



The transition between high and low wear regimes under multidirectional reciprocating sliding

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ARTICLE INFO

Article history:

Received 1 February 2008

Received in revised form 26 February 2009

Accepted 23 March 2009

Available online 2 April 2009

Keywords:

Wear mechanism

Transition amplitude

Wear model

Topography

Multidirectional wear

ABSTRACT

The wear phenomena and wear characteristics of reciprocating sliding wear with superimposed lateral vibrations were investigated using a ball-on-disc tribometer. The tribometer enabled two orthogonal oscillations, whereas one oscillation had a constant amplitude of 1 mm (primary oscillation) and the other one had a variable amplitude from 0 to 20.2 μm (secondary oscillation). Ball and disc were made of AISI 52100 steel. The ball surface was polished and the disc surface was unidirectionally grinded parallel to the direction of primary oscillation. Two regimes with different wear rates were found, being separated by a characteristic transition amplitude of $2.7 \pm 0.4 \mu\text{m}$ in the secondary oscillation. This transition correlated with a change of wear mechanisms from tribochemically to mechanically dominated wear. A wear model based on surface topography and particle motion was developed. The wear model is able to predict the value of the transition amplitude by means of characteristic topographical data and the size of wear particles.

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1. Introduction

In automotive engineering many mechanical components like pistons or valves with fuel lubricated steel–steel-contacts undergo a reciprocating sliding motion. Additional to that motion (primary oscillation), often an unwanted second oscillation is observed which can be caused, for example, by vibrations of surrounding parts or components. The direction of this oscillation usually is not parallel to the primary motion. It is of particular interest for engineers to know the influence of this additional secondary oscillation on the wear properties to define the maximum amplitude of this oscillation the system can tolerate.

So far multidirectional wear has only been investigated in connection with surface finishing processes like polishing and lapping. Reciprocating sliding wear or fretting wear on the other hand has only been studied for one-dimensional motion [1–3]. Especially for small amplitudes in the range of a few micrometers, significant differences are expected if a second oscillation is superimposed. The main reason is that stick and partial slip phenomena [4] are impossible if one of the two oscillations does a reciprocating sliding motion.

During the polishing process a multidirectional motion is usually applied by using two rotating discs with dislocated axis. One

disc has a rough surface and the other one a smooth surface with abrasive wear particles in between. The wear process that occurs is called chemical–mechanical polishing. A wear model that describes the development of the surface roughness during the polishing process is based on the Greenwood–Williamson theory of contact and a Hamilton–Jacobi-like conservation equation [5,6]. Another model that describes the material removal during chemical–mechanical polishing originates from the deformation of hyper-elastic asperities attached to a linear-elastic pad [7]. The wear phenomena under chemical–mechanical polishing were investigated by Liang et al. [8]. Edge defects, pits and scratches were often detected and do possibly occur by chemical reactions in combination with asperity contacts under high pressure. Zhang et al. [9,10] investigated the polishing process with fixed abrasive particles. They assumed that the pressure distribution is Hertzian and the abrasion rate follows Archard's wear equation. The experimental verification of the wear model was performed with a cylinder–cylinder-contact of hardened steel and white alumina in synthetic rubber. One cylinder oscillated and the other one rotated. In contrast to these polishing experiments, the present study focuses on multidirectional oscillation without the introduction of abrasive particles.

2. Materials and experimental methods

The wear phenomena and wear characteristics of multidirectional oscillation were investigated by means of a ball-on-disc tribometer (Fig. 1). Balls (hardness 7.2 GPa) and discs

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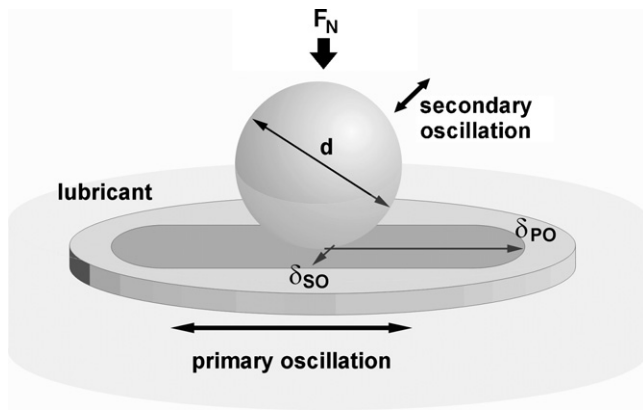


Fig. 1. Scheme of the wear path between ball and disc. The relative speed between ball and disc is generated by two oscillating sliding movements of different speed and stroke lengths. The primary and slower motion is connected to the longer stroke δ_{PO} , which is normal to the secondary and shorter stroke length δ_{SO} .

(hardness 7.8 GPa) were made of high quality AISI 52100 bearing steel with a martensitic microstructure. The diameter of the balls was $d = 10$ mm with a roughness of $R_a = (0.018 \pm 0.001) \mu\text{m}$. The discs were unidirectionally grinded to an average roughness value of $R_a = (0.056 \pm 0.005) \mu\text{m}$ measured orthogonally to the grinding marks. The ball-on-disc system was completely submerged in fuel-lubricant, which was conventional added diesel fuel (specifications acc. to DIN EN 590[11]) with a density of $0.825\text{--}0.835 \text{ g/cm}^3$ (at 15°C) and a kinematic viscosity of $2.5\text{--}3.0 \text{ mm}^2/\text{s}$ (at 40°C).

The primary oscillation was generated by a hydraulic pulsator with a frequency f_{PO} of 1 Hz and an amplitude δ_{PO} of 1 mm. The secondary oscillation was generated by a piezo-electric actuator and directed orthogonally to the primary motion with a frequency f_{SO} of 100 Hz and an amplitude δ_{SO} adjusted to a constant value between 0 and $20.2 \mu\text{m}$ during the test. The grinding marks were oriented parallel to the primary oscillation. Each experiment has been repeated two to three times. The normal load was 15 N and generated by dead weight. Each experiment was performed for 6 h.

During all experiments δ_{PO} and δ_{SO} were monitored every hour by a laservibrometer (Polytec CLV 1000, Polytec GmbH, Waldbronn, Germany) with a sample rate of 50 kHz. The linear wear depth was measured with a temperature compensated displacement transducer and recorded every 46 s. The friction forces F_{PO} and F_{SO} and the normal force F_N were measured by means of a multi-axial piezo-electric force sensor (Kistler 9251, Kistler Instrumente AG, Winterthur, Switzerland) and monitored for one loop every hour with a sample rate of 50 kHz for F_{SO} , F_N and 500 Hz for F_{PO} , F_N .

Ball and disc surfaces were cleaned before and after each experiment with a soft tissue and a solvent consisting of 50% acetone and 50% isopropanol. Afterwards the worn surfaces were visually inspected by means of an optical microscope (Leica DMR, Leica Mikroskopie und Systeme GmbH, Bensheim, Germany) and of scanning electron microscopy (LEO 1450 VP, Carl Zeiss AG, Oberkochen, Germany) with energy dispersive spectroscopy (EDS). The wear volume W_V of the ball and the disc was determined by measuring the surfaces with a confocal microscope (NanoFocus μsurf , NanoFocus AG, Oberhausen, Germany). The wear coefficient k_V was calculated from the measured linear wear depth and size of the worn areas of ball and disc. In order to measure the size of the wear particles, the fuel-lubricant including the particles was extracted by suction after the experiments. Afterwards the fuel-lubricant was evaporated and the diameter of the wear particles was determined with optical microscopy and image analysis software (Axio Vision 4.0, Carl Zeiss

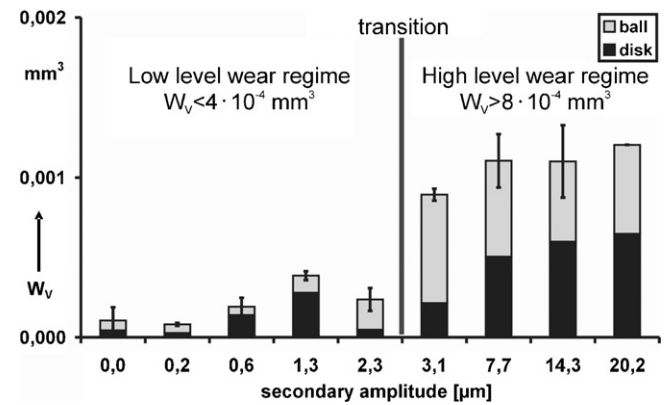


Fig. 2. The wear volume W_V plotted vs. the secondary amplitude δ_{SO} . Between 2.3 and $3.1 \mu\text{m}$ there is a distinct transition from the low to the high level wear regime.

AG, Oberkochen, Germany). The height of the particles was studied by AFM (atomic force microscope) (Park Scientific Instruments CP Autoprobe, Veeco Instruments, Woodbury, USA) in the contact mode. In order to evaporate the fuel-lubricant, the whole disc was heated on a hot plate at about 200°C . Afterwards the topography including wear particles could be determined.

3. Results

3.1. Wear volume and wear coefficient

Fig. 2 shows the wear volume W_V of ball and disc vs. the secondary amplitude δ_{SO} . For $\delta_{SO} \leq 2.3 \mu\text{m}$ the total wear volume is smaller than $0.4 \times 10^{-3} \text{ mm}^3$, which hereafter will be designated as low level wear regime. For $\delta_{SO} > 2.3 \mu\text{m}$ the total wear volume is larger than $0.8 \times 10^{-3} \text{ mm}^3$ and therefore significantly higher. It increases with δ_{SO} up to $1.2 \times 10^{-3} \text{ mm}^3$. This regime is called high level wear regime while the increase of W_V with δ_{SO} is attributed to the total sliding distance growing with δ_{SO} .

In **Fig. 3** the wear coefficient k_V (wear volume per sliding distance) is plotted as a function of the total sliding distance SD for different δ_{SO} -values. Obviously the wear coefficient can also be separated into two regimes. After a run-in period, k_V is nearly constant at about $2000 \mu\text{m}^3/\text{m}$ for $\delta_{SO} \leq 2.3 \mu\text{m}$ and at about $10,000 \mu\text{m}^3/\text{m}$ within the high level wear regime. **Fig. 3** also depicts that the run-in periods are different, depending on δ_{SO} . This can be attributed to the fact that the gross surface area that has to be worn during run-in becomes larger with δ_{SO} as well.

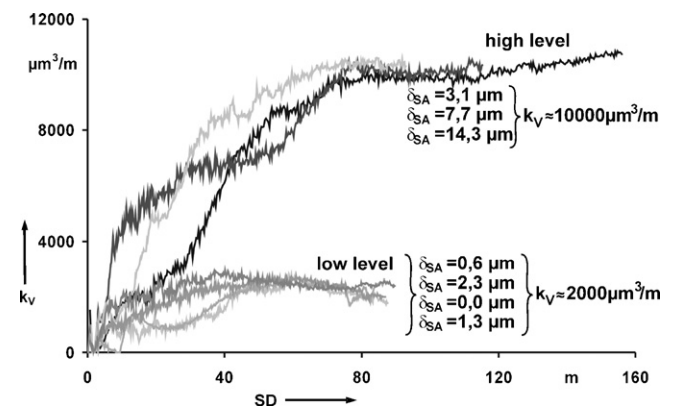


Fig. 3. Wear coefficient k_V plotted vs. the total sliding distance (SD). After run-in both regimes are distinctly separated by their k_V -values.

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