Tribology International 82 (2015) 287–296

Contents lists available at ScienceDirect

Tribology International

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

The alteration of micro-contact parameters during run-in and their effect on the specific dissipated friction power

D. Stickel*, A. Fischer

University of Duisburg-Essen, Materials Science and Engineering, Duisburg, Germany

ARTICLE INFO

Article history:
Received 1 August 2013
Received in revised form
25 July 2014
Accepted 27 July 2014
Available online 13 August 2014

Keywords:
Ultra-low sliding wear
Contact analysis
Dissipated friction power
Martensitic steel

ABSTRACT

Tribological systems are subjected to a steady decrease of friction and wear due to ecological and economical requirements. Ultra-low sliding wear rate, in the order of some nanometres per hour, is desired e.g. for gears of wind turbines, valve and drive train components, or artificial hip joints in order to increase service life time. One way to further decrease the wear rates is to manipulate the run-in behaviour in such way that the steady state phase is reached earlier. Concerning the run-in, the surface topography and surface near material parameters are the important factors. With respect to the surface topography the optimal surfaces are under discussion. Due to the very small wear rates which are achieved within the ultra-mild sliding wear regime, classical investigations of wear like weighting and microstructural analysis are made difficult by the scale and extent of occurring wear appearances. Here localized effects of dissipated friction power e.g. friction energy determine the acting mechanisms and, therefore, the performance of tribosystems during run-in. In this study, wear tests and associated numerical contact calculations will be presented and discussed. Thus carburized martensitic steel is milled and afterwards subjected to boundary lubricated sliding wear. This topography was chosen as model due to its periodical shape and distinct shape of asperities. This allows for microscopical investigations of the nominal contact area as well as for deriving input data of sufficient quality for numerical calculations for the real contact area. Corresponding wear appearances are to be expected to take place at the summits of the machining marks. The numerical contact analyses are carried out using the elastic-half-space model, because the also applied simpler approach by Greenwood–Williamson led to practicable results but does not consider contact pressure distributions. From the in-situ measurements of the friction force and the relative velocity in combination with the corresponding local surface topography the contact simulation of the real contact area allowed for the calculation of the local dissipated friction power per contact area.

1. Introduction

Non-conforming rough contacts are one of the classic examples in tribology. Many machine elements like gears and bearings run on such contacts. Today's economic and ecological directives on machinery design claim longer service life with reduced production cost and energy consumption. Following this it is necessary to further reduce friction and wear with an increased robustness of tribosystems in general. Currently a wide range of processes is used as countermeasures to minimize wear by modifying the properties of near-surface material (e.g. Carburizing) and/or applying a hard surface layer [1].

The stress distribution underneath the surface of sliding contacts is a parameter being characteristic for each tribosystem. The magnitudes as well as the gradients depend on the normal load, the coefficient of friction, and the geometry of the contacting bodies including the topographies. Beside other factors these contribute to the formation of wear particles, which then can either leave the contact spot immediately or interact with the contacting bodies as third body [2,3]. The mutual influence towards the formation and the stress distribution is difficult to quantify e.g. due to the unknown mechanical properties of materials in a tribological contact. Still regarding the size of wear particles and, therefore, the origin of their generation, the surface topographies, near surface properties and micro-contact stresses are essential. In order to at least estimate the stresses underneath the contact different micro-contact models are available [4–7]. As an input either real or arbitrary surfaces can be used. Either way, with surface height data contact parameters like the micro-contact area, the number of micro-contacts, the contact pressure etc. can be estimated by means of the statistical Greenwood–Williamson (GW) model [4,5] or the version, optimized for numerical calculations, of the elastic half-space concept [8]. Applying

* Corresponding author.
E-mail address: tribo-lyon2013@sciencesconf.org (D. Stickel).

http://dx.doi.org/10.1016/j.triboint.2014.07.024
0301-679X/

Nomenclature

Variable description (dimension)

$A_{c(EHL)}$	Microcontact area, elastic half-space concept (mm^2)
$A_{c(GW)}$	Microcontact area, elastic half-space concept (mm^2)
A_n	Nominal contact area acc. Hertz (mm^2)
d	Surface separation (mm)
D_{sum}	Summit density per unit area ($1/mm^2$)
$\dot{E}_{ds}$	Specific friction power (W/mm^2)
F_{Fricx}	Friction force in sliding direction (N)
F_{Fricy}	Friction force in perpendicular direction (N)
F_N	Normal force (N)
$f_{sampling}$	Sampling frequency (Hz)
f_{test}	Test frequency (Hz)
M_0	Zeroth spectral moment (mm)
M_2	Second spectral moment (–)
M_4	Fourth spectral moment ($1/mm^2$)
N	Number of surface elements (–)
$N_{patches}$	Number of contacts (–)
P_{ELH}	Contact pressure, elastic half-space concept (N/mm^2)
P_n	Nominal contact pressure (N/mm^2)
R_{asp}	Radius of roughness asperities (mm)
S	Displacement (mm)
T	Time (s)
$u_{z(EHL)}$	Surface deformation (mm)
v_{rel}	Relative Sliding speed (mm/s)
x	Domain point (mm)
x'	Domain point (mm)
y	Domain point (mm)
y'	Domain point (mm)
$Z1$	Surface height body (μm)
$Z2$	Surface height counterbody (μm)
α	Surface bandwidth parameter (–)
Δx	Patch length (mm)
Δy	Patch length (mm)
λ_{CL}	Cut of wavelength low pass (μm)
λ_{CH}	Cut off wavelength high pass (μm)
μ	Friction coefficient (–)
ν	Poisson's ratio (–)
σ_s	Standard deviation of summit heights (mm)

such physical models to wear tests should help to better control the tests and evaluate the outcome. Considering both, the huge amount of influencing factors as well as their multi-scale nature and the complexity of numerical calculations, it is difficult to model every aspect of wear. Thus one has to decide upon certain aspects and idealized test conditions. In doing so wear tests must then meet certain requirements, regarding the signal processing, sample treatment and analysing methods to maintain a database that allows for dependable calculations.

In this contribution the 1st step of an attempt is presented in which only elastic material behaviour is considered. The superior aim of this study is to compare different microcontact models as to the contact parameters and qualitatively validate them against the wear appearances from experiments. Herein wear tests and corresponding numerical analyses are presented in order to gain a quantitative approximation of how much energy per area is dissipated at what position of the nominal contact zone under boundary lubricated ultra-mild sliding wear conditions.

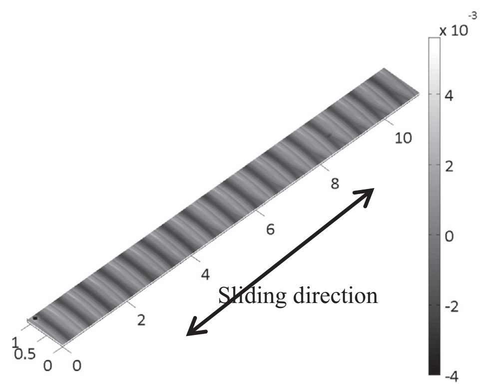

Fig. 1. Unworn surface of body, surface height data in mm (μSurf scan).

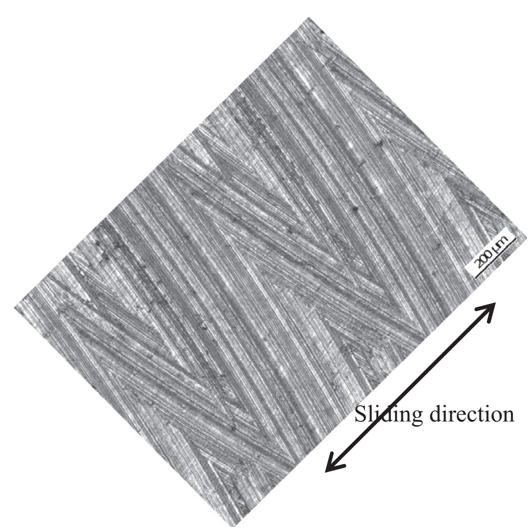

Fig. 2. Unworn surface of body, light microscopy image.

2. Material and method

Both contact bodies consist of carburized steel 18CrNiMo7-6 with a hardness of 650 ± 24 HV10. The surface was milled in order to a specific periodical topography supporting the optical analysing methods as to gain input data of sufficient quality. Figs. 1 and 2 show the unworn surface of the body and an extracted line profile parallel to the direction of sliding in the centre of the wear track (Fig. 3); Figs. 4–6 show this for the counterbody (pin with concave tip) accordingly. Such 2D- and 3D-surface topography data, the normal and friction forces as well as the reciprocating displacement of the counter-body recorded throughout the wear tests, serve as an input to numerical contact calculations.

2.1. Wear tests

Reciprocating sliding wear tests (half-sphere-on-plane) were carried out at $f_{Test}=5$ Hz with a stroke of $s=6$ mm on a custom made tribometer in gear oil (Mobilgear, SHC XMP 320, $\nu_0=335\ mm^2/s$). The radius of the spherical counterbody is 5 mm; the milled body is cuboid with 10 × 10 × 15 mm (height × width × depth). The normal contact force, the friction force in sliding direction F_{Fric_x} and orthogonal to that the second friction force F_{Fric_y} is measured in-situ with a 3-axis dynamometer (Type 9257A, Kistler Instrumente AG, Winterthur, Switzerland) every 200 cycles with a sampling rate of