

*Perspectives***A review of scuffing models**

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The mechanism of wear known as scuffing has been a research topic of great interest for many years. However, the question of how scuffing is initiated and the factors that contribute to its occurrence are still poorly understood. In general, it can be said that scuffing manifests itself as the sudden failure of lubricating films in mechanical equipment operating under extreme conditions of load and/or speed. Components such as cams, tappets, gears and piston rings are all prone to scuffing failure. Understanding the mechanisms that initiate failure would enable the development of criteria for scuffing prediction. This paper reviews the numerous scuffing models and theories that exist today and, in doing so, defines the important factors involved in scuffing initiation. Emphasis is given to existing theoretical scuffing models which have been correlated by a wealth of research data obtained from experimental test rigs.

Keywords: scuffing; wear; scuffing models; flash temperature

1. Introduction

The mechanism of wear known as scuffing has been a research topic of great interest for many years but the factors that contribute to its occurrence are still poorly understood. Even defining what constitutes scuffing has not yet been fully resolved. The majority of research has focused on the macroscopic observations of failure to define the onset of scuffing. Common terminology includes “gross damage” [1], “solid-phase welding” [2] and “adhesive wear” [3]. On a microscopic scale “a roughening of surfaces by plastic flow whether or not there is material transfer” has also been offered as an alternative definition [4]. In general, it can be said that scuffing manifests itself as the sudden failure of protecting lubricating films in mechanical equipment operating under extreme conditions of load and/or speed. The mechanism by which these films collapse is of special significance as it would enable criteria development in scuffing prediction.

It has generally been accepted that the breakdown of full EHL films in the contact is a necessary but insufficient condition for scuffing to occur [3,5]. The formation of micro-EHL and boundary films protects the contacting surfaces and it is the

failure of these films due to an unknown additional mechanism that is in question. Available evidence suggests that film breakdown depends upon the operating conditions, the physical and chemical nature of the lubricant, the surrounding atmosphere and the material properties of the surfaces. The relative significance of each is unknown, however. Nevertheless it is considered that scuffing failure is thermal in nature and frictional heating due to interacting asperities is an important parameter.

This paper attempts to bring together the numerous scuffing models and theories that exist today and in doing so define the important factors involved in scuffing initiation. Emphasis is given to existing theoretical scuffing models which have been correlated by a wealth of research data obtained from experimental test rigs. Firstly, the temperature occurring in the contact is investigated. The contact temperature has been recognised as an important factor in scuffing initiation and a large amount of temperature calculation research both theoretical and experimental has evolved over the years. A number of "contact models" are then investigated where scuffing has been attributed to a number of physical phenomena that can occur in the contact. These models are based on molecular desorption, lubricant decomposition, asperity fatigue, film removal and heat generation. This is followed by investigating a number of scuffing parameters, including minimum film thickness and asperity deformation parameters. Finally, a new model based on catalysed chemical decomposition of the lubricant as a prelude to scuffing initiation is discussed.

2. Temperature models

In nearly all scuffing models and theories today the effect of the contact temperature is of paramount importance. A great deal of research has been undertaken in order to define the role of contact temperature, how an increase in temperature effects the surrounding medium, and how these effects promote scuffing.

The majority of initial research involved predicting a single critical surface temperature rise at which a system will scuff. A variety of temperature prediction formulae were developed and correlated with results from primarily four-ball [6–9] and twin-disc scuffing tests [10–13]. Over time a greater understanding of contact mechanics and contact chemistry has developed [14–16]. This has resulted in calculated contact temperatures being associated with a number of physical phenomena that can occur within the contact. It has been found that any one of these or a combination of these phenomena can lead to scuffing. These phenomena include:

- collapse of the EHL, micro-EHL films,
- desorption of adsorbed surface films,
- thermal or oxidative decomposition of the entrained lubricant,

asperity heating, expansion and deformation,
removal of protective surface layers due to wear within the contact.

2.1. BLOK'S "CRITICAL TEMPERATURE" MODEL

An understanding into the factors promoting scuffing in concentrated contacts was initially proposed by Blok in 1937 in a series of papers introducing flash temperature theory [17–19]. Blok hypothesised that scuffing occurs when the temperature in the contact reaches a "critical value" independent of sliding speed, load and bulk lubricant temperature for a given lubricant/surface combination [20]. The original work was based on non-additive mineral oils lubricating hard steel gears but was unfortunately lost in May 1940 during World War II [21].

An important observation from this early work on scuffing was the notion that this critical temperature is a combination of the contacting materials' bulk temperature and the instantaneous "flash temperature" rise occurring as the surface passes through the contact. The bulk temperature measurement may be made or estimated without difficulty by measuring the temperature of the metal parts. However, measurement of an asperity's flash temperature is more difficult due to the rapid decay of temperature with depth into the surface, only a few micrometers underneath the surface, and the short time of duration. Blok's flash temperature equation takes into account the material properties, the contact geometry, the surface sliding speeds, normal contact load and the coefficient of friction. The problem in practical situations, however, is the evaluation of the friction coefficient, which is also a problem inherent to other empirical formulae.

The Blok postulate has been investigated by many workers, some of whom have found good agreement between the occurrence of scuffing and critical temperature [22,23]. However, the majority of evidence suggests that the critical temperature varies with experimental conditions [7–9,11] and the hypothesis may only be valid for a narrow band of sliding speeds, i.e. of 1.6 to 6.2 m/s [13]. For example, Jackson et al. in their work utilised Blok's critical temperature hypothesis in determining the effect of lubricant friction on scuffing [24]. The critical temperatures were calculated for a range of oils including basestock mineral and PAO oils, mineral and PAO oils containing EP additives, a mineral oil with ZnDDP and a PAO oil containing a phosphorus additive. The results indicated that a large drop in critical temperature occurred when the specific film thickness (λ) was reduced from > 6 to 1.2 even though test conditions remained unchanged. An excellent review of scuffing related research before 1975 based on Blok's critical temperature theory is presented by Dyson [25].

2.2. CONTACT TEMPERATURE CALCULATIONS

Since the introduction of Blok's concept of flash temperature and total contact temperature there was a great deal of effort to expand the knowledge concerning

the thermal behaviour of sliding systems. Other flash temperature models based on Blok's work have been developed by Fein [7], Leach and Kelly [22], Jaeger [26] and Archard [27]. Naturally there is some disagreement between the models. For example, Archard [27] suggested that the EHL film temperature may be more than five times greater than the surface flash temperature, while a flash temperature model proposed by Francis [8], predicts a temperature 33–38% higher than that predicted by Blok's formula. In these initial works the temperature rise of a single asperity in a contact is calculated, where the nominal contact temperature was assumed to remain relatively low.

The problem with calculating the flash temperature rise for a single asperity is that it is based on the assumption that there are no thermal interactions from asperities in the vicinity. For example, Ling [28] assumed a finite number of contacting asperities stochastically distributed in the contact, i.e., the position of heat sources is statistically random both in position and time. It was found that the temperature of these asperities was much higher than the temperature of the surrounding contact area. Other researchers have found that the relative location of multiple asperities can greatly affect the contact temperature even if the total heat generated is constant [29]. Small numbers of contacting asperities experiencing flash temperatures can also cause an additional surface temperature rise above the constant bulk temperature in a body of finite thickness. This is affected by the number and interaction of asperities and lubricant cooling [30].

In general, the above models endeavour to calculate flash temperatures under more realistic conditions and to correlate results with test data. The use of digitised surface profiles by Aramaki et al. [31,32] and Lee [5,33,34] when calculating asperity temperatures is a further extension of this work.

Flash temperature calculations associated with different contact conditions have also been developed. Kannel and Barber [35] predict a flash temperature of 130°C in a sliding contact in which wear alters the interfacial area. Surface films with different thermal properties than the substrate can also have a significant effect on the predicted total contact temperature [36]. For example, decreasing the thermal properties of either the substrate or surface film will increase the contact temperature.

In general, estimating the flash temperatures involved in the contact under highly transient conditions involves rigorous and often complicated analysis. The analysis typically involves the temperature calculation at a single contact event based on a range of experimental conditions such as speed, load, lubricant and material properties, contact geometry, surface roughness and friction data. The range of assumptions include among others, the shapes of asperity contacts, no heat convection into the surfaces and heat generation by asperity interaction only. As a result a large range of critical scuffing temperatures from as low as 150°C to over 400°C have been predicted. These models are based on the coefficient of friction which is difficult to measure accurately. This is one of the major drawbacks of these models and as a result they are difficult to implement as a scuffing criterion in a components design stage [25,37].

2.3. EHL-BASED MODELS

In the mid seventies, Dyson proposed a scuffing model based on the collapse of EHL films found, for example, in cams, tappets, gears and piston rings [15]. Essentially, the high pressures experienced within the Hertzian contact causes the lubricant viscosity to increase almost to the point of behaving like a solid. Offset against this, there is an increase in lubricant temperature due to frictional heating, which reduces lubricant viscosity. When a "critical temperature" is reached the viscosity suddenly falls causing the EHL film to collapse promoting scuffing. The fundamental difference between Dyson's and Blok's model is that according to Dyson the critical temperature occurs at the inlet of the contact while Blok's critical temperature occurs within the central region of the contact [38]. Later, Cheng and Dyson [39] expanded Dyson's model by including the effect of asperity heating at the inlet of the contact and successfully correlated experimental results using a twin-disk machine. From the results, it was proposed that failure occurs when the mean hydrodynamic pressure will no longer permit a high-viscosity liquid film to be formed between the interacting asperities.

Another model that incorporates the effect of micro-EHL lubrication occurring between contacting asperities has been formulated by Cheng [40] and later by Baglin [41]. According to this model, as two asperities approach each other, slide in the conjunction and collide, very thin films can develop which keep the asperities separated. The thickness of the separating micro-EHL film is dependent upon the orientation of the sliding asperities to the direction of sliding, the asperity radius, the sliding speed and most importantly viscosity enhancement by the surrounding EHL film. Without viscosity enhancement, the micro-EHL films would collapse as in Dyson's original EHL model and scuffing would occur. Although it was found in further work by Lee and Cheng [5] that breakdown of the main EHL and micro-EHL films was necessary for scuffing to occur, these were not sufficient conditions for scuffing to occur. Dyson's model modified by Lee and Cheng is illustrated in fig. 1.

2.4. DESORPTION MODELS

Adsorption of a lubricant's most polar constituents onto metallic surfaces has long been recognised as a defence against wear of sliding asperities in both mixed and boundary lubrication regimes. The lubricating effect is caused by the formation of a low-shear-strength interface between the opposing asperities. The rate of removal of these adsorbed molecules is related to the temperature of the asperity, i.e. the higher the temperature the more likely the molecule will detach itself from the surface due to thermal excitation.

In boundary lubrication, where the load is carried by contacting asperities and the lubricant pressure is low, the critical temperature associated with desorption of polar molecules naturally occurring in the lubricant has been found to be in the

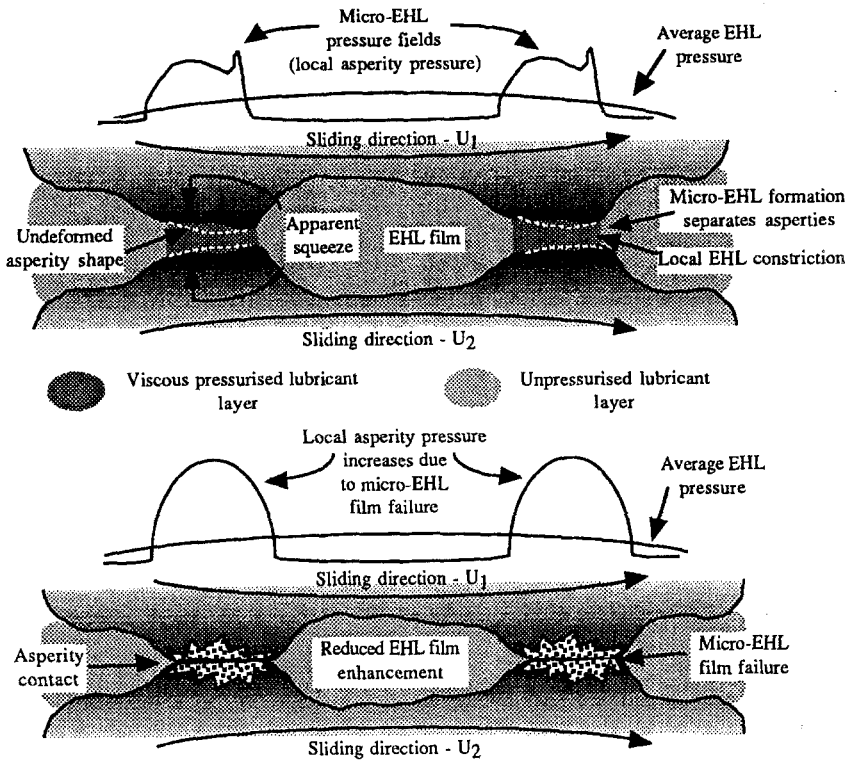


Fig. 1. Schematic illustration of Dyson's model modified by Lee and Cheng. Lubricant viscosity enhancement by EHL films is required for micro-EHL films to provide effective protection against asperity contact.

region of 150°C [6,42,43]. The addition of polar surfactants to a lubricant has been found in some cases to raise the critical temperature [44], but this effect varies according to both the polar molecule employed and its concentration [14]. These observations have been found to be in good agreement with a model developed by Frewing [14]. This model relates the concentration of the bulk polar compound, the heat of adsorption and the transition temperature to the concentration of molecules adsorbed onto the surface (θ). It has been developed based on experiments conducted under slow sliding speeds, i.e. usually less than 3 mm/s, where the asperity flash temperatures were negligible due to lack of frictional heating. However, Frewing's equation has been found to be valid at sliding speeds as high as 133 mm/s [45].

The majority of scuffing-related failure, however, occurs at higher sliding speeds with the formation of EHL and micro-EHL films known as mixed lubrication. Under mixed lubrication conditions pressurised films separate the contacting asperities. The increased contact pressure in comparison to the pressure found in boundary lubrication will, according to Langmuir's molecule adsorption theory

[46], cause an increase in the concentration of protective adsorbed lubricant molecules on the surface. This has the effect of raising the critical temperature of desorption. As a result, a competing process between pressure and temperature occurs, which alters the concentration of adsorbed lubricant molecules on the surface.

Lee and Cheng [33] attempted to correlate adsorption theory, based on Langmuir's work, with calculated scuffing temperatures occurring in twin-disc experiments. The experiments were conducted at contact pressures as high as 0.3 GPa. When the critical adsorbate coverage that is effective in reducing high friction is reached, i.e. about 0.5 [45], scuffing occurs due to a reduced number of molecules protecting the surface. The mean asperity temperature calculated from digitised surface profiles just prior to scuffing was found to be about 324°C, which correlated well with the temperature predicted by Langmuir's desorption theory.

This critical temperature–pressure theory (CTP) was confirmed later by Lee and Cheng [34]. It was demonstrated over a range of partial EHL conditions that the critical scuffing temperature in the contact increases due to higher contact pressures generated in the EHL region. The CTP theory is illustrated schematically in fig. 2.

Molecular desorption as a prelude to scuffing at high sliding speeds has also been investigated by Cameron et al. [44,45]. Under high sliding speeds, frictional heating occurs and flash temperatures become significant. It has been suggested that at these high speeds the contact time necessary for thermal excitation of adsorbed molecules from the surface is too short. Consequently, a higher contact tempera-

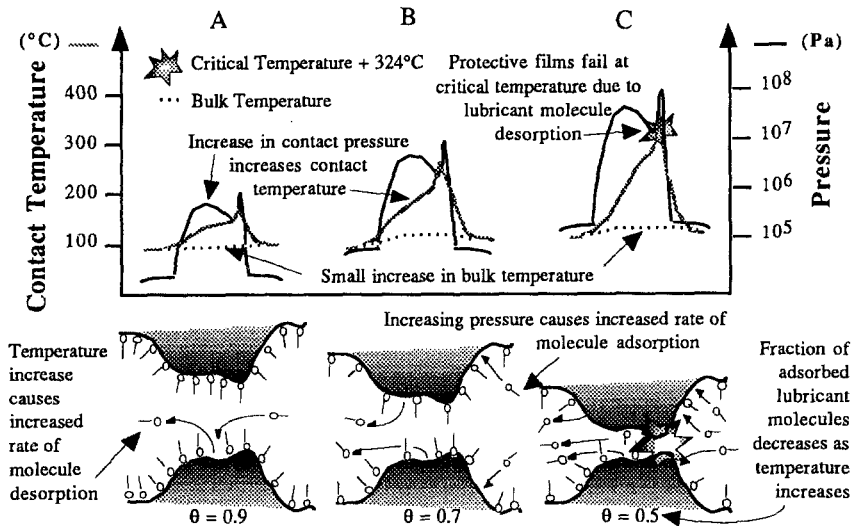


Fig. 2. Schematic illustration of critical lubricant temperature criterion. (A) Low temperatures and pressures, molecules adsorbed on contacting surfaces, (B) temperature and pressure increases, the fraction of lubricant molecules adsorbing/desorbing from surfaces decreases (θ), (C) at the critical temperature the fraction of adsorbed molecules falls below 0.5 resulting in film failure.

ture, and hence higher load, would be required to remove these molecules in the time available [45]. This hypothesis was based on Borsoff's [47] work with high-speed gear tests over 1000 rpm, who found an increase in scuffing load at speeds over 10000 rpm. Other workers also attributed the occurrence of scuffing in a four-ball machine to the desorption temperature of the lubricant [48]. Contact temperatures in the range 240°C, consistent with other findings [7,49], were predicted for various combinations of load and speed greater than 1000 rpm.

However, there were some problems associated with a constant-desorption-temperature model proposed for boundary and mixed lubrication regimes. The transition temperature reported by Khrushchov and Matveevsky [50,43] could not be confirmed for a mineral oil with tool steel pair, although a sharp transition at 150°C was found when a stainless steel pair was used [43]. Also, other researchers have found that a pure white oil operating at high speed in a modified four-ball configuration failed above 250°C, well above the 150°C desorption temperature expected [44].

Clearly, at low sliding speeds the initial work relating molecular desorption at a constant temperature to high friction and failure showed promise. However, inconsistencies with this theory, especially when different materials are used, indicate that other factors, such as metallurgical composition, are important. The significance of slow-speed boundary lubrication tests to scuffing related failure must also be questioned due to the fact that gears, cams and tappets etc., operating at high speed, fail when the pressurised EHL and micro-EHL films collapse. Lee and Cheng's desorption failure model considers these hydrodynamic factors and offers the most plausible explanation to account for molecular desorption as a prelude to scuffing.

2.5. LUBRICANT DECOMPOSITION MODELS

It is generally accepted that temperatures in excess of 150°C and as high as 800°C associated with scuffing are accompanied by untempered martensite transformations in four-ball tests [51]. At these elevated temperatures the role of the lubricant is called into question. A lubricant's ability to withstand thermal and oxidative degradation is dependent not only on introduced additives, such as antioxidants, but also on basestock characteristics. For example, the formation of insoluble sludge and varnish oxidation products in a transformer dielectric mineral oil has been observed after 60 h of oxidation at 140°C [52]. The sludge is initially formed from the free radical reaction between the bulk lubricant and oxygen in the presence of metals to produce lower-molecular-weight products such as ketones and aldehydes. Continual polymerisation of these products into soluble, long-chain condensation polymers then follows until the insoluble high-molecular-weight products form. Under mixed-lubrication conditions the soluble condensation polymers have been referred to as friction polymers [53].

The formation of friction polymers has been observed in a variety of sliding sys-

tems with a wide range of lubricants and metal pairs [54,55]. Chemical analysis suggests that the organic polymers are bonded with metallic iron [54,55] and form a shearable layer on the iron surface [54], which enhances mixed lubrication [16] and protects the system from scuffing. It is generally accepted that failure occurs at higher loads and temperatures, when the rate of film removal is greater than the rate of film formation [54].

A lubricant-decomposition model as a prelude to scuffing has been proposed by Hsu et al. [56]. In their work they attempted to calculate the temperature at scuffing using, firstly, a chemical kinetics model and secondly, a mechanical contact model. The chemical model is based on the formation of soluble friction polymers and depletion of these polymers due to continued polymerisation into insoluble products. The model requires that about 10% of the initial lubricant volume in a contact be polymerised before film failure occurs. The model assumes that there is no oxygen diffusion limitations, lubricant evaporation is negligible and the formation of low-molecular-weight liquid-phase oxidation products is also negligible. The calculated film temperature required to achieve this level of reduction at a mean boundary film thickness of $0.06 \mu\text{m}$ was found to be about 375°C .

The mechanical contact model is based on micro-EHL and EHL film failure model developed by Lee [57,56]. Lee's model applied to the four-ball test results indicated that the critical temperature occurring in the contact is in the range between 150 and 220°C . This was found to be in agreement with temperatures calculated using both the four-ball models developed by Fein [7] and Francis [8].

Clearly, there was a large discrepancy between models proposed and a number of possible alterations were discussed. The most successful modification involved incorporating wear into the mechanical model. This was achieved by redistributing the asperity friction into plastically deformed asperities, which experience higher friction, and elastically deformed asperities, which experience lower frictional forces. This redistribution did not violate the total asperity-contact friction estimated by the rough-EHL analysis. It was found that the maximum attainable temperature calculated under these conditions was about 410°C , which reflects the redistribution of energy within the contact and is comparable to the temperature predicted by the chemical model.

However, contact temperatures greater than 300°C measured by other researchers question the validity of both Blok's and Dyson's theory [38]. At these elevated temperatures, rapid lubricant oxidation or thermal degradation is significant, underlying the need to understand both the mechanical and chemical effects in scuffing initiation.

2.6. ASPERITY INTERACTION MODELS

Recent studies devoted to explain the onset of scuffing concentrate on mechanisms of asperity interaction and plastic flow. As a result a number of theories explaining the role of asperity contact as a prelude to scuffing have evolved. A

scuffing model based on the initial plastic flow of an asperity caused by normal and frictional stresses transmitted through both viscous and absorbed films has been proposed [4]. This model is schematically illustrated in fig. 3. Repeated cyclic plastic deformation of asperities causes them to break away by plastic fatigue. The particles formed agglomerate, work harden and eventually the majority of the contact load is carried by a few larger particles, resulting in large contact stresses. This, in turn, causes an increase in temperature, which chemically alters the protective films, resulting in localised film failure and eventual scuffing. However, no indication is given as to what constitutes chemical failure, whether this is due to film desorption, decomposition or some other process.

This model of plastic deformation causing asperity plastic fatigue is supported by the examination of wear debris collected from four-ball tests [51] and failed cams and tappets [58]. The scuffed surfaces examined indicated high levels of plastic flow and exhibited damage features in the form of delamination indicating fatigue failure. Large quantities of spherical wear particles were also found in the failed cams and tappets, but the significance of this type of wear was not discussed. Spherical particles, however, were found in four-ball tests and were attributed to a thermal-chemical reaction between the ball material and the lubricant [49].

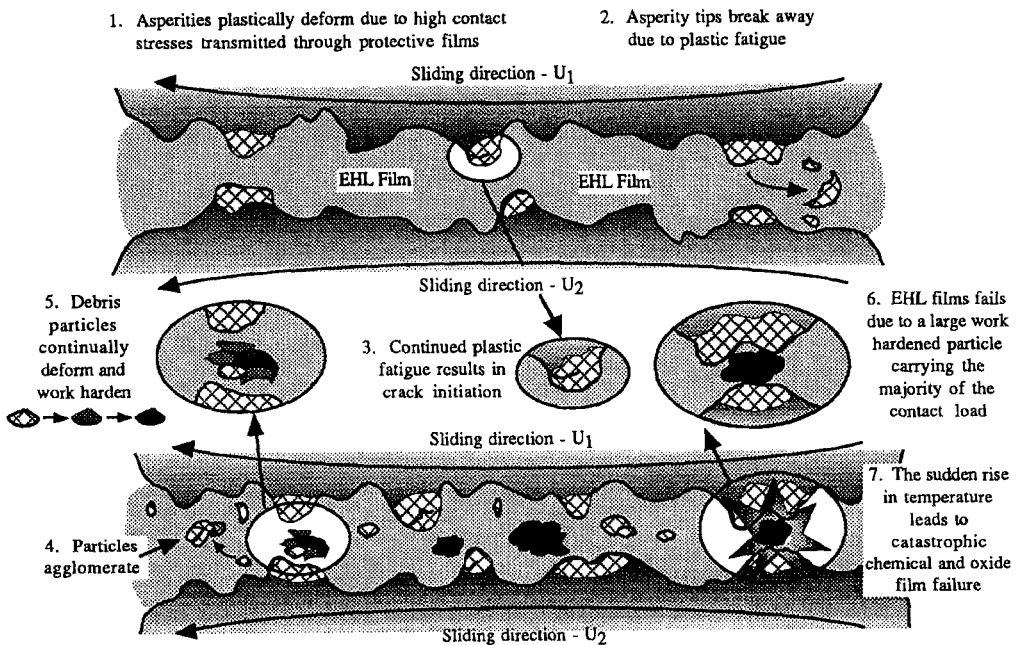


Fig. 3. Schematic illustration of the asperity interaction model proposed by Ludema [4]. Agglomerated debris formed from plastically deformed asperities work harden and carry the majority of the contact load. This in turn causes EHL and micro-EHL film failure, resulting in uncontrolled temperature rises and complete film failure.

Initiation of scuffing has also been explained by an asperity pop-up model assuming a non-Newtonian lubricant at very high shear stresses [59]. As asperities approach each other in the contact, the high pressure and hence the increased Newtonian lubricant viscosity causes the asperities to flatten. Once the lubricant reaches its maximum shear stress in the direction of sliding the lubricant behaves in a non-Newtonian manner and a pressure gradient is formed across the lubricating film. This gradient is sufficient to force the lubricant perpendicular to the direction of sliding and as a result empty the micro-contact. This allows the asperities to pop up and promote scuffing as schematically illustrated in fig. 4. Experimental analysis of rough surfaces confirmed that the introduction of even a small degree of sliding, 0.14 m/s over a slip distance of 2.5% of the Hertzian width, was enough to cause an oil film to collapse in comparison to pure squeeze motion. No other research, however, has appeared to confirm this hypothesis.

2.7. FILM FORMATION AND REMOVAL MODELS – THE ROLE OF OXIDE LAYERS

The importance of surface films is best demonstrated by considering the lines of defence against severe wear. The first line of defence is afforded by EHL and micro-EHL films, followed by chemical films, including extreme pressure (EP) and anti-wear additives, an oxide layer and finally the material itself. In the majority of scuffing-related research, especially in the development of scuffing models, only systems protected by additive-free straight mineral oils have been considered. This is primarily due to the complex nature of absorbed films associated with additive chemistry and a poor understanding of the detailed mechanism of their action. Atmospheric oxygen, however, is inherent in nearly all practical systems and, as a result, understanding the role of oxygen is essential before any reliable model can be developed and used in scuffing prevention.

The formation of protective oxide films either from oxygen present in the atmosphere or oxygen bonded to the lubricant molecule has been found to dramatically

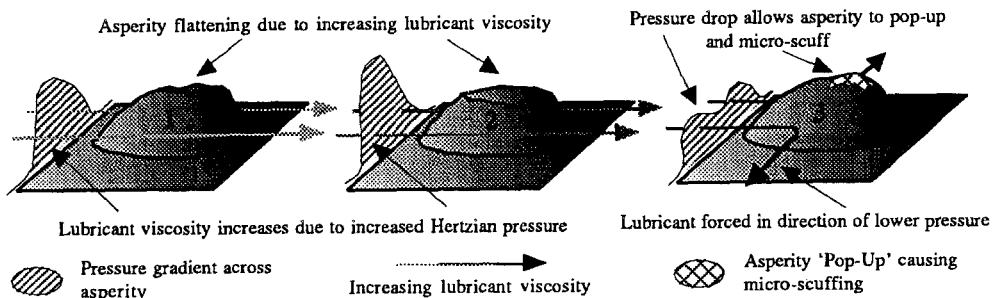


Fig. 4. Schematic illustration of the asperity pop-up model. When the entrained lubricant between approaching asperities reaches its maximum shear stress, the lubricant flows in the direction of lower pressure allowing the flattened asperities to pop up and promote scuffing.

increase the scuffing load during two-disk tests in comparison to tests conducted in an inert atmosphere [11]. It has been proposed that an oxide film thickness of 20 Å is critical in reducing the tendency of steel surfaces to scuff [60]. This depends on the ability of atmospheric oxygen to diffuse through the oil film and the temperature of the substrate. A scuffing relation was then formulated based on the assumption that asperities without protective oxide films are the origin of scuffing. From this it was deduced that even a small area of exposed nascent surface in a contact protected only by an oxide layer is critical to scuffing.

The kinetics of oxide formation and removal has also been proposed by Cutiongco and Chung [61] as a valid scuffing prediction model. As the asperity flash temperature increases, the concentration of adsorbed lubricant molecules on the oxide-covered asperities becomes lower, causing increased oxide wear. At the same time, however, the oxidation rate increases, resulting in a thicker oxide layer. At some critical temperature the oxide removal rate exceeds the oxide formation rate and failure occurs. Asperity flash temperatures from ball-on-disk scuffing experiments operating in a vacuum of 2×10^{-6} Torr were calculated according to procedures developed by Aramaki et al. [31,32]. These results were found to agree with a constructed pressure-velocity failure transition curve. The vacuum of 2×10^{-6} Torr, however, may not be high enough to prevent oxide formation.

Other experimental studies on the formation of boundary films have been conducted with surfaces protected by oxide films alone. For example, the effect of various concentrations of ZnDDP in mineral oil using a cylinder-on-disk machine operating at 0.06 m/s was studied by Choa et al. [62]. It was found that the removal of protective surface films is thought to depend on decreasing film durability rather than reduction of the hydrodynamic fluid film thickness caused by decreasing lubricant viscosity.

2.8. FRICTIONAL POWER INTENSITY, THERMAL STABILITY AND ENERGY MODELS

A criterion frequently used to predict the onset of scuffing is based on the frictional power intensity (FPI) model. The criterion is defined as the product of two parallel vectors, the sliding speed, V_s and the mean shear stress due to friction, i.e. $\tau = \mu p_H$, where μ is the coefficient of friction and p_H is the mean Hertzian stress. The result is the rate of heat dissipation within the contact divided by the contact area. According to this criterion, scuffing will occur when the level of frictional heat generation reaches a critical level [24].

In the earlier studies of scuffing, this criterion provided good correlation with experimental results [25]. Recent studies have also suggested that FPI results are more reliable than Blok's criteria [24,63], although problems can be encountered when the contact area increases during measurements, due to large amounts of surface wear [64]. Also, under real conditions wear and plastic deformation make cal-

ulation of the real contact pressure practically impossible. It has therefore been suggested that friction power, which uses the normal load in contact instead of the contact pressure [63], be used instead. This has been confirmed by other researchers who found that the friction power model [65] predicted scuffing failure better than critical temperature or FPI criteria [12].

The importance of effective heat removal was demonstrated by Crook and Shotter [10], using a twin-disk machine. It was found that disks thermally insulated from the hubs failed at lower loads than a similar set of disks without insulation. This demonstrated effective heat removal is important in scuffing prevention. It has also been found theoretically that a considerable reduction in bulk temperature but no flash temperature change occurs when hollow disks are cooled internally or disks are cooled from the sides [29]. A thermal stability boundary is formed beyond which heat or energy removal is ineffective in preventing scuffing.

Contact thermal stability has been proposed as a scuffing model by Barwell and Milne [66,25]. At a critical combination of speed and load, the temperature rise increases the contact area and friction resulting in an unstable temperature condition. More recently, Kelly and Barnes [67] have described a thermal model, where a boundary is formed in the sliding/rolling speed domain for disks operating under mixed lubrication conditions. Above the boundary, scuffing is expected when the localised asperity tip temperature approaches melting and below the boundary deformation without scuffing is expected. These models have been supported by other researchers who propose that scuffing failure occurs when a wear particle or surface irregularity causes an instability and uncontrolled heat generation results [68]. However, an explanation as to how this heat initiates the onset of scuffing is not provided. A critical contact load expression for sliding disks, beyond which failure occurs, was developed [68]. In later work, this expression provided viable results for specific film thicknesses (λ) greater than 4, but failed to predict scuffing failures at lower film thicknesses ($\lambda \leq 1.3$) under which the majority of practical systems scuff [69]. For example, the critical specific film thickness for gears has been found to be between $0.5 \leq \lambda \leq 1.7$, while the film thickness for cams is $0.25 \leq \lambda \leq 1$ [4].

Thermal stability models define the level of heat a system can withstand before failure. This can also be viewed as a critical level of energy. Critical interfacial energy has been defined as a combination of the thermal energy input, i.e. bulk temperature and viscous heating, and a mechanical energy or mechanical stress [9]. It was observed during four-ball tests that at high thermal energy levels the endured mechanical stresses are low while at high mechanical stresses the thermal capacity of the system is low [9]. Lubricant desorption or decomposition models may also be explained by a critical energy model, with molecules desorbing or decomposing when the critical activation energy is reached. Investigation into energy models remains an area of promising research.

3. Empirical formula

A range of empirical formulae have been proposed over the years to predict the onset of scuffing failure. The formulae utilise measurable quantities including, material properties, surface roughness, sliding speed, applied load and bulk temperature. For example, the local EHL film variation criterion proposed by Tallian [70] attempts to define the transition between safe and unsafe operation of sliding surfaces. In this model, local film variation is characterised as a function of surface roughness and minimum film thickness, i.e.

$$\lambda = \frac{h_0}{(\sigma_A^2 + \sigma_B^2)^{0.5}},$$

where h_0 is the minimum film thickness (m), $\sigma_{A,B}$ is the RMS surface roughness of surfaces A and B (m), λ is the specific film thickness.

Intuitively, it would appear that a λ value < 1 would result in asperity contact leading to a high degree of wear and possible scuffing [4]. However, it has been found that machinery can operate successfully with λ values as low as 0.3 and surfaces can survive λ values of 0.005 provided chemical additives are present in the lubricant and initial operation occurs at higher λ values [71]. It has therefore been suggested that the use of λ as a criterion may only be valid for contacts where no surface films are present, ruling out most practical situations. Also the tendency of a component to scuff cannot be based on initial values of λ , because of the surface roughness changes occurring during sliding [71]. Other researchers explained the limitations of λ as a critical value criterion in terms of the vast number of possible critical values at which a system will scuff. This is dependent on the lubricant, material microstructure, running-in process and surface topography, all of which are interdependent [72].

An asperity deformation model based on the plasticity index (ψ) has also been defined as an indicator of scuffing. The model implies that scuffing will occur within a contact, when the level of asperity plastic deformation reaches 2% [73]. This implies that asperities with greater slopes and made of soft materials will plastically deform more easily and hence they will scuff more easily.

The available literature shows some controversy on this topic. For example, some researchers have confirmed that scuffing occurs in low-speed boundary lubricated contacts due to asperity plastic deformation [73,74]. However, others have found no such correlation and question the validity of the plasticity index in predicting scuffing failure in practical situations, where sliding speeds are much higher and the amount of asperity plastic flow is much greater [72]. The suggestion that scuffing occurs in the early stages of plastic deformation is also not supported by other results, where high levels of plastic flow have been observed [51,58].

Other empirical criteria that fit a range of results include the ‘‘collapse parameter’’ suggested by Begelinger and de Gee [75]. These researchers found that the

load carrying capacity of EHL lubricated contacts operating between 0.0076 and 1.5 m/s can be expressed by, $v^{1/6}P_{\max} = C$, where P_{\max} is the maximum Hertzian contact pressure and C is a constant. The value of C depends on the composition and viscosity of the lubricant [76]. This parameter indicates that failure of EHL and micro-EHL films is a pressure-controlled rather than force-controlled phenomena.

In other work, Chichos and Kirschke [9] defined a "critical failure triplet". The triplet involves a three-dimensional graphic representation of normal load F_N , velocity v and bulk oil temperature T . A failure surface separating effective lubrication from ineffective lubrication is created by plotting the failure load versus combinations of bulk temperature and velocity. Failure occurs when a combination of the above variables lies above this surface. The critical failure triplet in combination with running time t has also been quoted as the failure expression: $F_N^{0.4}v^{0.2}t^{0.03}T$ [49].

In general, problems with empirical formulae are associated with dynamic contact conditions. For example, in running-in, changes in the asperity shapes alter the surface profile, contact pressure and load carrying capacity. This implies that the formulae developed may only be valid for a narrow range of contact conditions.

4. A new scuffing model

Recent work proposes a theoretical scuffing model based on the catalytic properties of the contacting surfaces and the thermal breakdown of protective EHL films [77]. The model tries to incorporate a number of common observations and influencing factors found in other models, such as

- asperity plastic deformation,
- contact temperatures greater than 150°C,
- high pressures within the contact,
- the influence of lubricant chemistry,
- the presence of protective surface films,
- the effect of surface roughness, texture and material properties.

The model is based on the failure of the EHL and micro-EHL films by chemical degradation of the entrained lubricant which has been catalysed by the surrounding surfaces. As two surfaces approach each other in the contact, interacting asperities experience high contact pressures and temperatures due to a viscous shear. This causes some asperities to plastically deform which increases the mechanical activation and raises the catalytic activity of the asperity material. This in turn catalyses the thermal degradation of the lubricant adjacent to the asperity causing localised film failure. The nature of lubricant chemical degradation is not critical. The major factor is that the lubricant has been replaced by a substance of either high or low molecular weight that is far more likely to be an inferior lubricant.

As a result of lubricant film failure, wear of the protective oxide surfaces then fol-

lows. The removal of these protective oxide surfaces may occur within a few revolutions leading to an exposed nascent surface. For example, it has been found that the low solubility and diffusivity of oxygen in oil, especially at high pressures, may allow a nascent surface to persist for as long as one second [60]. The nascent surface has a greater chance of wearing an opposing oxide surface than visa versa [60]. A self-accelerating process of lubricant film failure and oxide removal then occurs in a short period of time leading to complete EHL and micro-EHL film collapse and subsequent scuffing. This process is schematically illustrated in fig. 5.

According to this model, the properties of an ideal scuffing-resistant material can be summarised as follows: a material must exhibit a high hardness to suppress surface deformation, a high thermal conductivity to remove frictional heat and low catalytic activity [77].

5. Conclusions

A large degree of uncertainty still surrounds the conditions leading to scuffing, mainly due to the complex nature of interacting surfaces and the number of operating variables involved. Much useful work has been completed in recent years, especially in the area of chemical reaction products and functionally important

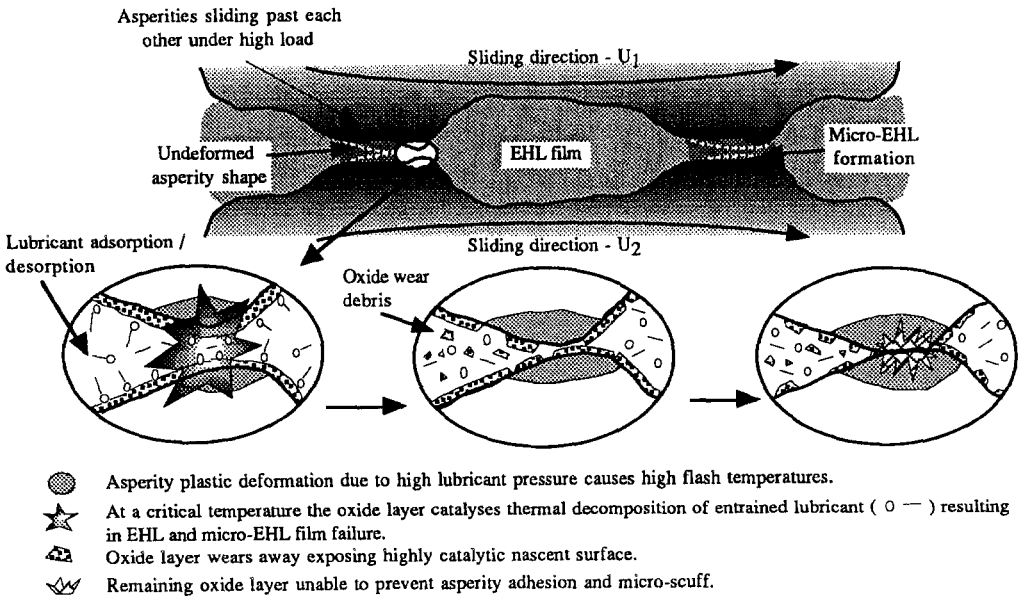


Fig. 5. Schematic illustration of lubricant catalytic decomposition model. At a critical temperature the entrained lubricant is catalysed by the contacting surfaces causing lubricant molecular breakdown. This allows the protective oxide layers to mutually wear away exposing the nascent surface below. When oxide removal exceeds formation, asperity adhesion occurs.

chemical species. Expansion of original empirical scuffing criteria to include the effects such as asperity contact area and deformation, contact pressures, micro-EHL conditions and surface films has also progressed. It has also been recognised that for scuffing to occur, which involves metal-to-metal contact, the protective EHL and micro-EHL films need to collapse first, followed by the removal of surface films.

However, the critical question of how scuffing is initiated still remains unclear. Theories that include molecular desorption, asperity fatigue and lubricant decomposition all involve high contact temperatures but the temperatures predicted have been found to vary greatly. This variation may be explained by surface energy. At a critical surface energy or level of heat generation the functional nature of the lubricant changes by one or a combination of the above models leading, to liquid-film failure. This is followed by the removal of surface films and eventual scuffing. Clearly, defining the role of the lubricant and whether it is involved in scuffing initiation and/or propagation would bring us closer to answering this intriguing question.

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