0043-1648/

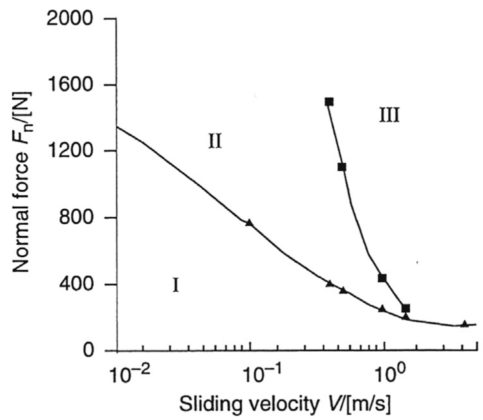

Fig. 1. Transition diagram reproduced from [6] showing the conditions for I EHL lubrication, II boundary lubrication and III scuffing.

film, then the loss of any micro-EHD films present at asperity conjunctions and finally the destruction of any boundary lubricating films. Any of these three films can provide the critical performance barrier to scuffing. This is reflected in de Gee's "transition diagram" [6,7], as illustrated in Fig. 1. This maps the scuffing response of a specific lubricated system in terms of sliding speed and applied load. Three transitions are recognised. At low sliding speeds (on the left hand side of Fig. 1), as the load is increased, first the EHD (or micro-EHD) film collapses (transition from I to II), generally resulting in an increase in friction and wear. Scuffing does not occur however, since a boundary lubricating film is still present. Then as the applied load is increased further, the boundary film collapses at the transition from II to III and scuffing occurs. At high sliding speeds, (on the right of Fig. 1), as soon as the EHD film collapses the contact conditions are so severe that any boundary film is also immediately destroyed, so the system passes straight from EHD lubrication conditions to scuffing, i.e. from I to III.

This diagram reflects an inherent complication in scuffing tests which will be addressed in this paper; that depending on the contact conditions such tests may be measuring quite different things, either the strength of the fluid film or that of the boundary film.

This complication has also imbued research to explore the mechanisms of scuffing. Many scuffing mechanisms have been proposed, as reviewed by Bowman and Stachowiak [8]. The earliest was the flash temperature hypothesis of Blok in the 1940s, which proposed that scuffing occurs when the temperature within the contact reaches some critical value [9]. Although this hypothesis does not require any assumption about the mechanism of film breakdown, Blok suggested the critical temperature corresponded to the breakdown temperature of the boundary film present [10]. Once the existence and properties of elastohydrodynamic lubrication had become established, however, it was realised that scuffing requires the collapse of any EHD film present and this led to models by Dyson and co-workers which proposed that scuffing resulted from loss of film pressure due to heating of the lubricant and the asperities in the contact inlet [11–12]. EHD film collapse also forms the basis of models in which catastrophic starvation results from accumulation of wear debris in the inlet [13]. Recently there has been considerable focus on the behaviour of the rubbing solid sub-surfaces during rubbing and models based on metallic transformation processes initiated by the rapid application of pressure, shear stress and temperature have been developed [14]. It is noteworthy that all the above interpretations of scuffing focus on different stages in the, presumably sequential, film breakdown process.

The aim of the current paper is not to explore the various proposed scuffing mechanisms but rather to describe an experimental test approach which should help both to measure the inherent scuffing-resistance properties of lubricants and to explore mechanism of scuffing in a systematic fashion. This is timely since engine lubricants are currently undergoing major formulation changes, driven by environmental concerns, which involve a reduction in concentrations of the additives that presently control valve train scuffing. There is also a continuing trend to reduce the size of engineered components such as gears so as to lower vehicle mass, and this is resulting in increased contact pressures and thus stronger demands on the lubricant performance.

2. Principle of test method

An often unrecognised problem in scuffing and wear testing is that, when one surface is stationary and the other moving, as is the case in most scuffing and wear tests, any increase in sliding speed also leads to a corresponding increase in entrainment speed and thus EHD film thickness.

If u_1 and u_2 are the speeds of the two surfaces with respect to the contact, the sliding speed, u_s is $|u_1 - u_2|$ and the entrainment, or mean rolling speed, U is $(u_1 + u_2)/2$. The slide roll ratio, *SRR*, is defined as the ratio of the sliding speed to the entrainment speed and is thus

$$SRR = \frac{|u_1 - u_2|}{(u_1 + u_2)/2} \quad (1)$$

In a sliding contact with one surface stationary, $u_2 = 0$, so that, from Eq. (1), the slide roll ratio has the value 2. Therefore in this type of rubbing contact the entrainment speed is always half the sliding speed. In elastohydrodynamic lubrication, the EHD film thickness, h is given by

$$h \approx k(U\eta)^{0.67} \quad (2)$$

where η is the dynamic viscosity [15]. Thus any increase in sliding speed must be accompanied by a corresponding increase in EHD film thickness, a relationship that only breaks down when heat generation due to sliding is so great as to reduce the effective viscosity of the lubricant in the contact inlet.

The above means that in most scuffing tests, as the sliding speed is progressively increased, the thickness (and thus "strength") of the EHD film also increases, greatly complicating our interpretation of scuffing results obtained over a range of speeds. It also means that fluids of higher viscosity will tend to show intrinsically higher resistance to scuffing than those of lower viscosity, regardless of the presence of boundary lubricating additives.

One way round this problem was suggested in a little-known paper by Blok in 1946 [16] and is adopted in the current study. He employed a rolling–sliding contact in which the two rubbing surfaces move in opposite directions relative to the contact, so that u_1 and u_2 have opposite signs. This enables the entrainment speed and the sliding speed to be decoupled so that high sliding speed can be combined with very low entrainment speed and thus negligible fluid film entrainment. The test thus focuses entirely on the effectiveness of lubricant additives and boundary films in preventing scuffing.

Fig. 2 reproduces a figure from Blok's original paper to demonstrate the principle of the approach. He used a twin disc machine and rotated the two discs at the same speed in opposite directions, to give nominally zero hydrodynamic entrainment (Blok's work preceded knowledge of EHD lubrication so he analysed his system in terms of isoviscous-elastic hydrodynamic lubrication theory).

Blok used this approach to investigate the effect of lubricant viscosity on scuffing performance for a set of mineral base oils. His results are summarised in Fig. 3 taken from [16]. His "conventional method" plot shows scuffing load against lubricant viscosity for tests in which the slide roll ratio was 1.75 (i.e. both surfaces moving in the same direction, but one much slower than the other). The "New Method" shows results for the same set of lubricants when the discs contra-rotate with respect to the contact to give at zero nominal hydrodynamic film thickness. In the first case there is a