Wear 269 (2010) 581–589

Contents lists available at ScienceDirect

Wear

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

On the transition from mild to severe wear of lubricated, concentrated contacts: The IRG (OECD) transition diagram

R. Bosman*, D.J. Schipper[1]

University of Twente, Drienerloolaan 5, 7500 AE Enschede, The Netherlands

ARTICLE INFO

Article history:
Received 18 February 2010
Received in revised form 21 May 2010
Accepted 10 June 2010
Available online 16 June 2010

Keywords:
Sliding wear
Thermal effects
Wear modeling

ABSTRACT

In this paper the transition from mild wear to severe wear of lubricated, concentrated contacts is dealt with. It is suggested that this transition is thermally induced. The transition from a mild wear to severe adhesive wear occurs when more then 10% of the surface transcends a predefined, critical temperature. A method for determining this critical temperature is presented. Using a BIM based numerical model including the local implementation of Archard's wear law, for the contact pressure and temperature the transition diagram for a model system is calculated and validated by experiments. The transition predicted by numerical calculations is in very good agreement with the experimental determined transition.

1. Introduction

The last decade an increasing demand for smaller machines/ components transmitting the same power or more is seen; hence the nominal contact pressures increase. Due to these increasing contact pressures components are operating in the boundary lubrication regime rather than in the full film or mixed lubrication regime. As a result the load is carried by the asperities rather than by the lubricant leaving the adsorbed/reacted boundary layers as the final protection against wear. To be able to design such a component in the most efficient way it is preferred to have beforehand knowledge at which nominal load and velocity the transition from mild wear to adhesive wear will take place. In the late 1980s and early 1990s the International Research Group on Wear of Engineering Materials did a lot of research on this topic, developing "IRG (OECD)" transition diagram [1], three different regions are distinguished as shown in Fig. 1. These regions are distinguished on the bases of the recorded friction-time signals.

Regions I/II indicate the save region at which mild wear will occur. Regions III* and III indicate a region where temporary and respectively permanent severe wear will be present. In this study only the transition from region I/II to III* and I/II to III (thick line in Fig. 1) will be studied, since it is assumed that after passing this transition enhanced wear will occur and the component has failed. Usually the transition diagram for a certain oil/system is based on experiments and lacks a predictive model or gives a relatively simple empirical relation [2–8]. However, it gives a good overview of the parameters influencing the transition to severe adhesive wear, the transition from mild to severe wear shifts to higher values for the load when for instance the viscosity is increased or the sample roughness is decreased. In this paper an attempt is made to predict the transition based on the postulate first made by Blok [9] and later on adapted by Lee and co-workers to include the presumed effect pressure has on the critical temperature [10,11]. Blok originally stated in the late 1930s that the transition from mild to adhesive wear is a thermal phenomenon. Using this assumption the transition diagram is given a more predictive nature. The postulate states that if the contact temperature is higher than the critical temperature of the protective boundary layer, the layer will fail and the system will undergo severe adhesive wear. Blok did not succeed in defining a uniform critical temperature for a defined system. This is probably due to the lack of good thermal models at the time able to deal with contact temperatures in a sufficient degree of detail. van Drogen [12] did succeed in defining a critical temperature for the transition from mild to severe wear using a thermal model taking into account the micro-geometry. However, as discussed in that work the influence of wear is not to be neglected, and was taken into account by adapting the macroscopic geometry by increasing the radius of the bodies in contact. In the current paper the Archard wear law is embedded in the contact model using the local pressure profile as an input. The modeling can be separated in three different parts: (1) contact model, (2)

Abbreviations: BEM, boundary element method; BIM, boundary gradient method; CGM, conjugated gradient method; DC-FFT, diskrete convolution FFT method; FEM, finite element method.

* Corresponding author. Tel.: +31 53 4892476; fax: +31 53 4894784.
E-mail addresses: r.bosman@utwente.nl (R. Bosman), D.J.Schipper@utwente.nl (D.J. Schipper).
[1] Tel.: +3153 4892476.

0043-1648/$ – see front matter © 2010 Elsevier B.V. All rights reserved.
doi:10.1016/j.wear.2010.06.005

Nomenclature

Roman symbols

C_p	specific heat [J/kg °C]
D, D_{ij}	displacement influence coefficient/tensor [m/Pa]
E	elastic modulus [Pa]
F_n, F_{in}, F_{nMesh}	normal force total, incremental, local [N]
g_{ij}	gap [m]
h_{ij}, $h(x, y)$	separation/separation tensor/height loss [m]
I_{discr}, I_c	collection of grid points (all grid points/points in contact)
k	specific wear rate [mm^3/Nm]
K	thermal conductivity [W/M °C]
L	length [m]
N_c	number of elements in contact
$p(x, y)$, p_{ij}	pressure field/tensor [Pa]
Ra	average roughness value $((1/N)\sum\sum h_{ij} - h_{ij}^{-})$
r_{ij}	distribution in multidimensional space [m^2/Pa]
S	sliding distance [m]
t_{ij}	search direction [m]
t	time [s]
$T(x, y)$, T_{kl}	thermal influence coefficient/tensor [Θ/Wm^2]
u	surface displacement in normal direction [m]
V	[m/s]
W	wear volume [m^3]
x, y	location of interest [m]
x', x'', y'	location of excitation [m]
Δx, Δy	element size [m]
$\bar{x}_{ij}$	average value of tensor

Greek symbols

ε, ε_{set}	relative error, preset relative error
Θ	temperature rise [°C]
κ	thermal diffusivity [K/ρCp]
μ	coefficient of friction [−]
ν	Poisson ratio [−]
ρ	specific density [kg/m^3]
τ	size of the iteration step [Pa/m]
ϕ	partition factor [−]

thermal model and (3) transition criterion. The different parts will be addressed in this order, starting with the contact model.

2. Contact model

The systems of interest in this study are boundary lubricated thus it is assumed that the contact load is carried by the asperities in contact rather than the lubricant as is assumed in [13]. This assumption makes it feasible to model the contact as a "dry" contact. The effect of the lubricant is only taken into account through a reduction in the coefficient of friction, e.g. the presence of the rather thin protective boundary layers. For dry contact the model used in this study is based on the single loop CGM fist discussed by Polonsky and Keer [14]. In their study a novel method is discussed capable of calculating the elastic contact of large meshes using the B(oundary) I(ntegral) M(ethod) in combination with a multi-level summation method. The efficiency of the model greatly depends on the reduction of iterative loops from two to one, namely only the approach (or rigid body motion) of the bodies relative to each other. Later this method is refined by Liu and Wang by replacing the multi-level summation method by a DC-FFT method, as discussed in [15], increasing the accuracy and calculation speed. This method is based on the basic assumptions that both bodies can be modeled as semi-infinite elastic half-spaces with homogenous properties throughout the bulk. The elastic displacement of the surface due to a pressure field ($p(x, y)$) can now be written as the convolution integral:

$$u(x, y) = -\iint D(x - x', y - y')p(x', y')dx'dy' \tag{1}$$

The influence coefficient ($D(x - x', y - y')$) is given by the Boussinesq formula [16]:

$$D(x, y) = \left(\frac{1 - \nu_1{}^2}{\pi E_1} + \frac{1 - \nu_2{}^2}{\pi E_2}\right)\frac{1}{\sqrt{x^2 + y^2}} \tag{2}$$

Diskretizising Eq. (1) gives at all points of interest (I_{discr}):

$$u_{ij} = \sum D_{i-k,j-l}p_{kl}, \quad (i, j) \in I_{discr} \tag{3}$$

Eq. (3) can be solved using the DC-FFT algorithm if the formula for the influence matrix is known, where

$$D_{ij} = \int_{-\frac{1}{2}\Delta x}^{\frac{1}{2}\Delta x}\int_{-\frac{1}{2}\Delta y}^{\frac{1}{2}\Delta y} D(x_i - x', y_j - y')dx'dy', \quad (i, j) \in I_{discr} \tag{4}$$

The elastic contact problem can now be described by the following equations and inequalities (as discussed in [14]):

$$\sum_{(k,l)\in I_{discr}} D_{i-k,j-l}p_{kl} = h_{ij} + \alpha, \quad (i, j) \in I_c \tag{5}$$

$$p_{ij} > 0, \quad (i, j) \in I_c \tag{6}$$

$$\sum_{(k,l)\in I_{discr}} D_{i-k,j-l}p_{kl} \geq h_{ij} + \alpha, \quad (i, j) \notin I_c \tag{7}$$

with h_{ij} is the original surface separation, α is the rigid body motion and I_c is all grid points in contact.

$$\Delta x\Delta y \sum_{(i,j)\in I_c} p_{ij} = F_n \tag{8}$$

where Δx and Δy are the grid spacing and F_n is the total normal force carried by the contact.

Using a CG based iteration method discussed next, the problem described is solved (for details the reader is referred to [14]). First the displacement of the current pressure guess is computed and from this the current gap:

$$g_{ij} = -u_{ij} - h_{ij}, \quad (i, j) \in I_c \tag{9}$$

The average gap is obtained by:

$$\begin{aligned} \bar{g} &= \frac{1}{N_c}\sum_{(k,l)\in I_c} g_{kl} \\ g_{ij} &\leftarrow g_{ij} - \bar{g} \end{aligned} \tag{10}$$

Here N_c is the current number of elements in contact and I_c are all grid points in the diskretized region where $p_{ij} > 0$. For the new gap the sum is calculated:

$$G = \sum_{(i,j)\in I_c} g_{ij}^2 \tag{11}$$

This value is then used to compute the new conjugate direction t_{ij}:

$$t_{ij} = g_{ij} + \delta\frac{G}{G_{\text{old}}}t_{ij}, \quad (i, j) \in I_c \tag{12}$$

$$t_{ij} = 0, \quad (i, j) \notin I_c \tag{13}$$