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EXTENDED ABSTRACTS

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INTRODUCTION

We live in times of rapid political and technological change and it is easy to overlook the basic issues of life, which remain largely unaltered. There are global concerns of rapid climate change and environment degradation. There are global concerns to provide enough food and clean water to a large proportion of human population. There are also issues associated with providing adequate mobility and sufficient power to allow people to pursue a civilized life. Many of those issues still remain largely unresolved.

These are the global issues and tribology, in its own way, makes a vital contribution to the resolution of these problems. The cost of parasitic friction generated at the power transmitting interfaces can be quite high, especially for the transportation industry. The cost is not only reflected in the fuel consumption but also in the exhaust gas/carbon emission contributing to the global warming. The mitigation of this parasitic friction might be possible through the application of textured surfaces. These surface textures, i.e. their shape, size, orientation, etc., would need to be tested and optimized before a commercial implementation. A diminution of wear would help to conserve machinery by prolonging their useful lifetime and conserve energy used in their manufacturing.

The importance of tribology in our highly technological society must not be therefore overlooked. Rapid progress in the development of most of the machinery, including high speed trains, aircrafts, space stations, computer hard discs, artificial implants, etc., and our technological advancement have only been possible through the research and advances in tribology. In the ASIATRIB congress series, it is hoped that the latest technical ideas in tribology are freely exchanged to help provide the answers so badly needed. The ASIATRIB series is convened in the spirit of international cooperation and sharing the latest research results and ideas in tribology, thus facilitating the solution of technical problems for the advancement of humankind.

The 4th International Tribology Congress, ASIATRIB 2010, has been planned to share the latest advancements in tribology amongst global research community. ASIATRIB 2010 has brought together the ideas and practices of scientists and engineers working on very many different tribology-related problems. Many papers from all corners of the world have been submitted. The extended abstracts from these papers have been assembled in the ASIATRIB 2010 proceedings which, I hope, will provide a useful reference to all people with an interest in tribology. All the abstracts were checked/edited by at least two people.

The organisation of any conference or congress depends on the efforts of many people and this event is no exception. I would like to express my sincere thanks to the members of the Organising and Scientific Committees and our sponsors, the University of Western Australia, SVT and Society of Tribologists and Lubrication Engineers. Special thanks are given to Grazyna Stachowiak for managing the conference, looking after all the delegates’ needs, organizing the social program, correcting abstracts and answering innumerable email messages. Thanks are also given to Marcin Wolski for setting up and administering the ASIATRIB 2010 website, Tomek Woloszynski, Pawel Podsiadlo, Agata Guzek, Mobin Salasi and Wen-Hsi Chua for making this congress possible.

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ASIATRIB 2010
Tribology in China

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1. Introduction

Tribology research in China has been increasing quickly, like the fast growing economy of China in past 30 years. Two State Key Laboratories in tribology, the State Key Laboratories of Tribology and the State Key Laboratories of Solid Lubrication, were founded in China. In addition, about 26 research institutes or laboratories are quite active in the area of tribology, among them 15 are in universities, 8 in research academe, and 3 in industry companies. There are more than 600 researchers involved in tribology related projects, and 400 Ph.D students or master students are admitted each year in China for studying tribology. According to ISI Web of Science in 2009, China has contributed 194 SCI cited papers to the subject of tribology, 849 papers to wear, and 1019 papers to friction. However, the energies wasted in tribological processes are still enormous and the total loss is estimated at about 327 billion RMB in 2006, which is about 1.55% of GDP of China in 2006. There is much work to be done. In this paper, recent progress in tribology in China will be concisely introduced, with the focus mainly on the following subjects: nano-tribology, bio-tribology, superlubricity, tribology in nanomanufacturing and integration across higher dimensional scales (micro-, meso- and macroscale), tribology in extreme conditions (i.e. heavy load, high/low temperature, high speed, etc.), green tribology and the environment-friendly lubricants, reduction of tribo-noise, environmental protection from pollution by wear contamination, and surface texture and related techniques.

2. Nano-tribology

Nanotribology has been a very hot area in the past 20 years. There are many journal papers and books written on it, particularly in nano-lubrication and nano-friction in which great deal of achievements have been obtained. In nano-lubrication, the research has been mainly focused on the lubrication in a nano-gap or the lubrication regime between fluid lubrication and boundary lubrication. In China, Luo and Wen et al. [1,2] have done a lot of work on thin film lubrication (TFL) which is also called extensive boundary lubrication from 1990s. Some significant progress has been made in this area.

3. Bio-tribology

Bio-tribology, proposed in 1970s, includes tribology in human body and bionic-tribology, which is related to mechanics, material science, physics, chemistry, biology, medicine, etc. In China, there are more than ten laboratories or groups working on bio-tribology, e.g. tribology in articulating joints, heart, eyes, mouth, blood vessel, and on skin, hair, as well as bionic-tribology. Chinese tribologists have obtained some good results in researching adhesion between the animal feet and solid surfaces.

Fig.1 Bio-tribology (a) shell surface, (b) friction of skin

4. Tribology innanomanufacturing

Nanomanufacturing includes both bottom-up and top-down processes. There are many areas in nanomanufacturing related to the tribology, such as scanning probe lithography, assembly and joining, material removal processes in Chemical Mechanical Planarization, and so on. Nanomanufacturing brings some new challenges to tribologists. The interaction between a nanoparticle and a solid surface, the measurement of the movement of nanoparticles, and the realization of a super-smooth surface in CMP have been investigated in China in the past 10 years. A series of experiments surrounding the interaction of nanoparticles with a solid surface have been conducted.

5. Tribology in other areas

In China, tribology in extreme hard condition and surface texture related theory and technique have also absorbed much attention. The tribology under a heavy load, at a high/low temperature, at a very high/low speed, in a high vacuum space, under acid/alkali corrosive condition, etc., have been investigated. Many tribologists are focused on the development of new lubricants and materials to fit the increasing needs. Various techniques like laser machining, mechanical machining, electrical machining, etc., have been employed to produce surface textures. Efforts have also been made to improve the current production techniques and to search for new methods of producing textures.

6. References
Tribology Research and Development in Korea

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1. Introduction
As with any other industrialized nation, the importance of tribology can be found in a variety of industries that are vital to Korea’s economy. Traditionally, tribology has been a big part of technological advances in automotive, electronic, petrochemical, machine tool, steel, and heavy industries. More recently, as bio and nano technologies emerge, the scope of tribology has expanded from traditional friction and wear issues to a broader arena that deals with functionalized surface design and fabrication that cover a wide spectrum of applications. In this presentation, an overview on tribology research and development in Korea is introduced within the context of the country’s industrial base.

2. Background
Korean economy has grown steadily over the last decade following the IMF crisis in the late 1990’s. In 2009, Korea was the 9th largest exporting country in the world and enjoyed a trade surplus of over 40 billion US dollars. Export was led by a variety of industries that include ship building, semiconductor, communications, electronics, automobile, steel, petrochemical, and machine tool. In many of these industries tribology has played an important role in advancement of the respective technologies.

Tribology research in Korea has been led by both academic and industrial members of the Korean Society of Tribologists and Lubrication Engineers (KSTLE), which was established in 1984. KSTLE publishes the bi-monthly Journal of the KSTLE which also carries English papers from the international community. KSTLE also holds Spring and Fall conferences each year where most recent topics in tribology research and development are presented. A major international conference that was sponsored by KSTLE was the 2nd ASIATRIB which was held in Jeju Island in 2002. Another major community related to tribology in Korea is the Korea Lubrication Oil Industries Association (KLOIA) that participates in various efforts and activities related to the lubrication industry.

There are numerous institutions and labs spread all over the country that are dedicated to tribology research. Funds from various government agencies as well as private companies are available to conduct both applied and fundamental research in tribology. A significant program in tribology research was recently launched as part of the Creative Research Initiative (CRI) operated by the Ministry of Education, Science and Technology (MEST). The Center for Nano-Wear was established at Yonsei University in 2010 with the aim to further the understanding of wear and develop advanced wear reduction technologies.

3. Selected Tribology Research Topics

Research and development topics in the Korean tribology community cover a wide spectrum of issues from traditional problems to emerging topics in bio/nano- technology. A good perspective on these topics can be attained by reviewing the presentations made at KSTLE conferences. The list of selected topics is as follows:

University research
- Cell adhesion of micro bioglass
- Adhesion of resins for nano-imprint lithography
- Wear characteristics of CMP pad
- Micro surface texturing for low friction engine
- Susceptibility of brake friction material on humidity
- Friction during polishing process of silicon wafer
- Nano-wear measurement and characterization

National lab and industry research
- Effect of additives on the lubricity of GTL
- Synthesis and characterization of nano carbon grease
- Inter-propeller seal in cryogenic environment
- Mixed lubrication analysis of thrust bearings in scroll compressors
- Friction of tappet in diesel engines
- Friction losses and dynamic analysis of piston pump
- Drag torque reduction of torque converter

4. Summary
A few years ago, the Korean government identified ten major economy driving technologies for the future. Though a significant part of these technologies involves electronics and communications, tribology is expected to continue to play a vital role as design of functional surfaces of various devices become more prominent in future technologies. Furthermore, as energy and environmental issues become increasingly important, tribology research is expected contribute significantly in the preservation of environment and the eco system.

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5. References
Tribology in India

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1. Introduction

The paper covers the present scenario of tribology in India. The publication record of various research organizations and academic organizations are tracked from the early days to 2010. “Scopus” and “Web of science” is used for this purpose. The paper also covers some of the research work being done in some of the R&D labs attached to industries. The various factors that affect research, in general, and tribology research, in particular, in India is also discussed.

Searching the progress of publications in tribology in India has become a rather easy task give the search sites that are available now. I will discuss more in detail the top 10 researchers in the academic field. The top 10 researchers are identified based on the number of publications in top journals of tribology. These include “Wear”, “Tribology International”, “Tribology Letters”, “ASME Journal of Tribology” and some others. The nature of research being done, the quality of papers being published and the citation record of some of the best papers will be highlighted and discussed.

Based on the number of papers being published, it is interesting to note that the intensity of research in tribology in India seems to show some peaks but has serious dips. An example of the number of papers being published through the years in the journal “wear”, “Tribology International” and Tribology Letters” is shown in figure 1.

![Figure 1 - number of papers published in “Wear”, “Tribology International” and Tribology Letters”](Image)

The citation of papers published, of the journals of figure 1, is shown in figure 2. It is interesting to note that the number of citations show a steady increase over the years. This may be due to the more easy access to papers over the world through internet. Or is it that the world is “waking up” to the research being carried out in India?

Data for funding for tribology research is difficult to get as there is no particular committee that exclusively funds tribology research and there is no particular one source to get this information. This data has to be gotten from individual sources. The conclusion drawn is that funding in general has improved and some of the R&D organizations affiliated to industries have been given massive funding for research in the last few years. This “massive” increase is when compared to the funding given during the previous years. It has been a general observation that getting big money to do fundamental research has been rather slow and many-at-times a frustrating experience.

The Tribology Society of India (TSI) needs special mention in this paper and talk. It is an organization that started formally in 1989 and has today grown quite well. It has been organizing the International Conference on Industrial Tribology once every two years since 1994. The last conference is being held (will have been held, by the time the print version of this paper is released) from December 2-4 at Ranchi, Chhattisgarh. This time the expected number of delegates is around 300. Today TSI has a membership of around 1200 and charges a nominal fee of Rs1,500 (AUD~40) for becoming a life member. It publishes the “Indian Journal of Tribology” which is given free to its life members. It also conducts courses around the country to expose scientists and engineers from academia and industry to fundamental and applied aspects of tribology. One must add that TSI which was an organization started by tribologists from industry has been successful in getting academia and industry closer to each other. This gained momentum since the ICIT 2006 when, for the first time, the conference was held by an academic organization at the Indian Institute of Science This fostering of industry-academia relationship is possibly the start of a long and useful partnership which will help tribology realise its potential in India in both fundamental and applied aspects of tribology.

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Tribology in Japan – Past, Present, and Future

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1. Introduction

With the publication of the “Department of Education and Science Report” in 1966, which is also known as the “Jost Report,” the word “tribology” has gained common usage all over the world. In Japan, the Japan Society of Lubrication Engineers (JSLE), which was established in 1956, changed its name to the Japanese Society of Tribologists (JAST) in 1992. JAST has made contributions towards advancing technological achievements in the field of tribology for more than 50 years in Japan.

This paper gives an overview of technological achievements that have been made in the field of tribology over the past 50 years in Japan and highlights various contributions to industry. Furthermore, this paper introduces the history of the development of rolling bearings used in high-speed Shinkansen bullet trains, which are the rolling elements at the core of tribological technology.

2. Technological achievements in the field of tribology in Japan

1) Development of magnetic disk storage for use in computers (1957);
2) Commercial operation of bullet trains made possible with tribological advances (see chapter 3) (1964);
3) Turbocharger mounted to a Japanese automotive engine using high-speed floating bush bearings and a specially developed engine oil (1979);
4) Establishment of engine oil standards for improving automotive fuel economy (1982);
5) Development of journal bearings and lubricants for refrigeration compressors using polyol ester, which replaced the use of chlorofluorocarbons (CFCs) (1989);
6) Use of liquid hydrogen and liquid oxygen turbopumps in Japan’s first domestically developed H-II rocket launch (1994);
7) Practical use of a half-toroidal CVT for passenger vehicles (1999); and
8) Practical use of solid lubricant lead-free overlays applied to journal bearings in automotive engine applications (2001).

3. History of the development of rolling bearings for high-speed shinkansen bullet trains

During the commercial operation of high-speed shinkansen bullet trains since 1964, top speeds have steadily increased. The 300 series bullet train was introduced to commercial operation in 1992 and operated at top speeds of 270 km/h, which was 1.3 times faster in comparison to the Zero series top speed of 210 km/h in 1964. The 500 series operated at a top speed of 300 km/h during commercial use.

For achieving such high speeds in commercial use, improved operational stability and lighter railway cars were necessary. Hence, bearing manufacturers tried to reduce the weight of the bearings, especially axle box bearings. For the axle box bearings of the Zero series, double-row cylindrical roller bearings and ball bearing were used as shown in Fig. 1(a). For the 300 series, cylindrical roller bearings with ribs were developed as shown in Fig. 1(b). Through research on the shape of the rib of the outer/inner rings and roller end face, it was became possible to prevent temperature rise of the bearing from exceeding 80 °C on a test bench under conditions corresponding to railway speeds of 325 km/h.\textsuperscript{1} As a consequence, the ball bearings became unnecessary and the weight of the axle box bearings was reduced from 81 kg for the Zero series to 31 kg for the 300 series. For the 500 series, sealed-type double row tapered roller bearings packed with grease are used. Weight was further reduced to 26 kg, which is less than one-third that of the Zero series.

4. References

Tribology in Australia - Past, Present and Future

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1. Introduction

Rapid development of new technologies, machines, farming methods and medicine over the century has massively improved the standards of living of hundreds of millions of people across the world. In our technological societies we no longer worry whether common disease will affect us or whether there will be enough food for supper. With this apparent progress it is easy to overlook the basic issues of life, which remain largely unaltered. There are global concerns of rapid climate change and environment degradation. There are global concerns to provide enough food and clean water to a large proportion of human population. There are also issues associated with providing adequate mobility and sufficient power to allow people to pursue a civilized life. Most of those issues still remain largely unresolved.

Tribology, in its own way, makes a vital contribution to the resolution of these problems. During the past few decades, tribology has advanced to a level where it is now possible to offer reasonable control of friction and wear and achieve savings in both the resources and energy. Performance expectations from industrial machinery are demanding and so the research into tribology continues at an ever increasing pace. The cost of parasitic friction generated at the power transmitting interfaces is high, especially for the transportation industry. This cost is not only reflected in the fuel consumption but also in the exhaust gas/carbon emission contributing to the global warming. Only tribology can provide the means to mitigate this parasitic friction, for example, through the application of textured surfaces.

2. Tribology in Australia - Past

Australia is a large continent populated by about 22 million people. The distances between major cities are large. For example, Perth is geographically separated from Sydney by five hours flight. Despite its large size and small population Australia’s contribution to tribology is substantial. Work conducted by Australian tribologists, for example, on boundary lubrication and friction had a dominating influence across the world for many decades.

Anthony Michell, from Melbourne, provided the mathematical analysis to lubrication problems leading to the elegant solution of a pivoted pad bearing. These bearings are used today to efficiently transmit thrust loads in ships, pumps, turbines, etc. Frank Philip Bowden, from Hobart in Tasmania, together with David Tabor developed the foundations of boundary lubrication between sliding metal interfaces. John Conrad Jaeger, from Sydney, solved the problem of transient heat phenomena in sliding contacts. He also made a number of important observations on the role of surface finish, gouge formation, and the stick-slip phenomenon. Peter Oxley and Mark Challen, from the University of New South Wales, developed a theory explaining the linkage between friction and wear in partially lubricated systems; the energy required to force the waves to move across the surface is the cause of frictional energy dissipation.

3. Tribology in Australia - Present

Australian academics and engineers have been advancing tribological studies in many different areas. Most of research work is conducted at the universities or centers located in major cities. For example, academics from the University of New South Wales (UNSW), University of Wollongong (UW) and Queensland University of Technology (QUT) have advanced the study of hydrodynamic lubrication. Notably, at UNSW extensive studies of ‘squeeze film bearings’, stability and unbalance response of rotor bearing systems with hydrodynamic bearings were conducted. At the UW pad bearings were extensively studied. Work on friction coefficient during rolling of steels has also been conducted providing vital information to the local industry. QUT provided much needed solutions and expertise to industries such as sugar cane processing and coal-mining, which presented entirely different array of tribological problems ranging from the wear of sugar cane shredders to rail fatigue under extreme axle loads.

At the University of Sydney work is continued on the high performance lubricants, composites, carbon nanotubes and nanoparticle wear resistant materials while at the Swinburne University in Melbourne work is conducted on contact mechanics, wear, rolling contact fatigue, friction and adhesion with applications to railways and also on the development of coating technologies. At Ian Wark Research Institute, University of South Australia, work is done on bio and polymer interfaces, colloids and nanostructures, materials and environmental surface science and minerals processing. At the University of Western Australia, Perth, work is focused on the development of techniques for 3-D surface characterization, optimization of surface texture shapes, synergism between abrasion and corrosion and prediction of osteoarthritis.

4. Tribology in Australia - Future

Future of tribology in Australia will inevitably be linked to mining, mineral processing and agricultural industries. Work will also be focused on nano and bio-tribology, development of new methods for the characterization of textured surfaces. However, long-term future will strongly depend on proper training of the young tribologists.
Tribology Research in the Age of Transformative Technologies

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1. Introduction

The 21st Century began with troubling signs of pending crisis in energy and climate change, as well as hopes for unprecedented technological advances spurred by nanotechnology. Ten years into the century, research efforts in energy, climate change, and sustainability are in full swing as the world recognizes that unless we can solve the grand challenges of energy and climate change, the future of mankind may be doomed.

Renewable energy sources such as wind, hydro-wave, geothermal, and net zero energy buildings are needed. Biomedical devices and materials recovery/recycling/waste mining are essential to create a self-sustaining ecosystem. In fact, the paradigm of human existences is changing from unrestricted expansion to resource-limited choices. Institutions world-wide have called for new and innovative solutions to meet these challenges.

At the national level, after losing jobs and manufacturing capability for 50 years, the creation of green jobs in the US has become a national goal. This calls for the development of transformative technologies, i.e. new technological concepts that are so powerful and cost-effective that it will transform the current technology within two to three years. This results in calls for transformative research from government agencies to private foundations.

Tribology can be broadly defined as the study of materials interfaces to enable the development of durable, energy efficient machineries and devices. The history of Tribology research is filled with incremental improvements and continuous optimization with occasional breakthroughs, what would constitute transformative changes in tribology? The following sections describe some of my thoughts.

2. Friction control

With energy supply declining, fuel economy increase is a world-wide priority. IEA has projected that a 150 mpg for autos may be needed by 2050 to just cope with oil supply taking into account of penetration of electric cars, hybrids, fuel cells, etc. So doubling the current CAFÉ fuel economy every fifteen years may get us to the target by 2055. DOE target for SuperTrucks is to achieve 30 mpg by 2030 fully loaded from the current 5-8 mpg. Light-weighting, electric-hybrid, and elimination of parasitic losses are needed. Scientifically, what are the tribological breakthroughs we need? A complete rethink of surface design including textures, thin films, and built-in surface molecular organization may lead to transformative concepts.

3. Wear control

Self-repairing technology coupled with remote sensing and control may allow the next generation of engines, gears, bearings to become super-efficient self-sustaining autonomous systems. Reports have appeared in the literature suggesting some materials and their catalysts when added to tribological systems can repair the worn surface, restoring the surface to its original tolerances. While these reports do not offer a clear mechanistic understanding and their effectiveness sometimes is erratic, they offer a glimpse of what may be possible.

4. Adaptive materials and lubrication systems

Smart materials that can adjust their properties to changes in environments and operation conditions are clearly possibility in many applications. Using nanoparticles and induced diffusion, surface properties can be adjusted according to thermal, electrical, gradient stimuli. If these materials are introduced into the tribological systems, and coupled with a multiscale, heterogeneous lubrication system that can effect lubrication as the material properties change with stimuli, this may transform our thinking about lubrication design. Smart materials coupled with smart lubrication system may create new opportunities in robotics, autonomous systems, devices that can function in extreme environments with multiple backup systems.

5. Molecular engineered lubricant molecules

The use of petroleum base oils and a mixture of functional molecules (additives) have dominated our lubrication practice for the last hundred years. Mobil and Shell have succeeded in developing simple molecular engineered molecules to control traction based on molecular dynamic simulations. The magnetic hard disk industry relies on a monolayer of purified PFPE molecules in conjunction with diamond-like-carbon thin films to achieve 7 years of durability. It is time to rethink the whole issue of lubricant structures. In fact, nature uses just in time, minimum lubrication with continuous regeneration to achieve superbly functioning systems with biodegradable, non-toxic, and life time durability.

Opportunity exists for clean and biodegradable molecules which can be applied at the minimum amount for a specific application needs and duration. Simple structures with functional groups built-into the structure to achieve lubrication. Sacrificial wear protection will be replaced by permanent surface structures capable of adjusting the organization according to stresses and self-reassembly for repeated protection.
Tribological Aspects of Wind Power Plants
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1. Introduction
It becomes more and more necessary to use sustainable and renewable resources. One possibility is the use of the energy content of blowing winds. Nowadays more than 75 countries worldwide operate wind power plants and the main growing market is still Europe, followed by the USA and Asia.

Especially in Europe it becomes more and more difficult to find suitable onshore places for more wind power plants. As a consequence, offshore places may be chosen as alternatives. Their advantages are the higher energy potential due to higher wind speeds. But on the other side some disadvantages have to be taken into account. These are characterized by higher investment and maintenance costs.

2. Sizes and Dimensions
Nowadays wind energy plants are built with a power output of more than 5 MW. The shafts in these plants are supported by bearings with a diameter of more than 2 m and the rotor diameters can reach 100 m and more. They need towers with a height of 120 m and more. The huge drive line equipment with all gears, generators and auxiliary parts is accommodated in gondolas in which humans can stand in upright position.

3. Tribological Contacts
Frictional contacts are located in the following elements: shaft bearings, gear boxes with gear wheels and bearings, hydraulic systems and yaw mechanisms. In more detail the following machine elements have to be lubricated: self-aligning radial roller bearings, cylindrical roller bearings, tapered roller bearings, ball bearings, tooth gears and worm gears.

4. Operating Conditions
The non-steady operating conditions for wind energy plants are characterized by the following effects:
- Vibrations of the plants resulting in eigenfrequencies;
- Speed changes from slow to fast;
- Extreme load ranges and load changes;
- High sudden load peaks;
- Extreme environmental conditions regarding temperature, humidity, salt;
- Difficult maintenance procedures.

Without optimum lubricating procedures with high performance lubricants bearing and gear failures cannot be avoided.

5. Lubricants Needed
To lubricate and operate the main components the following lubricant types and fluids are necessary:
- Gear oils and gear greases;
- Roller bearing oils and greases;
- Hydraulic fluids.

Common requirements for gear oils include the following tests:
- FZG Scuffing test DIN 51517;
- FVA Micro-pitting test 54/7;
- Freudenberg seals test;
- Foam test (acc. To Flender or ASTM 892);
- FAG and SKF Specifications.

AGMA and AWEA Standards complete the long list of requirements which have to be tested and approved.

6. Oil Inspection and Maintenance
For the evaluation of the operating conditions the following parameters have to be measured continuously and recorded: wind speed, power output, temperature of many parts and vibrations. Remote control equipment is necessary for a long time trouble-free operation.

In addition, a several oil properties have to be measured periodically in order to define a necessary oil change, if specific limiting data are reached.

7. Summary
- Frictional contacts in wind energy plants are found in gears and in bearings.
- Due to the severe operating conditions high performance lubricants (gear oils and greases) and hydraulic fluids have to be used, in order to avoid failures.
- By the monitoring process using remote control systems the conditions of the machine elements and the changing properties of the lubricants can be evaluated.
Polymer Composites in Tribo-Applications with Elongated Maintenance Intervals and Reduced Energy Consumption

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1. Introduction

A great deal of all energy used in industrial countries goes to overcome friction. High friction often results in high wear, and high efforts in industries are needed to replace worn out products with new ones. A better control of wear would result in longer product lifetimes and less energy consumed for replacement production. Controlling and reducing friction and wear is one major challenge in the attempts to reach a sustainable society with low energy consumption and reduced environmental climate effects [1].

2. New Composite Developments

The good news is that recent scientific developments and technical innovations have opened new possibilities for reducing friction and wear in some tribo-applications even by several orders of magnitude. This includes also the use of polymer based composites. The traditional method of reducing dry friction against smooth steel counterparts is to introduce an internal lubricant into the polymer matrix, whereas the use of fiber reinforcements provides a better wear resistance. More recently, an additional use of nano-sized ceramic particles in combination with the traditional tribo-fillers has resulted in further improvements (Fig 1). Relative to the neat epoxy, the wear resistance (inverse of the wear rate) could be improved by almost a factor of 3 after the addition of 4 to 6 vol.% of 300 nm sized TiO$_2$-particles. Adding traditional fillers for wear and friction improvement, e.g. short carbon fibers and graphite flakes, the wear improvement was more effective than just in the case of the nanoparticles. However, a combination of both of them led to a synergistic effect, i.e. both advantageous mechanisms superimposed each other.

3. Tribo-Applications

One example for wear resistant polymer composites is their use for important parts in the paper making industry, e.g. calendar roller covers and roller cleaning blades. Hard, micrometer sized SiC particles in combination with nano-sized Al$_2$O$_3$ particles led to remarkable improvements in a variety of properties of the rollers, including their wear resistance. Improvements in the intervals for re-grinding of the roller surfaces by 30 to 50% could be achieved. Relative to the roller quality of the early 90’s, the lifetime of the rollers could be enhanced by a factor of about 3. Similar conclusions can also be drawn for thermoplastic matrices, e.g. polyetheretherketone (PEEK). The addition of nano-particles led to an improved performance in wear and coefficient of friction at room temperature, but also tested under sliding against smooth steel counterparts at elevated temperatures. This has finally led to the use of these compounds as thin coatings on steel substrates, a material used for hybrid bushings in wear loaded components of automotive aggregates. The tribological performance of this material in comparison to a commercial product is illustrated in Fig 2. Especially at elevated temperatures (up to a testing temperature of 225°C) the new nanoparticle-modified PEEK composites exhibited both a much lower coefficient of friction and specific wear rate [2], leading to a pronounced reduction in fuel consumption and a better engine efficiency.

Fig 2: Coefficient of friction and specific wear rate of a nano-modified PEEK composite and a commercial reference material as a function of temperature

4. References

Optical Approach to EHL Problem Studies

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1. Introduction

Scientific explanation of the fact observed or a phenomenon occurring is usually established through a development of an idea or model and its subsequent experimental and theoretical verification. A serious problem in the tribological studies is that direct observations of phenomena occurring between contacting surfaces are extremely difficult, or sometimes even impossible, to conduct. Discovery of elastohydrodynamic lubrication (EHL) is arguably one of the greatest tribological achievements of the 20th century. This discovery was only possible due to the direct observations of the contact between two surfaces, followed by the development of the numerical models.

In 1949, Grubin presented EHL theory written in English. Dowson and Higginson provided the computer-based numerical solution for the line contact EHL problems in 1959. The direct observations of point contact EHL films were conducted using the optical interferometry technique developed by Archard and Kirk in 1962 and Gohar and Cameron in 1963. Since then, extensive investigations have been carried out to solve the numerous problems encountered in non-conforming machine elements through a number of theoretical, experimental and observational stages.

The purpose of this presentation is to discuss the importance of direct observations in tribology research using the example of point contact EHL films with the results gathered by the author and his co-workers using the classical optical interferometry technique.

2. Surface roughness effects on EHL

The film thickness in EHL regime is of the same order of magnitude as the surface roughness. The surface failures seem to be generated by a local film breakdown and locally produced high pressure. To confirm this, researchers have developed special techniques to observe the behaviour of contacting surfaces with real surface roughness. As the surface features are very small this proved to be experimentally difficult problem to solve. The optical interferometry technique could be used in these studies but there are some problems. The other approach is to use artificially produced single and multiple bumps, dents and grooves.

The observations of local film shape changes produced by these artificially produced single surface defects have proved that the local film moves through the EHL conjunction with approximately the average speed of the contacting surfaces while preserving both its thickness and shape.

The use of multiple defects made clear that a local fluctuation of the EHL film caused by surface irregularities depends strongly on the surface kinematics and the roughness wavelength. The minimum film thickness occurs at the position where the highest ridge overlaps with the side-lobes of the macroscopic horse-shoe shaped constriction and its value depends on the kinematics of the surfaces. The fluctuation of the vertical deformation occurring in the mid-plane of each ridge passing through the EHL conjunction depends on the kinematics and the film factor or the lambda ratio. These facts prove that wear and scuffing occur initially at the horseshoe shaped constriction area when the velocity of the smooth surface is larger than that of the rough surface, and the rolling contact fatigue occurs easily near the mid-plane when the film factor is less than 3.

Furthermore, experimental data contribute to theoretical models and improve the accuracy of numerical simulations.

3. Thermal EHL

The oil film viscosity is a function of the pressure and temperature. Both pressure and temperature within the EHL film rise significantly. The oil temperature is a measure of the average kinetic energy of oil molecules occupying a small local region, so that the temperature at each small local region is independent. Hence, the temperature varies across the EHL film. The temperature rise in the EHL film is mainly determined by the thermal properties and the velocity of the contact bodies. When there is a difference in the thermal conductivity between contacting surfaces and the velocity of the surface having a low thermal conductivity is faster than that of the surface having a high thermal conductivity, a positive pressure is generated and a local increase in film thickness, i.e., a dimple occurs. Such a local variation in film thickness depends also on the contact area and its shape. Thus the thermal EHL is important to the understanding of working performance of numerous machine elements.

The dimple was first observed under the glass and steel contact with oils having a high viscosity-pressure coefficient. It was an accidental observation showing that researchers should never ignore any peculiar phenomena. Deep scientific understanding often develops rapidly through an observation of a series of minor abnormal facts.

4. Conclusions

Direct, real time, observations of contacting surfaces are essential to our understanding of tribological phenomena. The development of new observation techniques is anticipated. The next challenging topic in the area of EHL is to observe the flow pattern of lubricants across the EHL films.
Soft, wet, and slippery: polymers as the key to aqueous lubrication
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1. Natural lubrication with water

Natural lubrication is based on water. Pure water is, however, a very poor lubricant, except at higher speeds when it enters the hydrodynamic regime. Nature improves the lubricating properties at low speeds by the use of polymers: mostly glycosylated proteins, which act to separate the sliding surfaces with a layer of immobilized water.

2. Man-made polymers for aqueous lubrication

Polymer brushes are formed when polymer chains are end-grafted to surfaces in close proximity to each other under a good solvent. The result is that the polymer chains stretch out into the solvent, forming a brush-like configuration. These coatings have a number of interesting properties, including high lubricity, especially when one brush-covered surface is slid against another, prompting the suggestion that these systems are actually mimicking the situation in natural lubrication. Since the pioneering work of Klein, a number of studies have been carried out involving the tribological properties of polymer brushes, and it has been found that under low loads, frictional values are almost vanishingly small, while above a critical load the friction coefficient increases substantially, probably due to the forced interdigitation of the chains.

3. Grafting-to and grafting-from methods

There are two principal approaches to end-grafting polymer chains to a surface: grafting-to and grafting-from (Figure 1).

![Fig.1 Grafting-to (left) and grafting-from (right) approaches to attaching polymers to a surface](image)

The grafting-to method offers the advantage of being experimentally straightforward, since it only requires that the polymer chains be synthesized with a reactive end group that attaches to the surface. The disadvantage is relatively limited surface coverage, since polymer chains that are already attached, sterically hinder the attachment of further chains. In most studies performed in our laboratory, grafting to has been accomplished by means of graft co-polymers, such as poly-L-lysine-graft-poly(ethylene glycol) (PLL-g-PEG), which attaches PEG chains to a negatively charged surface by means of a positively charged backbone.

Brushes formed in this way are worn away from the surface relatively easily, but also readsort (i.e. “self-heal”) if free PLL-g-PEG is present in the lubricant.

Grafting-from approaches involve a chemically more complicated attachment procedure, in that polymers are actually synthesized on the surface itself, growing out of preadsorbed initiator species, and thus allowing much greater grafting density than with the grafting-to approach.

Recently a significant advance has been made in the grafting-from approach, in that ultraviolet-initiated surface polymerization has been carried out very conveniently by means of UV light-emitting diodes. The very narrow spectral range of these light sources leads to a very clean, surface-specific polymerization, and a number of different polymer systems prepared by this method are currently under investigation for their tribological properties. Although the self-healing behavior observed with PLL-g-PEG systems cannot occur with grafting-from brushes, initial tribological results appear very promising, yielding brushes with very high lubricity as well as impressive wear behavior.

4. References

Green Tribology
The Way Forward to a Sustainable Society

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1. Introduction

As the earth has been faced to serious energy and environmental problems now, building a low-carbon economy became an urgent mission for existence and development of mankind. However, to tap the latent potentialities of saving energy into full play has peculiar significance for developing low-carbon economy. Obviously, green tribology is duty-bound.

Green tribology is the science and technology of the tribological aspects of ecological balance and of environmental and biological impacts. The expression “Green Tribology” was first used by the present author in China as early as 2001 and raised again it before the tribologists of every country in the world to the Fifth China International Symposium on Tribology (5th CIST 2008) in 2008 in Beijing. Later on, Professor Jost put it as the subject of his opening address to the Fourth World Tribology Congress (4th WTC) in 2009 in Kyoto.

2. Main Objectives and Mission

Though green tribology is within the concept of tribology, it is particularly emphasized those considerable important aspects in today’s environment.

The main objectives of green tribology are the saving of energy and materials and the enhancement of the environment and the quality of life. Its mission is to research and develop the tribological technologies of reaching the above objectives, thus making the sustained artificial eco-systems of both tribological parts and tribo-systems in the course of lifecycle.

3. Main Contents

3.1 Tribotechniques and its integrated technologies for the saving of energy and materials, including various technologies of friction reduction and wear resistance.

Friction reduction is the most important measure for the fuel efficiency improvement. Hayasi and Fuwa have shown seven approaches (Fig.1):

3.2 Tribotechniques and its integrated technologies for removing or reducing the harmful effects to ecological balance (including health) produced by both tribological parts and tribo-systems in the course of lifecycle, including various low-carbon lubrication (green lubrication) and noise-reduction lubrication techniques, lead free bearing and life cycle assessment (LCA) applied to tribological technologies and so forth.

3.3 Research on the tribological aspects of natural environment and natural disaster, mainly focused on the role, mechanisms and effects of friction.

4. Prospects

Green tribology has certainly provided many technological supports to solve the serious problems emerged on a global scale over the years, but it is far from over, and calls for efforts directed toward the further development.

1. Devoting major efforts to the spreading and making practical application of existing knowledge and technologies of green tribology.

2. Developing new green tribological technologies, such as novel coatings, green lubricants and so on.

3. Methods of analyses and evaluation of sustainability for tribological parts and tribo-systems, and tribo-techniques.

4. Tribo-techniques to support energy diversification and hybridization.

5. Concluding Remarks

There have great possibilities for green tribology to develop low-carbon economy and to deal with the climate change and energy crisis on a global scale. Therefore, green tribology is one of the ways forward to a sustainable society.

6. References

Optimal PIFS Models for Characterization of Textured Surfaces in Hydrodynamic Bearings

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1. Introduction

Textured surfaces can increase load capacity and reduce friction coefficient in hydrodynamic bearings. Currently there is no generally accepted method that could provide an accurate and automated 3D characterization of the surfaces.

A promising way to characterize textured surfaces is partition iterated function system (PIFS) method\(^1\). In the method, a surface image is represented as a set of contractive affine transformations (i.e. \(PIFS = \cup_{j=1}^{n} f_j(\text{DOM}_j)\)), called a PIFS model. The model encapsulates information about 3D topography of textured surfaces such as dimples depths, sizes and localizations. Once obtained for a surface image, the model can be applied iteratively to any initial image, and the original surface image can be reconstructed. However, the reconstructed image is not an exact copy of the original, since some of the texture details are lost. Therefore, before PIFS can be reliably used in the characterization of textured surfaces in hydrodynamic bearings, effects of this information loss on load and friction need to be studied.

In this paper, this issue was addressed using a fully textured hydrodynamic parallel pad bearing with elliptical dimples.

2. Methods

The pad bearing with elliptical dimples was modeled as a parallel square slider with one surface textured and other surface smooth (Fig.1(a)). This bearing configuration was chosen since hydrodynamic pressure is generated by individual dimple effects only (i.e. local cavitation at each dimple). The pressure distribution was calculated using 2D Reynolds equation for steady-state incompressible Newtonian fluid in a laminar flow. The equation was solved numerically using a MultiLevel grid method\(^2\).

Effects of information loss in PIFS models were studied by calculating differences in load capacity and friction force obtained for original textured surface (Fig.1(b,c)) and reconstructed images. Four textured surfaces of different complexity were used. Each surface exhibited 64 uniformly distributed elliptical dimples. Two examples of range-images of the surfaces used, along with corresponding pressure distributions, are shown in Fig.1(b,c) and (d,e) respectively.

3. Results

Textured surface range-images were encoded into PIFS models. Accuracy of PIFS models depends on a number of parameters\(^3\). If parameters are not selected correctly, a considerable loss of surface details may occur, and subsequently, there would be errors in calculations of load and friction. Thus, values of the parameters that minimize the loss would need to be found. For this purpose, for each surface, an exhaustive search on PIFS parameters was performed by minimizing Baddeley’s distances\(^4\) between the original and reconstructed surface images.

Optimal PIFS models were found that minimize the loss, and subsequently, the effects on load and friction. The load and friction calculated for the optimal PIFS models differed slightly (i.e. <2\% and <0.04\%) from those calculated for the original surface images.

4. Conclusions

For optimal PIFS models, results obtained showed that effects of information loss on load and friction are negligible. Thus, PIFS might become a useful tool in the characterization of textured surfaces in hydrodynamic bearings.

5. References

Friction and Wear of PEEK Composite Sliding against Rough Steel Ring at High Speed in Oil Lubrication

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1. Introduction

Polyetheretherketone (PEEK) is a high performance thermoplastic polymer. A large number of papers on the tribological properties of PEEK materials have been published1. However, the tribological properties under severe lubricated conditions have not been studied in detail. In this paper, the friction and wear behaviors of PEEK materials, when slid against rough mating surface under oil lubricated condition and high speed, were studied. The friction tests were repeated and the durability to the repetition of severe sliding friction was evaluated and discussed.

2. Experimental apparatus and procedure

Experiments were carried out with a block on ring wear tester2. The ring temperature was measured with a thermo-couple of diameter 0.5mm, which was located at 1mm below the surface. The frictional torque and the fluctuation of rotational speed were measured. The testing materials are summarized in Table 1. The ring was a forging steel (SF540A) and the block was PEEK composite filled with 30wt.% of carbon fiber. For the comparison, a white metal (WJ2) and unfilled PEEK were also tested. The experimental conditions are summarized in Table 2. The friction tests were repeated without replacing new specimens after the ring had cooled down to room temperature. They were repeated up to 10 times.

Table 1 Properties of testing materials.

<table>
<thead>
<tr>
<th>Material</th>
<th>Hardness</th>
<th>Ra (㎛)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ring</td>
<td>SF540A</td>
<td>HV189 8</td>
</tr>
<tr>
<td></td>
<td>PEEK</td>
<td>HRR126</td>
</tr>
<tr>
<td>Block</td>
<td>PEEK comp.</td>
<td>HRR124</td>
</tr>
<tr>
<td></td>
<td>WJ2</td>
<td>HV26</td>
</tr>
</tbody>
</table>

Table 2 Experimental conditions

<table>
<thead>
<tr>
<th>Sliding velocity</th>
<th>10.2 (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load</td>
<td>588 (N)</td>
</tr>
<tr>
<td>Test duration</td>
<td>~15min. for each friction test</td>
</tr>
<tr>
<td>Lubricant</td>
<td>Turbine oil(ISO VG46), flow rate : 40cc/min., oil temp.:30 3</td>
</tr>
</tbody>
</table>

3. Results

Fig.1 shows the relationship between the friction coefficient and the run time obtained in the 1st test of WJ2. The friction coefficient in WJ2 fluctuated over a wide range of 0.01 to 0.11 as well as in unfilled PEEK. Thus the 2nd test for their materials could not be conducted as the catastrophic wear occurred only in 1st test. Fig.2 shows the relationship between the friction coefficient and the run time obtained in the 1st, 5th and 10th tests of PEEK composite. The friction coefficient didn’t depend on the repetition of friction and showed almost the same trend. Its value at the steady state was almost constant at 0.05-0.06. The ring temperature increased up to 150℃ and its trend was also similar irrespective of the repetition of friction. The average specific wear rate of composite from 1st to 10th test was small and of the order of 10^-8 (mm^3/Nm). Based on the SEM observation and EDS analysis of wear debris and wear scar, the wear mechanisms were studied and discussed.

4. Conclusions

The PEEK composite had the high durability to the repetition of severe lubricated sliding friction.

5. References

Mixed Lubrication Analysis of an Offset Barrel Faced Top Compression Ring in a Reciprocating Engine

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1. Introduction
The thermal efficiency of a reciprocating engine can be improved by reducing the mechanical friction loss. The frictional contribution of the piston assembly to the total friction losses is significant. The ring with lower tension has an effect on reducing the friction loss. It, however, has difficulties in the functions of controlling the lubricant consumption and sealing the combustion gas. The offset barrel faced top compression ring has been employed as measures for the problems on the lubricant consumption and the combustion with higher gas pressures. A theoretical analysis is presented on the mixed lubrication of offset barrel faced top compression rings. The characteristics of film thickness, friction and lubricant transport for the offset top ring are examined.

2. Basic theory
The analysis comprises Patir and Cheng’s average flow model1-2 and Lee and Ren’s asperity interaction model3. The former model determines the mean hydrodynamic pressure generated between ring and liner surfaces, and the latter model gives the asperity contact pressure and the real contact area. The analysis is developed to consider the shear thinning effect of the multigrade lubricant.

3. Results and consideration
Numerical calculations are conducted for the mixed lubrication of a piston ring pack in a 4-stroke cycle, 6 cylinder diesel engine with a bore of 135 mm and stroke of 140 mm. The piston has two compression rings and an oil control ring. The offset barrel faced top ring is selected as the subject of present calculations. The top position of sliding surface is offset to the second ring side as illustrated in Fig.1. The loci of nominal instantaneous minimum film thickness \( h_0 \) are calculated on the condition of an engine speed \( N = 2200 \text{ rpm} \) and a 4/4 engine load, and then calculations of friction force \( F \) and nondimensional net lubricant transport \( Q_{\text{net}} \) are made. The frictional characteristic values of maximum friction force \( F_{\text{max}} \) and frictional mean effective pressure \( P_f \) are obtained from the cyclic variation of \( F \).

Figure 2 shows the effects of offset ratio \( B_l/B \) on the characteristic values of \( F_{\text{max}} \), \( P_f \) and \( Q_{\text{net}} \) for the offset top ring. \( F_{\text{max}} \) is generated just after the compression TDC due to the high combustion gas pressures. With a decrease in \( B_l/B \), \( F_{\text{max}} \) decreases since the external ring load is lightened. The offset top ring improves the lubrication condition around the compression TDC. On the other hand, \( P_f \) slightly decreases, reaches the minimum at some value of \( B_l/B \) and then increases. As the offset ratio decreases, \( Q_{\text{net}} \) increases. The positive sign of \( Q_{\text{net}} \) indicates that the lubricant flows towards the crank case. The offset top ring limits the upward lubricant flow.

4. Conclusions
The offset barrel faced top ring reduces the friction spike and improves the lubricant transport characteristic. The friction loss increases when the offset ratio is excessively small.

5. References
Preliminary Study of the Effect of Micro-Scaled Dimple Size on Friction and Wear Under Oil Lubricated Sliding Contact

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1. Introduction

The UNSM technology is a patented technology which was developed and commercialized by DesignMecha Co., Ltd.1 The tungsten carbide (WC) ball is attached to the ultrasonic device and strikes the surface of a workpiece 20,000 or more times per second with 1,000 to 10,000 shots per square millimeter. This process improves surface integrity, hardness, produces micro-scaled dimples and induces compressive residual stress in surface layers.2 The main object of this research is to understand the effect of micro-scaled dimple size on tribological characteristics under oil lubricated sliding contact.

2. Experimental

Tribological experiments were conducted with a pin-on-disk tribometer using ball-on-disk contact geometry. The disk specimens used in this research are made of SAE1045 carbon steel, are 30 mm in diameter and 10 mm in thickness, and the ball is made of silicon nitride (Si3N4) 3 mm in diameter. The UNSM treatment parameters are shown in Table 1. Typical ball-on-disk setup is shown in Fig.1. Table 2 shows the test conditions.

Table 1. Detailed UNSM treatment parameters to produce various micro-scaled dimple sizes

<table>
<thead>
<tr>
<th>UNSM Cond.</th>
<th>Feed Rate, mm/rev</th>
<th>Amplit., µm</th>
<th>Spindle Speed, m/min</th>
<th>Load, N</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>0.07</td>
<td>30</td>
<td>30</td>
<td>40</td>
</tr>
<tr>
<td>II</td>
<td></td>
<td>40</td>
<td></td>
<td></td>
</tr>
<tr>
<td>III</td>
<td></td>
<td>50</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig.1 Principle of the ball-on-disk tribometer

Table 2. Test conditions

<table>
<thead>
<tr>
<th>Force, N</th>
<th>Linear Speed, m/s</th>
<th>Sliding Distance, km</th>
<th>Track Radius, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>0.15</td>
<td>1</td>
<td>9</td>
</tr>
</tbody>
</table>

3. Results and Discussions

Fig. 2 shows the size and profile of single micro-scaled dimple on the disk specimen which were measured by AFM.

Diameter: 1.7 µm
Depth: 0.198 µm
UNSM-I

Diameter: 1.25 µm
Depth: 0.073 µm
UNSM-II

Diameter: 1.0 µm
Depth: 0.054 µm
UNSM-III

Fig. 2 Magnified view and profilometry of the single micro-scaled dimples with different diameter and depth

The coefficient of friction of the micro-scale dimpled surfaces is much lower than polished surface as shown in Fig. 3.

Fig. 3 Friction coefficients of the polished and the UNSM-treated surfaces as a function of sliding distance

4. Conclusions

The preliminary research on micro-scaled dimple size has shown that the UNSM-treated surfaces have lower friction coefficient than polished surface and it is feasible to use a micro-scale dimple depth with 0.198 µm and diameter 1.7 µm on a workpiece surface in order to obtain the lowest friction coefficient. Proper dimple size is essential to improve friction properties.

5. References

1. Information on http://www.designmecha.co.kr
Studies on Reduction of Frictional Power Loss for Internal Combustion Engines

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1. Introduction

In order to prevent the global warming by the improvement of fuel economy for internal combustion engines, some past, present and future studies on the reduction of frictional power loss have been reviewed, considered, foreseen and summarized as below.

2. Total frictional power loss

The total friction loss has been measured by the run-out method for fired and braked engines. The total friction mean effective pressure \( P_{tf} \) has been examined for some design factors concerned with the piston system and the lubricating oil. As shown in Fig.1 the effects of the surface roughness or coating of piston components on the reduction of \( P_{tf} \) have been evaluated, and the influence of the oil deterioration and soot contamination on \( P_{tf} \) has been examined, followed by those of the oil properties, namely lowering the viscosity of multi-grade oil and applying the low-SAPS oils to protect the diesel particulate filter and the deNOx catalyst in the after-treatment device of exhaust-gas.

3. Friction between cam and follower

The other studies regarding the tribotechnology for the valve train to reduce the friction have been reviewed so that the research and development points, namely design concepts for cam and follower elements, have been investigated and summarized.

Firstly, through the experiment with a cam / slipper follower friction test rig, for the cam follower with sliding contact, it has been made obvious that the hard materials combination and the small surface roughness are effective to improve the running-in and lubricating properties. In particular the combination of the surface coating and the oil property effectively acts in the friction reduction. As shown in Fig.2 the compatibility of follower DLC coating with low-SAPS oil SAE5W-30(DH-2) is the most effective to reduce the friction.

Secondly, through the experiment with a cam / roller follower friction test rig it has been made obvious that the effect of the rolling contact between cam and follower on the friction reduction becomes large under the low camshaft rotation speed as shown in Fig.3.

Lastly, according to a mechanism of newly designed direct-type valve-lifter (2) with a shim (3) offset on camshaft (1) shown in Fig.4, the hybrid of sliding and rolling contacts probably reduces the friction.
Optimization of Grease Properties to Prolong the Life of Lubricating Greases

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1. Introduction

In grease lubrication of rolling element bearings, the life of greases due to poor lubrication, caused by grease degradation, is often shorter than the bearing fatigue life due to the pitting of ball or race. In high speed conditions, the extension of the life of greases would be of great interest. In this study, grease properties that could affect the grease lubricating life were evaluated. The effects of grease properties on the life of greases were considered using multiple regression analysis. The aim was to optimize the properties to prolong the life of lubricating greases.

2. Evaluation of grease properties and grease lubricating life

The mechanism of grease lubrication1), where the base lubricating oil is held in a thickener, is critically related to the life of greases. Grease properties that contribute to the lubrication mechanism, such as the onset temperature of oxidation, rate of evaporation, rate of oil bleeding, viscosity of base oil, oil separation and additive concentration, should be considered when optimizing the grease formulation. Therefore, these properties were evaluated for several greases. The grease lubricating life was evaluated using a rolling four-ball tester.

3. Correlation between grease properties and grease lubricating life

The effects of grease properties on the grease life were analysed using a multiple regression model. The model was defined using the grease lubricating life as explained variable, and grease properties as explanatory variables, as:

\[
\text{Life} = \sum a_i X_i^{a_i},
\]

where \(a_i\) and \(\alpha_i\) are coefficients of the model and \(X_i\) is a dimensionless grease property, related to grease lubrication mechanism.

The correlation between the grease lubricating life obtained from the model and the grease lubricating life evaluated using a rolling four-ball tester is shown in Fig.1. The variance explained was \(R^2 > 0.96\), indicating that the grease properties determine the grease lubricating life.

The value of the parameter \(a_i X_i^{a_i}\) indicates the degree of influence of each property on the grease lubricating life. The values of the parameters averaged over several greases are shown in Fig.2. Positive value of the parameter indicates a beneficial effect on longevity of greases, and the negative value means the detrimental effect, i.e. shortening longevity of greases. Based on these results, it is clear that the rate and amount of oil bleeding are two important factors to consider in grease formulation aimed at prolonging the life of lubricating greases.

4. Conclusions

The effects of grease properties on the grease life were clarified using multiple regression analysis. It was found that the grease lubricating life is determined by the grease properties and could be estimated from the property values. In order to improve the life of greases, especially the rate and amount of oil bleeding should be considered.

5. References

Chemical Reactions during Lubricated Tribological Contact between Diamond-Like Carbon and Steel Investigated by XPS

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1. Introduction

Recently, diamond-like carbon (DLC) coatings have received increasing attention, especially, in terms of interaction with lubricants and additives1. It is essential to understand the chemical reaction in detail under lubricated sliding condition for advanced application2.

In this study, we focused on investigating the chemical reaction during lubricated tribological contact between various DLC coatings and steel.

2. Experimental

The hydrogenated DLC coatings were deposited on polished steel substrates using plasma assisted chemical vapor deposition (PACVD) method. The coatings were varied in terms of hydrogen content, mechanical properties and surface morphology. The surface morphology and roughness were estimated by atomic force microscopy (AFM). The tribological behaviors of DLC-coated disks sliding against steel balls were examined under engine oil lubricated conditions (see Table 1). The worn surfaces of both disks and balls were analyzed by X-ray photoelectron spectroscopy (XPS).

Table 1 Ball-on-disk test condition

<table>
<thead>
<tr>
<th>Load</th>
<th>5 N</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hertzian contact pressure</td>
<td>1 GPa</td>
</tr>
<tr>
<td>Sliding speed</td>
<td>100 mm/s</td>
</tr>
<tr>
<td>Test time</td>
<td>3000 s</td>
</tr>
<tr>
<td>Temperature</td>
<td>R.T. (25 °C)</td>
</tr>
</tbody>
</table>

3. Results

The XPS results showed that molybdenum and sulfur, derived from lubricant and additives, were particularly observed on the steel balls. The amount of molybdenum and sulfur correlated with the tribological behavior.

Fig.1 shows that the coefficient of friction is generally reduced from approximately 0.09 to 0.04 with an increasing amount of Mo4+ on the steel balls. The correlation with the friction behavior was shifted depending on the combination between the hydrogen content and surface roughness of the DLC. The coatings with low hydrogen content tend to generate low friction.

Fig.2 shows that the S2-/Mo4+ ratio continuously decreases with increasing the surface roughness of the DLC, eventually reaching 2.0, except the coatings with more than 40 at.% hydrogen content. It is reasonable to assume that DLC coating properties influence the tribo-chemical reactions leading to molybdenum sulfide formation involving carbon transfer on steel.

4. Conclusions

We suggested that the hydrogen content and mechanical properties as well as the surface microstructure of DLC coatings would have great influence on the effects of lubricant additives, either positive or negative. It would be useful for controlling the chemical reaction and the friction behavior under lubricated sliding condition.

5. References

The Effect of Hardness Ratio on Friction: Role of Surface Texture

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2Department of Materials Engineering, Indian Institute of Science, Bangalore, Karnataka 560 012, India
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1. Introduction

Surface texture is one of the most important factors that control friction during sliding. Attempts have been made to study the role of surface texture on friction1, 2. Efforts have also been made to study the influence of hardness of materials on friction3. However, the effect of hardness as a function of surface texture on friction has not been well studied. Thus, in the present investigation, the effect of hardness of soft materials as a function of surface texture of hard material on friction during sliding is ascertained.

2. Experimental

In the present investigation, various kinds of textures (undirectional, 8-ground, and random) were attained on a set of steel plate surfaces. The roughness of the textures was varied using different grits of emery papers or polishing powders. The surface textures of steel plates were characterized in terms of roughness parameters using an optical profilometer. Soft pins made of various materials (Al, Mg, Pb, Cu, Sn, Zn, Al-4Mg alloy, Al-8Mg alloy, Mg-8Al alloy) were then slid against hard steel plates of various surface textures and roughness using a pin-on-plate sliding apparatus. Tests were conducted under both dry and lubricated conditions at a velocity of 2 mm/s in ambient environment. The pins were slid both in perpendicular and parallel direction to the unidirectional grinding marks on the plate. Thus, four sets of topographic conditions were used for a given material. Normal loads increased up to 120 N during the test. SEM was used to study the pin damage and morphology of the transfer layer formation on the plates.

3. Results

It was observed that the coefficient of friction did not vary significantly as a function of normal load up to 120 N during the test. Figure 1 shows the variation of average coefficient of friction with hardness ratio for various surface textures under lubricated conditions. In the figure, UPD and UPL respectively represent sliding direction perpendicular and parallel to the unidirectional grinding marks. Each symbol on Fig. 1 refers to the average coefficient of friction of five roughness of the same texture. It was observed that the range of surface roughness, Rq, varied between 0.02 and 0.8 µm for different textured surfaces. For a given texture, the average coefficient of friction did not substantially vary over this range of roughness. Under both dry and lubricated conditions, the friction was highest for the UPD case, followed by the 8-ground, UPL case, and was the least for the randomly polished surfaces. It was seen that the coefficient of friction values are much higher under dry conditions when compared to the lubricated conditions. For a given material pair, it was observed that the transfer layer formation on the steel plate depends on coefficient of friction. It was also found that among the surface roughness parameters, the mean slope of the profile, \( \Delta z \), correlated best with the friction. It was noticed under both dry and lubricated conditions that the variation of coefficient of friction with hardness ratio depends on surface texture. Under lubricated condition as shown in Fig. 1, the coefficient of friction decreases with increasing hardness ratio for the UPD surfaces. The correlation between friction and hardness ratio is found to be less for 8-ground and UPL surfaces. However, the coefficient of friction does not vary with hardness ratio for the random surfaces. These variations could be attributed to the extent of plane strain conditions taking place at the asperity level during sliding. Thus, it can be deduced that the coefficient of friction is not necessarily lower for harder materials and hardness alone cannot be used as a criterion for predicting coefficient of friction.

4. Conclusions

It was observed that the variation of friction as a function of hardness ratio depends on surface texture of the harder mating surface.

5. References

Development of a Tailor-made Low Ash Lubricant for Heavy Duty Natural Gas Engines in Chinese Market

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1. Introduction

Lubricating oils used in natural gas engines have special formulations that differ from diesel and gasoline engine oil formulations.

Therefore, the development of new natural gas engine oil (NGEO) must consider the following factors: the level of additives in the oil, oil properties such as resistance to oxidation and nitration, type and control of deposit and reduction of valve train wear. Based on these requirements, we developed a low ash NGEO by selecting the proper additives and base oils. The performance of this newly developed NGEO was evaluated by different standard engine tests and OEM's engine tests.

2. Experiment

2.1 Development of natural gas engine oil

The properties and engine test results of 10W-40 NGEO developed are listed in Table 1. The low ash NGEO has been developed by selecting the proper base oil and additives. This oil can meet the engine requirements in their appearance, they need tailor-made lubricants because gas engines can expose the oil to severe nitration and oxidation conditions, and can be very sensitive to ash content and composition.

2.2 Engine tests with Euro III natural gas engines

Two common types of Chinese domestic Euro III natural gas engines were selected to conduct the lubricant qualification tests. These were: 4CT180 engine manufactured by Shanghai Diesel Engine Corporation Limited, and EQD230N-30 engine from Dongfeng Motor Corporation. The main parameters of both natural gas engines are given in Table 2.

3. Results

The new NGEO developed has passed the Cat.1M-PC and L-38 engine tests and meets the API CF specification.

After the 1000h engine test on the EQD230N-30 engine, the valve train, piston ring and liner, and the engine bearing were inspected and no abnormal wear was found. The piston was clear. Also, no abnormal wear was found on these critical components after the 500h engine test on the 4CT180 engine. The piston was clean but a little light carbon was found on the top land. During the OEM's engine tests, the lubricant used was analyzed and the results were all at the normal level.

4. Conclusion

It is envisaged that in China the natural gas will be replacing fuel oil, especially in buses, the power generation industry and in the urban use. Buses powered by natural gas fueled engines are becoming increasingly popular in big and smaller cities, where natural gas has become available in recent years. Natural gas engine numbers are expected to grow as the gas distribution infrastructure is built.

Although natural gas engines may resemble other engine types in their appearance, they need tailor-made lubricants because gas engines can expose the oil to severe nitration and oxidation conditions, and can be very sensitive to ash content and composition.

The low ash NGEO has been developed by selecting the proper base oil and additives. This oil can meet the natural gas engine requirements in the Chinese market, as evaluated by the API CF engine tests and typical OEM's engine tests.

5. References


Quantum Chemical Molecular Dynamics Simulation on Tribochemical Reaction Dynamics and Super-Low Friction Mechanism of Diamond-Like Carbon

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1. Introduction

Classical molecular dynamics simulation is frequently employed to investigate the tribological phenomena on atomic scale. In addition to the atomistic understanding of the tribological behaviors, recently the electronic-level clarification of the tribochemical reactions is strongly demanded. However, the classical molecular dynamics method cannot simulate the chemical reaction dynamics. Therefore, we developed a quantum chemical molecular dynamics simulator for the elucidation of the tribochemical reaction dynamics1.

Diamond-like carbon (DLC) has been expected as a promising material having super-low friction for reducing carbon dioxide emissions and saving energies. Here, it is experimentally shown that the electronic-level understanding of the tribochemical reactions of the DLC is necessary for fully explaining its super-low friction. Therefore, in the present study we applied the quantum chemical molecular dynamics simulator based on our original tight-binding theory to the investigation on the tribochemical reaction dynamics and the super-low friction mechanism of the DLC.

2. Theory

Our quantum chemical molecular dynamics simulator “Colors” was employed for the simulation on the tribochemical reaction dynamics of the hydrogen-terminated DLC. This simulator is based on our original tight-binding theory and can compute over 5,000 times faster compared to the first-principles molecular dynamics method.

3. Results

Figure 1 shows the tribochemical reaction dynamics of the hydrogen-terminated DLC under 1 GPa pressure condition. Our calculation results indicate that the hydrogen-hydrogen repulsion at the interface leads to the super-low friction of the DLC. Moreover, we observed the formation of H2 molecule at the friction interface after 1.82 ps by the tribochemical reaction of hydrogen atoms which terminate carbon atoms. The time profile of the friction coefficient of the hydrogen-terminated DLC is shown in Fig. 2. This figure shows that the super-low friction coefficient of 0.07 is realized after the formation of H2 molecule. Then we suggested that the formation of vapor phase at the solid-solid interface is the reason for the super-low friction of the DLC. Moreover, it is very interesting that the distance between the DLC substrates increased just after the formation of H2 molecule. The increment of the DLC substrates distance was found to be another important reason for the super-low friction of the DLC.

We also calculated the tribochemical reaction dynamics of the hydrogen-terminated DLC under the higher pressure condition. At that time, the creation of new C-C bonds at the friction interface was observed. Moreover, the formation of new C-C bonds increased the friction coefficient of the DLC. Therefore, we suggested that the avoidance of the C-C bond formation is key factor to realize the super-low friction of the DLC. Moreover, we also suggested that the experimental fluctuation of the friction coefficient of the DLC is due to the dynamic tribochemical reactions of the hydrogen and carbon atoms in the hydrogen-terminated DLC.

4. References

Friction Force of Locust *Locusta migratoria manilensis* (Orthoptera, Locustidae) on Slippery Zones Surface of Pitchers from Four *Nepenthes* Species

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1. Introduction

Carnivorous plants of the genus *Nepenthes* mostly grow in nutrient-poor habitats and have evolved special organs called pitchers, which efficiently capture, retain and digest predominantly insects being used as nitrogen and phosphorus source. It is accepted that the slippery zone in pitchers, which is covered by plenty of lunate cells and wax crystals, plays a crucial role in trapping and especially in restraining preys by means of reducing insect attachment. Though the mechanism underlying this phenomenon has attracted many scientific studies, little attention has been paid to the influence of the slippery zone surface from different *Nepenthes* species on the anti-attachment functions. The principle goal of our present study is to investigate whether the friction force of locust on the slippery zone surface from different *Nepenthes* species varies. Based on the surface morphologies and microstructures of slippery zones from different *Nepenthes* species, we try to explain the difference of locust friction force on the slippery zones. The results will presumably supply suitable theoretical foundations for biomimetic structure and function of slippery zone to design and manufacture slippery plates for trapping plague locusts or other agricultural pest.

2. Experimental

Locusts of similar in size (imagines: length=46.75 ±3.82mm, N=18; nymph: length=14.28±0.95mm, N=20) and weight imagines: mass = 1.53±0.39g, N=18; nymph: mass=0.56±0.12g, N=20) were selected. *N. alata*, *N. fusca*, *N. khasiana* and *N. gracillima* were acquired from nursery, and the reference surface (stainless steel plate) was acquired from market, roughness and stiffness of the reference surface is Ra=0.2 µm, 90 HRB.

Claws and pads were cut from locusts and air-dried for SEM (Hitachi S-3400N) examination. To measure friction force generated by the locust on slippery zones and reference surface, a force sensor (load cell force transducer, 1-PW4C3) was used. Following a similar method previously introduced in detail, the locust was attached to the force sensor (along the load direction) using a thin thread (about 10 cm long) fastened to the locust’s neck, and then was put on the substrates. The friction force we measured was the force generated when locust walking ahead desperately on the substrates. Small pieces were cut from slippery zones of pitchers and air-dried, then observed with SEM. Dimensions of the structures in slippery zones were acquired from the SEM images. Surface profiles of the slippery zones were examined on fresh and untreated slippery surfaces with scanning white-light interferometer (Zygo New View 5000). The height and roughness can be acquired from the obtained images.

3. Results

Values of the friction force generated by locusts on the slippery zones and the reference surface showed obvious difference (Fig. 1). Among the slippery surfaces of the four species, the friction force generated by locusts on *N. khasiana* exhibits the greatest values. However, it is obviously lower than that on stainless steel plates. On both the stainless steel plates and the slippery surfaces, the imagines produced a much higher friction force than the nymph.

Slippery zones of pitchers via restricting the mechanical interlock and adhesive attachment, which is generated by locust’s claws and pads respectively, to reduce the friction force. Specifically, the wax crystals and the lunate cells play a crucial role in providing the functions, and the size discrimination of surface architectures probably leads to the significant difference in reduction of the friction force.

4. Conclusions

Our results demonstrated that, compared with the reference surface, the slippery zone of *Nepenthes* species pitchers can decrease the locust friction force by means of restraining the mechanical interlock and adhesive attachment. The friction force of locusts on slippery surfaces from different *Nepenthes* species exhibited distinguishable difference. The difference probably resulted from the discriminations in the geometrical dimensions of lunate cells and waxy crystals, as well as surface roughness in the slippery zones.

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5. References
Behavior of Thermo-Reversible Gel-Lubricants under Impact Load

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1. Introduction

The tribological properties of a new unique thermo-reversible Gel-Lubricant (TR Gel-Lube) were investigated. TR Gel-Lube, which includes 10-40% of amide type gelling agent in base fluid, is able to change repetitively from gel-state to liquid-state at the melting point of its gelling agent1). TR Gel-Lube has better tribological properties than the conventional grease2). In this study, the squeeze film forming ability of TR Gel-Lube is studied under the entrapped oil film behavior at halting EHL and the impact load by a falling bearing steel ball against flat anvil made of mild steel.

2. Test Lubricants

The base fluid P-N-0 used for this investigation is PAO (poly-α-olefin) of VG68. Density \( \rho \), kinematic viscosity \( \nu \), and the pressure-viscosity coefficient \( \alpha \) of P-N-0 are shown in the Table 1. The two TR Gel-Lubes used were P-A-10 and P-B-10. In P-A-10, 10% mono-amido is added as the gelling agent to the base fluid and in P-B-10, 10% bis-amido is added as the gelling agent. Li grease P-L-8 and urea grease P-U-12 with the same base fluid P-N-0 was used for comparison. Properties of TR Gel-Lube and greases are shown in Table 2.

Table 1 Properties of base fluid

<table>
<thead>
<tr>
<th>Oil</th>
<th>( \rho ), g/mL</th>
<th>( \nu ), mm²/s</th>
<th>( \alpha ), GPa</th>
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<tbody>
<tr>
<td>P-N-0</td>
<td>0.842</td>
<td>7.18</td>
<td>17.4</td>
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Table 2 Properties of TR Gel-Lube and greases

<table>
<thead>
<tr>
<th></th>
<th>P-A-10</th>
<th>P-B-10</th>
<th>P-L-8</th>
<th>P-U-12</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid</td>
<td>Gel</td>
<td>Gel</td>
<td>Grease</td>
<td>Grease</td>
</tr>
<tr>
<td>PAO</td>
<td>90</td>
<td>90</td>
<td>92</td>
<td>88</td>
</tr>
<tr>
<td>Mono-amido</td>
<td>10</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Bis-amido</td>
<td>10</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Stearic Li</td>
<td>-</td>
<td>-</td>
<td>8</td>
<td>-</td>
</tr>
<tr>
<td>Di-urea</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>12</td>
</tr>
<tr>
<td>Consistency</td>
<td>271</td>
<td>268</td>
<td>279</td>
<td>284</td>
</tr>
<tr>
<td>M.P, °C</td>
<td>100</td>
<td>150</td>
<td>100</td>
<td>150</td>
</tr>
</tbody>
</table>

3. Entrapped Oil Film Behavior at Halting EHL

This experiment was carried out under rolling speed of 0.26 m/s and mean pressure of 0.45 GPa using the optical interferometry technique3). Lubricant was applied to 1 mm thickness on the optical flat surface. First, it drove for one minute. Next, to obtain a shut down motion, the power supply of motor was cut off. The speed of the disc was ramped down linearly to halting. The subsequent pure squeeze motion stage with zero entraining velocity was investigated for a long time, i.e. for 3600s and 7200s. Figure 1 shows the change of entrapped films at halting EHL with times for each lubricant. The experimental entrapped film thickness at 0s of base fluid P-N-0 is 0.10 μm as shown in Fig.1. The base fluid P-N-0 is a liquid under this condition. Therefore, the entrapped film of 3600s and 7200s is perfectly squeezed out. The entrapped oil film thickness of grease P-L-8 and P-U-12 are also shown in Fig.1. The entrapped film remains in Hertzian contact region at 7200s. Base fluid P-N-0 is squeezed out from Hertzian region as shown in Fig.1 and this residual film is the grease thickener. Gel-Lube (P-A-10 and P-B-10) exhibits a marked difference in the feature of the entrapped film pattern compared to other lubricants; that the entrapped film pattern shows no change after 7200s. This confirms an entrapment of gel agents in the Hertzian contact.

Fig.1 Change of entrapped films at halting EHL

This result indicates that gelling agent adsorbs on the metal surface. Next, the impact test was conducted to see the action of the gelling agent under the high pressure generated by a falling bearing steel ball against flat anvil made of mild steel. These results also confirmed the above results.

4. References

1. Introduction

Recently, there is much research on the possibility of using natural fibres as reinforcement for polymeric composites. One of the most promising natural fibres is kenaf fibre, which has high interfacial adhesion with synthetic matrix, and great specific tensile modulus [1]. Few attempts have been made to study the effect of oil palm, coir, or betelnut fibres [2, 3], on tribological characteristics of polyester composites. In those works, the composites exhibited very poor wear performance, due to the poor interfacial adhesion of the fibres with the matrix leading to pulling out of the fibres during the sliding. In the current work, the effect of high interfacial adhesion of kenaf fibre on the abrasive wear performance of epoxy composite was studied under abrasive loading conditions. The worn surfaces were examined using SEM.

The kenaf fibres were chemically treated with 6% NaOH solution. The resin used in the current work is liquid epoxy (DER 331) and the curing agent is JOINTMINE 905-3S. The kenaf fibre reinforced epoxy (KFRE) composite was developed using closed mould technique. The volume fraction of the fibre in the matrix was about 48%. Surface of the developed KFRE composite is shown in Fig. 1. Three different orientations of fibres, with respect to the sliding direction of the counterface, were considered (Fig. 1). Block-On-Disk (BOD) machine was used for the current experiments, and the tests were performed by abrading the sample with an apparent area of contact of 11.5x11.5 mm² against three different grades (400, 1000, 1500, and 2000) of Silicon Carbide (SiC) abrasive paper under 5-25 N applied loads and rotational speed of 50 rpm corresponding to 0.157 m/s for test duration of 180 sec.

2. Results and discussion

Example of the results is presented in Fig. 2 showing the wear rate of the KFRE composite at different orientations compared with neat epoxy (NE). The figure indicates that kenaf fibre orientation has a very significant effect on the abrasive wear behavior of the epoxy composite. The lowest wear rate can be obtained when the fibres are oriented in N-O at all the SiC grades. Moreover, about 45% enhancement in wear performance of epoxy can be achieved by using kenaf fibres, in N-O, as reinforcement. Besides, increasing the SiC grade reduces the Wr of all the materials. This is due to the large size of the SiC at lower grade.

The micrographs of the composite, tested against 400G, show high interfacial adhesion of the kenaf fibres with the epoxy matrix, i.e. there is no sign of debonding. However, it seems that the fibre oriented in N-O is slightly damaged compared to the other orientations, especially P-O. When the fibres are oriented in N-O, the end of the fibres carries the load and protects the resinous matrix. On the other hand, the exposure of the whole fibres to the rubbing area causes high removal of materials, especially in P-O due to the fact that the SiC particles plough the resinous and fibrous regions individually. In the case of N-O, both regions are rubbed normal to the side force which allows the particles to transfer from one region to another and gives opportunity to the damaged particles to transfer from one region to another. This could be the reason for the lower Wr of the composite in N-O compared to P-O.

Fig.1 Sample of the specimen showing the orientations

![Fig.1 Sample of the specimen showing the orientations](image)

Fig.2 Wear rate with corresponding SEM images of the KFRP at different orientations

![Fig.2 Wear rate with corresponding SEM images of the KFRP at different orientations](image)

3. References

Low Friction Property of Lead-Free Aluminum Alloy Bearing Material With Molybdenum Disulfide Layer

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1. Introduction

A newly developed, lead-free aluminum alloy bearing is coated with low friction layer of molybdenum disulfide of sub-micrometre thicknesses on the plain bearing inner surface [1,2].

In this research, friction behavior of the newly developed bearing material was investigated using a reciprocating friction tester under lubricated condition.

2. Experiment

The test materials were the currently used lead-free aluminum alloy bearing and the newly developed lead-free aluminum alloy bearing with low friction MoS2 layer. A special shot peening process was developed for the new bearing[3] reducing the friction coefficient by the direct adherence of the MoS2 powder to the surface of a current lead-free Al-Sn-Si bearing alloy without using a resin binder. The mechanical properties of test materials used are shown in Table 1.

The material was rubbed against 6 mm diameter SUJ-2(JIS) bearing steel ball under load of 2N (Hertzian maximum contact pressure: about 560MPa) and at a sliding speed of 100 mm/min in oil. Before the friction test, the additive-free paraffinic base oil of 10 micro liters was dropped onto the sliding surface. The oil viscosity was 77 mm/s² at 20°C.

Table 1   Detail of test materials

<table>
<thead>
<tr>
<th></th>
<th>Current material</th>
<th>Developed material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Size (W×L×T), mm</td>
<td>17×18×1</td>
<td>17×18×1</td>
</tr>
<tr>
<td>Curvature radius, mm</td>
<td>26.5</td>
<td>26.5</td>
</tr>
<tr>
<td>Vickers hardness H1V</td>
<td>48</td>
<td>50</td>
</tr>
<tr>
<td>Ra, µm</td>
<td>0.12</td>
<td>0.13</td>
</tr>
</tbody>
</table>

3. Results and Discussion

The variation of the friction coefficient versus reciprocating cycles is shown in Figure 1. For the currently used material, the friction coefficient was about 0.18 at the early stage of the test and subsequently fluctuating from 0.14 to 0.20 regardless of the number of cycles. On the other hand, coefficient of friction for the developed material was very low of about 0.05 for more than 1600 cycles. After that, the friction coefficient increased rapidly to about 0.16. Thus, the friction coefficient was reduced by 60% compared to the currently used bearing material.

SEM images and EDX maps mappings of the worn surfaces of the currently used and newly developed materials after 1200 cycles are shown in Figure 2. The evidence of adhesion of Al on the counterface and large scratches in the sliding direction were found on surfaces of currently used material. On the other hand, for the newly developed material, the worn surface was smooth. Transfer layer of Mo and a little adhesion of Al were observed for the counterface. As a result, MoS2 layer prevents adhesion of Al on the newly developed material. However, when the area of Mo (MoS2)-transfer layer was larger than that of Al-transfer layer, the coefficient of friction decreased.

4. Conclusions

The newly developed bearing material gives about 60% reduction in friction coefficient compared to the currently used bearing material under reciprocating test conditions.

5. References

The Effect of Stearic Acid as Lubricant in Cold Forward Extrusion Process

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1. Introduction
Due to the increasing concern about environmental and health, development and application of green technology have become very important. Vegetable oil with high stearic acid content is considered to be a substitute for conventional mineral oil based lubricating oils, because they are biodegradable and non-toxic. Besides that, they have better intrinsic boundary lubricant properties due to the presence of long chain fatty acids on their composition 1, 2.

A series of cold forward extrusion experiments, using paraffinic mineral oil VG460 and RBD palm stearin as test lubricant, were carried out. The experimental and analytical results obtained were compared. We confirmed that the lubrication performance of RBD palm stearin is as effective as paraffin mineral oil in its ability to reduce friction forces in a cold metal forming.

2. Experimental Procedure
The experiments were conducted using plane strain extrusion apparatus. The apparatus consists of container, workpiece (billet) and 45-degree die half angle taper die. The taper die is made of tool steel SKD11, hardened and tempered. Surface roughness of taper die and container wall is 0.1µm. The extrusion ratio is 3. 5mg of lubricant was applied on die surface before the experiment. The other surfaces of experimental apparatus were lubricated with same type of test lubricant. Billet was aluminum alloy AA5083. Test lubricants are RBD palm stearin (abbreviated as PS) and paraffinic mineral oil VG460 (abbreviated as P3).

The plane strain extrusion apparatus was assembled and placed on the press. The experiments were conducted at room temperature while load and displacement data were recorded by computer. Extrusion was stopped at piston stroke of 40 mm. After that, extruded billet were taken out for the surface roughness measurement and metal flow analysis 3.

3. Results and Discussion
3.1 Extrusion Load and Surface Roughness
From the experiments, the steady state extrusion condition for both lubricants occurred at piston stroke of 15 mm. The steady state extrusion load applying RBD palm stearin as test lubricant is 97 kN which is lower than extrusion load applying paraffinic mineral oil VG460 as test lubricant which had 101 kN. The presence of stearic acids in the palm oil reduces the effect of friction constraint 4. Due to this, the extrusion speed for RBD palm stearin and paraffinic mineral oil VG460 is 10.3 mm/s and 8.7 mm/s respectively. The average value of the arithmetic mean surface roughness, \( Ra \), at the extruded area (product area) for billet extruded with RBD palm stearin is almost the same as with the paraffinic mineral oil VG460, which is 0.50 to 0.54 micron.

3.2 Velocity Distribution
After the experiments, billets were taken out and the flowlines of billet were traced into digital data. By using visioplasticity method, the \( v \)-component in deformation area of billet was calculated.

The \( v \)-component of the velocity relative to the ram speed is shown in Fig. 1. From figure, at distance between 1 to 4 mm, the velocity of billet extruded with RBD palm stearin is higher compared to the billet velocity extruded with paraffinic mineral oil VG460.

4. Conclusion
The investigation of stearic acid, used as a lubricant in cold forward extrusion process, was carried out on plane strain extrusion apparatus. The results show that stearic acid has the advantage in reducing forming load and increasing the sliding velocity.

5. References

Fig.1 \( v \)-component of velocity distribution.
Wear Properties of Submicrocrystalline Pure Fe Produced by High-Pressure Torsion Straining

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1. Introduction

Bulk nano/submicron-structured materials produced by severe plastic deformation (SPD), such as high-pressure torsion (HPT), equal-channel angular pressing (ECAP) and accumulative roll bonding (ARB), have been drawing strong attention owing to the extraordinary high strength and ductility of the SPD-processed materials. Among the SPD techniques, HPT process is one of the most powerful techniques to prepare ultrafine-grained materials due to non-homogeneous deformation with large strain and large strain gradient\textsuperscript{1). In the present study, wear properties of the submicrocrystalline pure Fe produced by the HPT process were investigated in detail.

2. Experimental

The material used was pure iron (11C, <30Si, <30Mn, <20P, <3S, <2B, 8N, 14O, 300Al, <20Ti, <30Cr, <30Cu, mass ppm) which was annealed at 1273 K for 3.6 ks in pure argon atmosphere. The pure iron was cut and polished into discs of φ 10 mm in diameter × t 0.85 mm in thickness for HPT-straining. As shown in Fig.1, the disc was placed between two anvils and torsion-strained at a rotation speed of 0.2 rpm under a pressure of 5 GPa at room temperature. The number of turns (N) was changed in the range from 1/4 to 10. Microstructure and Vickers hardness were examined for the HPT-processed discs.

Wear properties of the HPT-processed discs were studied using a ball-on-disc friction method. All wear tests were carried out in air at room temperature with a load of 39.6 N and a friction speed of 0.05 m/s for 10 minutes. The rubbing track had a diameter of 6 mm. The ball materials used were quenched and tempered high carbon-chromium bearing steel (SUJ2) and cemented carbide (WC-Co).

3. Results and discussion

Figure 2(a) shows the optical microstructure of the annealed disc, and Fig.2(b) is the FE-SEM/EBSP image of the HPT-processed (N = 5) disc at the region of r (distance from the disc center) = 3 mm. It can be seen that the grains were refined to less than 0.5 μm by the HPT-straining.

Figure 3 shows the effects of number of turns in HPT-straining on Vickers hardness at r = 3 mm and wear depth of discs. It was found that in the case of bearing steel (SUJ2) ball, wear depth of the discs increased with the number of turns in the HPT process despite the high hardness of the HPT processed discs. This is probably due to the high adhesion force between the submicrocrystalline pure Fe and the bearing steel ball. On the other hand, in the case of WC-Co ball, the wear of the discs was relatively small and decreased with the number of turns in the HPT process.

4. Conclusions

Wear behavior of submicrocrystalline pure Fe produced by HPT straining depended on the counter materials.

5. Reference

Influence of Surface Topography on Speed-dependent Boundary Friction Characteristics of Steel

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1. Introduction

Regarding the effects of surface roughness and topography on tribological characteristics, although much interest has been attracted on the elastohydrodynamic and so-called mixed lubrication regions, study regarding their effect on boundary lubrication is very limited. The previous study by the authors reported that the surface roughness orientation affected the friction under ultra-slow speed region where hydrodynamic action of lubricating oil is believed ineffective1. The findings suggested that the surface topography such as roughness and orientation might affect formation of tribofilm and eventually its friction reduction behaviour. In this study, friction characteristics of surface-roughened steel were investigated under lubrication with zinc dialkyldithiophosphate (ZnDTP)-formulated oil, focusing on the difference of friction reduction effect among different hydrocarbon moieties of ZnDTP.

2. Experimental procedure

Simultaneous friction and electrical contact resistance measurements were carried out using the self-produced tribometer2. The disk was attached to a pulse-controlled servomotor, and the cylinder was fixed in a sample oil cup. After the sample lubricating oil in the cup was heated to the test temperature, a dead weight was applied to the rotating disk against the cylinder. Firstly, a running-in was carried out to promote the formation of the tribofilm from the lubricating additives. Then, the friction measurements were carried out. For the measurement, the sliding speeds were changed stepwise, ranged from 8.5 cm/s to 5.0 μm/s. Table 1 summarizes the experimental conditions.

Two types of surface finish were applied to the disk specimens. One is isotropic smooth surface produced by polishing (ISO-PO) with a roughness of approximately Ra ≈ 0.08 μm. Another is isotropic rough surface produced by shot-peening (ISO-SP) with a roughness of approximately Ra ≈ 0.30 μm. Polyalphaolefin (PAO), was used as the base oil in this experiment. Three types of ZnDTPs (sec-C6, prim-C8, prim-C12) were used in a phosphorus concentrations of 0.1 mass%, respectively.

3. Results and discussion

Figure 1 shows the friction characteristics obtained from both ISO-SP and ISO-PO specimens under lubrication with three different kinds of ZnDTP formulated oils. Compared with the results in case of ISO-PO, both significant reduction in friction coefficient and increase in separation ratio were observed in case of ISO-SP. It was proposed that the separation ratio had a value of 1 if a nonconductive film was formed from ZnDTP on the friction surface. This suggests that the surface roughening resulted in fully formation of ZnDTP tribofilms that brought the reduction in friction. It can be also found that ZnDTP with longer chained hydrocarbon provided lower friction under lower speed region below 10⁻³ m/s in case of ISO-SP. It is supposed the hydrocarbon moieties of primary straight-alkyl ZnDTP exhibited similar load-carrying function to the fatty acid.

4. References

Finite Element Analysis on the Adhesive Contact between an Elastic Sphere and a Rigid Half-Space

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1. Introduction

The adhesive contact between an elastic sphere and a rigid half-space has been a topic of interest for a long time. The JKR [1] and the DMT [2] models were proposed in the 1970s. In 1997, Greenwood [3] proposed a numerical simulation. In 2006, Wu [4] proposed another numerical simulation for the adhesive contact in nano-scale. Finite element analysis is another approach used in this area. Cho and Park [5], Saur and Li [6] used finite element method to find the adhesive contact. However, these analyses were performed only for small Tabor parameters, and they were not compared with those obtained using the numerical simulation.

In this paper, a new finite element model is proposed. The adhesive force is derived from the Lennard-Jones force law. The results of FEM analyses are compared with those obtained using the numerical simulation.

2. Analyses

The Lennard-Jones potential law can describe the potential between two molecules. By integrating the Lennard-Jones potential over all molecules of two bodies, the traction \( p(h) \) of the deformed sphere surface due to the half-space is [2]

\[
p(h) = \frac{8\Delta\gamma}{3\varepsilon} \left[ \left( \frac{\varepsilon}{h} \right)^3 - \left( \frac{\varepsilon}{h} \right)^9 \right]
\]

where \( h \) is the gap between two surfaces, \( \varepsilon \) is the equilibrium distance between them and \( \Delta\gamma \) is the surface energy.

The radius of the sphere is \( R \). The approach between the sphere and the half-space is \( \alpha \). The non-dimensional total force is

\[
W = \frac{1}{R\Delta\gamma} \int p(h) dh
\]

The adhesive contact is governed by the Tabor parameter [7]

\[
\mu = \left( \frac{R\Delta\gamma^2}{E^*\varepsilon^*} \right)^{1/3}
\]

where \( E^* \) is the equivalent Young's modulus,

\[
1 - \frac{1}{E^*} = \frac{1 - \nu^2_1}{E_1} + \frac{1 - \nu^2_2}{E_2}
\]

3. Finite element model

Analyses are conducted with the commercial finite element software ABAQUS v. 6.9.

The sphere is divided into three regions. Each region is meshed using different seeds. The region near the half-space is meshed using small elements. The adhesive force is applied by the user subroutine UTRACLOAD. A quasi-static increment technique with large deformation geometry is used. In the analyses, the quasi-Newton and the line-search methods are used.

Analyses are conducted with different \( R \)’s (from 100 to 1000) and different Tabor parameters.

The results obtained using the finite element analyses are compared with those obtained by using Greenwood’s and Wu’s methods [3,4]. The difference for the load-approach curves between the finite element analyses and the numerical simulation is large for small sphere and large Tabor parameters. Part of the results is shown in Figure 1.

![Figure 1](image)

Fig.1 Load vs. approach for the adhesive contact for Greenwood’s method and finite element method.

4. Conclusion

A finite element model is proposed. With the proposed model, the adhesive contact can be obtained for large Tabor parameters. The results obtained were compared with the results of Greenwood’s [3] and Wu’s [4] numerical simulations.

5. References

Wear Of SiC-TiC-TiB₂ Ceramics At High Temperatures And Sliding Speeds In Air

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1. Introduction

In previous papers it has been shown that tribo-oxidation of non-oxide ceramics as e.g. SiC-TiC-TiB₂ may be used to form soft interfaces in-situ leading to mild wear under oscillating sliding conditions [1]. The reason for this is the formation of a mixed-oxide sliding interface with relatively lower shear strength. The properties of the interface are largely dependent on composition [2]. In this paper the results are shown confirming that the previous findings are valid for sliding speeds up to 4 m/s.

2. Experimental

SiC-TiC-TiB₂ 3 phase ceramics with composition of 50-25-25 mol % has been investigated under continuous dry sliding conditions against sintered silicon carbide (SSiC) up to 800°C in air at sliding velocities of 0.1 to 4 m/s in a pin on disk arrangement. An extended long distance test was carried out at 800°C in air at a sliding velocity of 4 m/s over a running distance of 92 km.

3. Results

The main wear mechanism is tribo-oxidation due to the thermo-dynamically unstable non-oxide phases. Thermodynamic calculations have been carried out to verify possible oxidation mechanisms.

The experimental results reveal that the amount of glassy phase is high enough to form a continuous oxide layer at 800°C leading to a low wear coefficient of less than 1*10⁻⁶mm³/Nm over a sliding distance of 92 km. Oxidation at high temperatures is hindered largely by the dense oxide layer most probably due to the low diffusion coefficient of oxygen in the siliceous glassy layer. However, the formation of the oxide film is dependant on the temperature. At low and intermediate temperatures of about 500°C sliding velocity is of crucial importance for the amount of oxide formed for film formation and wear rates are significantly dependant on sliding speed. This can be seen in the dependence of the wear coefficient on ambient temperature as well as on sliding speed as shown in Fig.1.

Fig.1 Coefficient of wear versus (a) ambient temperature and (b) sliding speed

The analysis of the continuous glassy layer formed during long distance sliding revealed that Ti and B are resolved in the glassy SiO₂ matrix at high temperatures. During cooling to room temperature rutile and different Magneli phase nano crystals like Ti₂O₇ are precipitated from the silica matrix, rutile nearer to the surface and the Magneli phases nearer to the material bulk. This is in accordance with thermochemical model calculations of amount of substance revealing that during the oxidation process at 1 bar oxygen partial pressure first SiC is oxidized, then TiB₂ and eventually TiC. The calculations also reveal that during the oxidation of the carbides carbon is generated and is stable during intermediate oxygen partial pressures, especially when embedded in a siliceous matrix, see Fig. 2. This explains the experimental evidence of graphite found in the wear scar by micro Raman analysis. As a consequence, these findings point at reducing conditions in the deeper zone of the sliding interface although all experiments have been carried out in air.

References

The Impact of Ethanol on the Tribological Performance and Degradation of Automotive Engine Lubricants

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1. Introduction

The reciprocating internal combustion engine is the prime mover of automobiles around the world. Thus optimising its operation in terms of durability, fuel economy, fuel economy retention and emissions is of significant importance. Some 7.4% of the fuel energy is lost due to engine friction of which the ring pack contributes 25%. As a whole, the piston assembly (piston and ring pack) accounts for 40-60% of total engine friction. Therefore, investigating this region of the engine is a necessity.

Increasing environmental concerns, legislation and customer demands have stipulated the use of alternative renewable fuels such as biofuels. Ethanol, a well established gasoline substitute, is being widely used both as a neat fuel and blended with gasoline. Thus, there is a need to investigate the effects of ethanol on lubricant degradation, piston assembly friction and ultimately fuel economy and its retention in the overall context of engine friction power loss and wear.

A bench-top screening test programme is reported that investigated the effect of ethanol on a fully formulated lubricant in terms of friction behaviour under simulated piston ring/cylinder wall conditions.

2. Experimental

A test matrix was developed using the Taguchi statistical experimental design method. Test parameters were selected to simulate cold-start/warm-up/short-trip engine operating conditions as ethanol accumulation in the sump is significant under such conditions. The tests were conducted using a Plint TE77 reciprocating tribometer modified to lubricate the contact from two sumps via in-situ lubricant feed changeovers as shown below;

Table 1 Control factors and their levels

<table>
<thead>
<tr>
<th>Control Factor</th>
<th>Symbol</th>
<th>Level 1</th>
<th>Level 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ethanol</td>
<td>A</td>
<td>5wt%</td>
<td>10wt%</td>
</tr>
<tr>
<td>Water</td>
<td>B</td>
<td>8wt%</td>
<td>16wt%</td>
</tr>
<tr>
<td>Temperature</td>
<td>C</td>
<td>25°C</td>
<td>40°C</td>
</tr>
<tr>
<td>Load</td>
<td>D</td>
<td>100N</td>
<td>150N</td>
</tr>
<tr>
<td>Speed</td>
<td>E</td>
<td>25Hz</td>
<td>33.3Hz</td>
</tr>
</tbody>
</table>

3. Results

Figure 2 shows the changes in friction obtained when the contact was lubricated with the mixture as opposed to when lubricated with the SAE 5W30 reference lubricant with no friction modifier, conforming to the API/SL/CF/A3/B4-08 specifications.

Fig.2 Comparison of reduction in friction coefficient

4. Conclusions

- Lubricant/ethanol/water mixtures exhibited varying levels of friction reduction compared to the lubricant alone.
- The ratio of ethanol to water in the mixture impacted on the level of friction reduction.
- Total percentage contamination by ethanol and water also affected friction reduction.

5. References

Tribological Properties of Fe-Mo Alloy Coating Prepared by Low-Pressure Plasma Spraying

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1. Introduction

It was found in our previous study that Fe7Mo6-based alloy specimens prepared by spark plasma sintering1 exhibited much lower friction coefficients and much lower wear rates than gray cast iron specimens in poly-alpha-olefin (PAO) containing no additive2. Therefore, the Fe-Mo alloys such as Fe7Mo6-based alloy are expected to be the candidates for low friction and high wear resistant coating materials. However, the tribological properties of the Fe-Mo alloy coating have not been investigated in detail. In this study, we prepared the Fe-Mo alloy-coated steel specimens by low-pressure plasma spraying3, and we examined their tribological properties in the PAO.

2. Experimental Procedures

In this study, the Fe7Mo6-based alloy powder with a composition of Fe-42at% Mo and with the particle sizes of less than 45 μm were low-pressure plasma-sprayed onto several SS400 steel substrates at the arc currents of 100 A to 750 A in 13.3 kPa Ar gas. After the plasma spraying, the microstructure, x-ray diffraction pattern and micro-Vickers hardness of each coated specimen were examined using a scanning electron microscopy (SEM), x-ray diffractometer (XRD) and micro-Vickers hardness tester, respectively.

The friction and wear properties of each coated specimen sliding against an ASTM 52100 steel ball in the PAO containing no additive were investigated using a ball-on-disk tribometer. Before the friction and wear tests, each coating specimen was polished using a diamond paste with a particle size of 1 μm. The ISO viscosity grade of the PAO was VG32, while the friction and wear tests were carried out at a sliding speed of 1.9 × 10⁻² m/s with an applied load of 4.9 N at 292 K in air with a humidity of 40 percent.

3. Results

SEM analyses revealed that all of the coating layers had homogeneous microstructure, and firmly adhered to the substrates. XRD analyses showed that the coating layers obtained at an arc current of 750 A mainly consisted of metastable BCC phase, while the coating layers obtained at arc currents of 100 A, 200 A and 400 A were mainly composed of FeMo (σ) phase. It is considered that this difference in the constituent phases was caused by the difference in the temperature of the powder particles during the plasma spraying. Figure 1 shows the friction behavior of the coated steel specimens. The metastable BCC-coated and FeMo-coated specimens prepared by plasma spraying exhibited friction coefficients as low as the Fe7Mo6-based alloy specimens prepared by spark plasma sintering. Also, the specific wear rates of the metastable BCC-coated and FeMo-coated specimens were as low as those of the Fe7Mo6-based alloy specimens.

![Fig.1 Average friction coefficients of the coated specimens and the Fe7Mo6-based alloy specimens.](image-url)

4. Conclusions

In this study, Fe7Mo6-based alloy powder was low-pressure plasma sprayed onto SS400 steel substrates, and their microstructure, hardness and tribological properties were examined. The conclusions are as follows.

1. Metastable BCC-coated and FeMo-coated steel specimens were obtained by low-pressure plasma spraying of the Fe7Mo6-based alloy powder.
2. The metastable BCC-coated and FeMo-coated specimens exhibited friction coefficients as low as the Fe7Mo6-based alloy specimens prepared by spark plasma sintering.

5. References

Experiments on Burridge-Knopoff Model Using Sandpaper Covered Surfaces and Granular Material Covered Surfaces

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1. Introduction

Burridge-Knopoff model is well known as a simplified earthquake model. In this model many blocks connected to each other with springs are placed on the moving belt conveyor with constant velocity, and they are also connected to the rest ceiling with driving springs. The stick-slip motions between the blocks and the belt conveyor are regarded as earthquakes. We have conducted experiments on this model using two sets of contact surfaces of different type. One set consists of sandpaper covered surfaces and the other one of the granular material and acrylic resin covered surfaces. The objective of this research is to examine the detail behavior of blocks and the magnitude distributions of the slip events in each case.

2. Experimental devices and methods

We conducted the experiments under the conditions shown in Table 1. The symbol ‘S’ and ‘G’ designate ‘sandpaper covered surfaces’ and ‘granular material covered surfaces’ respectively. The values of the spring constant of coupling spring \(k_C\) and that of driving spring \(k_D\) were set up as the rightmost column in Table 1.

Table 1  Experimental conditions

<table>
<thead>
<tr>
<th>Dimen -sion</th>
<th>Number of blocks</th>
<th>Contact Surfaces</th>
<th>(k_C(N/m))</th>
<th>(k_D(N/m))</th>
</tr>
</thead>
<tbody>
<tr>
<td>1D</td>
<td>10</td>
<td>S</td>
<td>776/156</td>
<td></td>
</tr>
<tr>
<td></td>
<td>10</td>
<td>S</td>
<td>156/156</td>
<td></td>
</tr>
<tr>
<td></td>
<td>4\times4</td>
<td>G</td>
<td>153/57</td>
<td></td>
</tr>
<tr>
<td></td>
<td>4\times4</td>
<td>G</td>
<td>57/57</td>
<td></td>
</tr>
<tr>
<td></td>
<td>4\times4</td>
<td>S</td>
<td>153/57</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3\times3</td>
<td>G</td>
<td>153/57</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3\times3</td>
<td>G</td>
<td>57/57</td>
<td></td>
</tr>
</tbody>
</table>

The experimental device in the case of 3\times3 block arrangement with granular-covered materials is shown in Fig.1. The mass and the size of one block are 150g and 30mm \(\times\) 30mm \(\times\) 10mm. The belt conveyor velocity is 0.10mm/s. The bottom of each block was attached to the acrylic resin plate (180mm \(\times\) 180mm \(\times\) 1mm) and the surface of the conveyor was covered with PVC.

3. Results

The temporal evolution of the displacements of the three blocks in the rightmost column in the case of Fig.1 experiment is shown in Fig.2. We can see that the motion of each block is a random stick-slip motion and all blocks in the same column easily slip simultaneously.

A group of neighboring blocks in slip state at successive time steps is regarded as a cluster and the magnitude of the slip event is defined as the logarithm of the sum of the total displacement of all blocks contained in a cluster. The magnitude distribution in Fig.1 case is shown in Fig.3. The magnitude distributions were obtained in all cases and the detail features were examined.

Fig.2 Temporal evolutions of the displacements of blocks in the rightmost column.

Fig.3 Magnitude distribution in the case of Fig.1

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Identification Method of Oil Film Dynamic Coefficients of Fluid Bearing for HDD Spindle

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1. Introduction

Recently, the fluid film bearings for HDD spindles are used as the rotating shaft support elements for a lot of machinery such as mobile PCs and car navigation systems. However, compared to ball bearings, the fluid film bearings have low stiffness. When an external vibration is applied to the spindle, the rotating shaft has a possibility to come in contact with the bearing and it causes wear or seizure to the bearing surface. Therefore, it is extremely important to verify the dynamic characteristics of the fluid film bearings for spindle. However, verification from both theory and experiment when an external force is added to the spindle is rare and few. In this paper, the bearing vibration characteristics when the HDD spindle is oscillated are investigated theoretically and experimentally. And then the identification method of oil film coefficients of fluid bearing spindle is described.

2. Experimental test rig

Figure 1 shows the outline of the vibration test rig. This test rig consist of FFT analyzer, the vibration exciter, 2.5” HDD spindle motor, motor fixing jig and eddy current proximity sensors. In addition, the test rig is set on the stone surface plate to eliminate the influence of the outside vibrations. In this experiment, the axial displacements of the disk are measured when the spindle oscillates in the radial direction and from the axial displacement data the angular amplitude of the shaft inclination is obtained. Since the angular displacements mainly depend on the two journal bearings, we can obtain the oil film characteristics easier then by direct measurement of the radial displacements which are very small.

3. Results

Figure 2 shows the measurement results of the amplitude of the axial disk displacements $z_x$, $z_y$ under various rotational speeds of 4200 [rpm]. From this figure, it is found that the amplitude decreases with the increase in acceleration frequency in the $x$ direction and the $y$ direction. The tendency of the decrease of amplitude is attributed to the constant amplitude with the increase of frequency. It is confirmed that the measurement errors are very low and the measured data is in a relatively good agreement with theoretical data for a wide range of experimental conditions.

Figure 3 shows the experimental results of the spring coefficients with rotational speed. In this figure, the comparison of experimental data with the data obtained by solving the Reynolds equation is shown. It is found that the spring coefficient of upper journal bearing $k_{xx}$ increases with increasing rotational speed, and the measured and calculated results agree qualitatively and quantitatively. Consequently it was confirmed that enough data were obtained in the identification of spring coefficients.

![Fig. 1 Experimental test rig](image1.png)

![Fig. 2 Frequency response](image2.png)

![Fig. 3 Identification results of spring coefficients](image3.png)
Erosion/corrosion of \(\text{Si}_3\text{N}_4/\text{TiO}_2\) Nanocomposite Coatings Fabricated by Plasma Electrolytic Oxidation

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1. Introduction

Plasma electrolytic oxidation (PEO) is a successfully industrialized method for fabrication of anti-corrosive and wear-resistant coatings of some light alloys such as aluminum. However, this method did not produce appropriate coatings on titanium based layers in comparison with similar electrolytic methods [1, 2]. PEO is considered as environment friendly (chromate free) process. It is highly desirable to find ways of improving mechanical properties of titania based layers synthesized by this method. Therefore, in this study an attempt was made to fabricate nanocomposite layers with embedding \(\text{Si}_3\text{N}_4\) nanoparticles into the PEO formed layer on titanium substrates.

2. Experimental procedure

Disc shape samples of plasma electrolytic oxidized CP-Ti were used in these tests [3]. Erosion/corrosion tests were performed using EG&G rotating device electrode. The erosive/corrosive media contained zirconia hard powder (average size around 1 micron) in 3.5wt% sodium chloride solution in the form of suspension. Samples were rotated for different periods of time at 600 rpm in suspension (weight of solid/weight of liquid = 4:1). Potentials of -600, +600 mV (versus OCP) and the open circuit potential (OCP) were applied to the test samples for studying the effect of corrosion phenomena. The effects of rotation speed, applied potential, \(\text{Si}_3\text{N}_4\) nanoparticles concentration in layer and time of rotation were studied. The weight change for the coated samples was plotted every 15 minutes during 3 hours of rotating. Figure 1 shows the results obtained. Mass gain from corrosion product and mass loss due to erosion/corrosion tests were occurring simultaneously but overall mass loss of samples was detected. The linear surface profiles, obtained from a Surtronic 25 (Taylor-Habson) roughness profiler and its data were analyzed through Taly profile (gold package) after converting into ASCII instructions using the same software.

![Fig.1 Weight loss versus time of rotation for different applied potentials](image)

3. Results and discussion

Based on the slopes shown in figure 1, different trends were related to the characterized phases in the coating [3]. First, agglomerated zone (top surface of coating) was removed slightly and the removal rate was relatively slow. After that, a soft interlayer appeared and consequently a transition zone could be observed in section II. Material removal was high in this section (mainly consisting of soft titania and aluminum titinate). In the third zone, a barrier layer consisting of hard alpha alumina was detected which could support hard nanopowders in the nanocomposite for better erosion/corrosion resistance.

Based on the SEM images (and EDS analysis) of eroded surfaces (not shown here) it can be noticed that coating removal changes from partial to uniform removal from zone I to II and comes back to partial removal from zone II to III. Corrosion products with their different morphology and higher Ti content were also analyzed, but it seems that their effects were negligible. Coatings with agglomerated nanoparticles in the surface remained almost unchanged during the first stages of the tests. Diffusion of zirconia particles in softer zones was detected. Surface profiler showed wider cracks with low h/w ratio for inner layers of coating than top layers.

4. Conclusions

It was shown that the coatings with agglomerated nanoparticles in the surface remain approximately unchanged during the first stages of the tests but are weaker than uniformly distributed samples in the second stage of erosion/corrosion tests. Treated coatings show some diffusion of zirconia particles in softer zones with partial removal of the coatings in top layers of coating. Surface profiler shows wider cracks with low h/w ratio for inner layers of coating than top layers.

5. References


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1. Introduction

In order to clarify the mechanism of wear and establish the wear equation, it is necessary to elucidate the elementary process of wear. The mechanism of adhesive wear is explained by the model for the formation of a wear particle through a mutual transfer and growth process1). In the initial stages of this process, elemental debris of wear particles, which are called wear elements, are generated at a true area of contact. The wear elements with diameters of 10–30 nm have been observed by an atomic force microscopy (AFM) in our previous study2). In this study, the influencing factors for the generation of wear elements and the formation of transfer particles have been investigated.

2. Experimental

Friction and wear experiments were performed on the pin-on-block-type micro-sliding friction tester. By means of a piezoelectric actuator, the pin was slid once through a distance of 120 µm on a block. The shape of the pin was a hemisphere of diameter 4 mm. Seven pin materials (iron, cobalt, nickel, aluminum, lead, copper, and silver: a purity higher than 99.9 %) were slid on an iron block specimen. Experiments were performed under a normal load W between 0.05 and 1.0 N and at a sliding velocity v = 120 µm/s. The influence of lubrication was also examined by using a paraffin oil. After the experiments, the surface of the block at the center of the wear track was examined by AFM.

3. Results

Fig. 1 shows AFM images of the surface of the iron block before and after rubbing. In Fig. 1(a), before rubbing, no dirt-like particles can be observed. On the other hand, in Fig. 1(b), after rubbing, the formation of extremely fine wear elements on the friction surface can be clearly observed. Also, transfer particles, which are aggregates of wear elements, can be seen.

Figs. 2 and 3 show the data for the quantity of wear elements and transfer particles formed by rubbing seven materials on iron, respectively. It can be seen from these figures that the adhesion force and the mutual solubility of combined materials affect their quantities.

4. Conclusions

Differences in the normal load and the lubrication conditions affect the quantity of wear elements and transfer particles. Furthermore, the quantities of them differed depending on the combination of materials.

5. References


Fig. 1 AFM images of the friction surface of the iron block: (a) before rubbing; (b) after rubbing (Fe/Fe, dry, W = 0.1 N). The values indicate the height of the particle.

Fig. 2 Relationship between the quantity of wear elements generated by rubbing seven materials on iron and the adhesion force to iron (dry, W = 0.1 N).

Fig. 3 Relationship between the quantity of formed transfer particles and the mutual solubility of two rubbing materials (dry, W = 0.1 N).
Development of Pneumatic Servo Bearing Actuator with Multiple Bearing Pads for Ultraprecise Positioning in Long Stroke

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1. Introduction

Advances in various areas of precision mechanics have required ultraprecise positioning techniques with nanometre-scale accuracy to be developed. To meet such specific needs, many kinds of actuators capable of ultraprecise motions have been proposed and developed. This need for ultraprecise positioning systems continues to expand. Our laboratory developed a ‘pneumatic servo bearing actuator (PSBA)’ as a novel actuator based on pneumatic servo technology for ultraprecise positioning. The actuator consists primarily of an aerostatic thrust bearing and a pneumatic servo valve. However, the stroke of the developed actuator was rather short, less than 20 µm, because the bearing clearance directly corresponded to the displacement of the actuated spool. Therefore, to lengthen the actuator stroke, the PSBA with multiple thrust bearing pads arranged in a laminate structure was proposed and developed. In this study, we conducted experiments to investigate the basic performance and positioning performance by adding position feedback control of the laminate type PSBA with three thrust bearings.

2. Principle of Operation

A simple schematic of the laminate type PSBA is shown in Fig. 1. The actuated part is the output spool, as shown in the figure. Its position is set by balancing two forces; force $f_s$ generated in the aerostatic thrust bearing and acting to the left, and, force $f_b$ (= $a_h \cdot p_b$) a constant preload force that is acting to the right. The input air pressure, $p_c$, from the compressor to the bearing clearance is controlled by a servo valve. The air functions effectively like a fluid in the aerostatic bearing clearance, and then it passes into the atmosphere. Because aerostatic bearings have an innate bearing stiffness, a change in inlet pressure $p_c$ alters the bearing clearance, which directly corresponds to a change in the displacement of the actuated spool. The spool is capable of frictionless movement because it is supported by aerostatic journal bearings.

![Fig. 1 Schematic diagram of laminate type pneumatic servo bearing actuator (PSBA)](image)

3. Basic Performance of Laminate Type PSBA under Open-Loop Control

The step response of the laminate type PSBA under open-loop control with a 6-nm-high step was tested without noise filtering in the long stroke mode. The sampling frequency was 1 kHz. Although there were some small fluctuations, the actuator performed 6-nm-high clear steps. This shows that the developed actuator has a high potential for stable motion and ultraprecise positioning with nanometre-scale accuracy, even though the response was not very fast due to the compressibility of air.

4. Improved Performance of Laminate Type PSBA under Closed-Loop Control

To improve stiffness and to obtain faster response and positioning characteristics of the actuator, position feedback was added to the system.

The controllability of axial force of laminate type PSBA with the position feedback was examined first. The results are plotted in Fig. 2(a). The white circles in Fig. 2(a) show the axial force of the laminate type PSBA when the servo valve was kept fully open under constant air supply pressure into the valve (=0.3 MPa), and the black triangles show the axial force controlled by the position feedback for maximum stiffness. The zero position was 30 µm, even though the response was not very fast due to the compressibility of air.

![Fig. 2(a) Axial force of laminate type PSBA with position feedback (3-nm-high steps).](image)

(a)  

(b)  

5. Conclusion

The most suitable motion range of our new actuator shifted to the long stroke mode, with the result that the actuator was confirmed to be suitable for longer stroke motion. The minimum resolution of the new actuator was 3 nm, even in the longer stroke mode, by using a simple position feedback.
Structural Analysis of DLC Film under Hydrostatic High Pressure by Raman Spectroscopy and Synchrotron X-Ray Diffraction

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1. Introduction

Diamond-like carbon (DLC) film has been widely applied as modified surface to improve its tribological properties. It is a metastable, hard amorphous carbon containing a high proportion of carbon in sp³ hybrid orbital. Under tribological condition, surface is generally exposed to the high pressure. Though the structure of DLC film is predicted to be changed by high pressure, there are not any reports on investigations available on structural change of DLC film due to pressure. It is believed that the structural information of DLC film under high pressure will contribute to clarify a basic mechanism for low friction and provide a guideline to produce a better surface.

In this study, to gain the information on structural change of DLC film, Raman spectroscopy and synchrotron X-ray diffraction were applied to DLC film containing hydrogen under hydrostatic high pressure.

2. Experimental Methods and Results

2.1 Raman Spectroscopic Analysis

Structural change of DLC film under hydrostatic high pressure was analyzed using Raman spectroscopy. For measurement in Raman spectroscopy, a diamond anvil cell (DAC) was used in applying hydrostatic high pressure to the DLC film. An argon ion (Ar+) laser (514.5nm) as light source in a laser Raman spectrometer (T64000, Horiba) was used.

Fig.1 shows the wavenumber in G-band peak around 1555 cm⁻¹ under normal and hydrostatic high pressures.

2.2 Synchrotron X-Ray Diffraction

Structural change of DLC film under high hydrostatic pressure was analyzed using synchrotron X-ray diffraction. DLC film was pressurized by multi-anvil apparatus installed in PF-AR NESC in KEK. The scattered X-ray at 2θ-angles of 2, 3, 5, 6, 8, 10, 12, 14, 16, 18, 20, 22 and 25 degrees until about 3000 counts for each degree were detected.

Fig.2 shows peak values in pair distribution functions g(r) obtained from diffraction profiles under normal and hydrostatic high pressure.

3. Discussion

Fig.1 shows that the wavenumber in G-band peak has shifted from about 1550 cm⁻¹ to 1560 cm⁻¹ as hydrostatic pressure increased. It has been reported by A. C. Ferrari and J. Robertson that the wavenumber shift of G-band to be higher attributed to the increase in volume of sp³ sites.

Fig.2 shows that the peak values of pair distribution functions at r=1.54 Å decreased by 1.0%, while that at r=3.24 Å decreased by 2.5% as hydrostatic pressure increased. The peak at r=1.54 Å corresponds to sp³ and sp² orbitals, and that at r=3.24 Å corresponds to sp² orbital, respectively.

These results show that volume of sp³ sites increased and volume of sp² sites in DLC film decreased as hydrostatic pressure increased.

4. Conclusion

(1) Raman spectroscopy profiles of DLC film showed that the wavenumber in G-band peak of DLC film shifted higher as hydrostatic pressure increased.

(2) Peak values in pair distribution functions g(r) of DLC film, obtained from X-ray diffraction profiles, at r=1.54 Å decreased by 1.0%, while those at r=3.24 Å decreased by 2.5% as hydrostatic pressure increased.

(3) Comprehensive discussion based on the two kinds of analysis showed that the volume of sp³ sites increased and volume of sp² sites in DLC film decreased as hydrostatic pressure increased.

Reference

Study of Novel Clutch Utilizing Self-Locking Property of Belt

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1. Introduction

The belt friction equation is useful for designing belt drive or band brake. But the conventional belt friction equation does not consider an effect of belt overwrapping on belt friction. Imado has found the self-locking property of belt in the course of his study of a belt buckle1. Provided the condition was satisfied, the self-locking occurs even in the case of a belt wrapped two times on a single axis2. He proposed a novel clutch utilizing this self-locking property of belt3. The clutch was confirmed to work effectively even with the offset centered condition. This paper reports some experimental results together with the extended belt friction equation.

2. Theory

Mechanical model of a belt wrapped three times around an axis is shown in Fig.1. Relation of the belt tensions between the outer belt tension T1 and inner belt tension T2 is expressed by Eq. (1).

\[ T_1 = \frac{e^\alpha_{\theta_0} - T_2}{A - B} \]

where,

\[ A = \frac{\mu}{\mu_b - 1} \left( e^{\alpha_{\theta_0} - \theta_0} - e^{2\alpha_{\theta_0}} \right) - 1 \]

\[ B = \frac{(\mu/\mu_b - 1)\left( e^\alpha_{\theta_0} - 1 \right) \left[ 1 + e^{\alpha_{\theta_0}} - e^{2\alpha_{\theta_0}} \right] + e^{\alpha_{\theta_0}}}{e^{2\alpha_{\theta_0}} \left( 1 + e^{2\alpha_{\theta_0}} - e^{4\alpha_{\theta_0}} \right)} \]

The symbol \( \mu \) denotes the coefficient of friction between a belt and axis and \( \mu_b \) denotes the coefficient of friction between the belt and belt. Self-locking occurs when the denominator of Eq. (1) becomes negative i.e., \( A < B \). These conditions can be easily simulated and the self-locking was confirmed to occur by the experiment. Provided \( \mu = 0.25 \) and \( \theta_0 = 350° \), some calculations were made by using Eq.(1) and Eq.(2).

\[ T_1 = \frac{e^{\alpha_{\theta_0} - \theta_0}}{1 - e^{2\alpha_{\theta_0}}} \left( \frac{\mu}{\mu_b - 1} + e^{\alpha_{\theta_0} - \theta_0} \right) T_1 \]

Results are shown in Fig.2. With an increment of \( \kappa \), the larger wrap angle is required to enter a self-locking state.

3. Experiment

The test rig of the novel clutch is shown in Fig.3. MoS2 was used to get smaller coefficient of friction for \( \mu_b \). The coefficients of friction were evaluated in a pendulum test. According to Eq. (2), the minimum wrap angle required for self-locking is \( \theta_0 = 105° \). The self-locking occurred when \( \theta_0 > 105° \) and it never occurred when \( \theta_0 < 105° \).

4. Conclusion

Belt friction equation with an over-wrapped condition was derived. Belt-type one-way clutch was developed utilizing the self-locking property of belt.

5. References

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Friction and Wear Characteristics of Aluminum-Silicon Alloy Impregnated Graphite Composite under Drop-feed Lubricated Reciprocating Sliding Conditions

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1. Introduction

Tribological properties of porous graphite composite impregnated by molten Al-Si alloy (ALGR-MMC) with high content of graphite (56 vol%), reduced porosity (2%) and low density (2.34 g/cm3) are investigated. Previous pin-on-disk wear tests [1, 2] revealed that ALGR-MMC, due to its self-lubricating properties, was able to endure high loads under unlubricated and insufficiently lubricated conditions. Since the previous tests were conducted under unidirectional sliding, the results could not be generalized. Thus, in this study the wear tests are conducted under unlubricated and drop-feed lubricated reciprocating sliding conditions.

2. Testing procedures

Ball-on-disk type reciprocating wear tests were conducted in air with relative humidity of RH ≤ 1% and RH = 70% at 24°C under the following conditions: (a) under unlubricated sliding, (b) under drop-feed lubrication, where oil drops having the volume selected in the range of 0.0001 to 0.02 cm³ were deposited on the disk surface before the tests by using a micro-pipette, and (c) under immersion lubrication, where the ball-on-disk contact was submerged in a 30 cm³ oil (SAE30) bath. Disk specimens (35 mm in dia.) were made of ALGR-MMC. Ball specimens (12.7 mm in dia.) were made of JIS SUJ2 bearing steel. After polishing, the center line average roughness (Ra) of the disk and ball was 0.093 and 0.014 μm, respectively. Measured Vickers hardness of the ball and disk (Al-Si alloy part) was 855 and 90, respectively. Contact load (40 or 80 N) and before OFB exhibits low values equal to those under immersion lubrication. There is a transition region where OFB sometimes occurs or does not occur during a test. The oil amounts in the transition region are larger at 80 N than at 40 N.

3. Test results and discussion

At larger amounts of oil, an oil film breakdown (OFB) does not occur throughout a test but at smaller amounts of oil it does. Although a relationship between MKFC and amount of oil is not shown, MKFC without and before OFB exhibits low values equal to those under immersion lubrication. MKFC after OFB shows high values similar to those under unlubricated sliding.

Fig. 1 shows MWR as a function of the amount of oil dropped on the disk surface only before each test. MWR with OFB at 40 N and 80 N is higher than that without OFB and decreases with the increasing amount of oil. The longer duration of oil remaining within the contact area results in lower MWR. MWR without OFB shows low values similar to those under immersion lubrication. There is a transition region where OFB sometimes occurs or does not occur during a test. The oil amounts in the transition region are larger at 80 N than at 40 N.

4. Conclusion

ALGR-MMC tested in the present research is one of the best sliding materials that can be used even under insufficiently lubricated sliding conditions.

5. References

Observation of Meniscus Shape at Reciprocating Lip Seal Edge and Numerical Simulation Considering Liquid Surface Tension

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1. Introduction

Seal technology has been widely used for many years to prevent the working fluid from leaking out in a sliding unit. Despite the importance and wide use of various mechanical components, the design specifications of seals are still specified empirically, and few studies have examined the meniscus (air-liquid interface) shape at the seal edge from a scientific viewpoint. However, the behaviour of the meniscus shape in the seal considerably affects the sealing performance and thus requires further investigation. To clarify the meniscus behaviour in a seal, we focused on a lip seal in reciprocating motion and prepared a visualisation unit containing a lip seal component. We observed the change in the meniscus shape with microscope during the shaft’s reciprocating motion. At the same time, we conducted numerical simulations based on the Navier-Stokes equation on the flow in the tapered edge in the lip seal. The meniscus shape at the lip seal edge was predicted with considering the surface tension of liquid in the simulation.

2. Visualisation Unit

To directly observe the change in meniscus shape during axial motion, a visualisation unit with a lip seal was prepared. The lip and shaft were made of stainless steel; thus, no elastic deformation occurred in the seal part. We observed the change in the meniscus shape with a microscope. Fig. 1(a) shows a schematic diagram of the lip seal in the unit, and Figs. 1(b) and 1(c) are photographs of the meniscus of water as the working fluid.

![Fig. 1 Schematic view and photographs of lip seal part in visualisation unit](image)

Fig. 1 Schematic view and photographs of lip seal part in visualisation unit

In the experiment, the shaft speed for axial motion was 0, 1, 2 and 3 mm/s and the initial fluid level was 1.8, 2.5 and 3.2 mm in height. The photographs in Figs. 1(b) and (c) were taken at the shaft speeds of 0 and 2 mm/s, respectively, and an initial fluid level of 2.5 mm. A comparison of the photographs indicated that the meniscus shape changed due to the surface tension of the fluid in the axial motion of the shaft.

3. Numerical Simulation

We conducted a numerical simulation to measure the meniscus shape in the lip seal edge. In the simulation, a 2D Navier-Stokes equation was applied to the working fluid in the lip seal to obtain the pressure distribution in the fluid. The meniscus shape was calculated from the pressure distribution at the edge of the fluid taking into consideration the surface tension of the fluid.

The experimental photographs and analytical meniscus shapes are shown in Fig. 2, where (a), (b) and (c) were taken at respective shaft speeds of 1, 2 and 3 mm/s and at an initial fluid level of 2.5 mm. We can see that the analytical meniscus shape agreed well with the experimental one.

![Fig. 2 Meniscus shapes for various shaft speeds](image)

(a) 1 mm/s  (b) 2 mm/s  (c) 3 mm/s

Fig. 2 Meniscus shapes for various shaft speeds

The photographs show that the shaft side contact angle $\alpha$ decreased with an increase in the shaft speed. The analytical results of the flow field and the pressure distribution are shown in Fig. 3.

![Fig. 3 Flow field and pressure distribution](image)

(a) 1 mm/s  (b) 2 mm/s  (c) 3 mm/s

Fig. 3 Flow field and pressure distribution

In Fig. 3, arrows show velocity vectors, and colour distribution charts show the pressure distribution for various shaft speeds. From the analytical results shown in Fig. 3 it can be seen that velocity vectors and dimensionless pressure values increase as the shaft speed increases.

4. Conclusion

We clarified the behaviour of meniscus shape, flow field and pressure distribution in a lip seal using a visualisation unit and numerical simulation.
A New Method for Measuring Normal Forces with Accurate Gap Control Using A Microfabricated Quartz Resonator for Lubrication at Nanometer Gaps

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1. Introduction

Lubrication at small sliding gaps is important for micro/nano mechanical devices. In high-density hard disk drives, the gap between the head and disk is required to be less than 10 nm. This means that the head slides on a nm-thick lubricant film on the disk. However, the lubrication phenomena are unclear due to the difficulty of their measurement. Especially, measurement of normal force during sliding with accurate gap control is difficult because conventional force sensors use their deformation in the same direction as the normal force, which alters the gap.

In the previous study we presented the combination of a double-ended tuning fork (DETF) resonator with an optical fiber probe can provide simultaneous measurement of normal and lateral forces with accurate gap control1). The DETF resonator can provide highly sensitive force detection and accurate gap control because the oscillator is soft laterally but rigid normally. Although this electro-discharge machined resonator showed the feasibility, improvement of its sensitivity was difficult because its size was limited by the fabrication method. In this paper, we present a new measurement method that uses a microfabricated DETF quartz resonator.

2. Quartz resonator-based method for measuring normal force with accurate gap control

A schematic structure of DETF quartz resonator is shown in Fig. 1. Because the resonator is fabricated by micromachining techniques, a smaller resonator is possible. This can provide higher force sensitivity. The double-ended quartz oscillators vibrate in the anti phase. When a normal force is applied to the quartz oscillator, the resonant oscillators vibrate at the anti phase frequency shift due to this frequency shift, the applied normal force can be obtained. When the normal force is not large, the phase shift is proportional to the applied force. In addition, both ends of the oscillator are nodes of the oscillation. This can reduce the energy dissipation from the oscillator, which provides a low damping coefficient or high Q-value. This also can provide the oscillation that is not susceptible to attachment of additional parts to the end, for example, gluing an optical fiber probe shown in Fig. 1. The optical fiber probe is used as a probe for lateral force detection with an accurate gap control. The viscoelastic force exerted by lubricants confined at nm-gaps can be obtained by measuring the amplitude and phase shifts of the probe vibration. Thus, the quartz-based DETF resonator with the optical fiber probe can provide normal and vertical forces with accurate gap control.

3. Results and discussion

In the experiment, the support frame was glued to the end of the resonator in order to increase its lateral rigidity. The attachment of the support frame affected the resonator oscillation less as mentioned above. Using this optical-fiber-attached quartz resonator, the relationship between normal force and phase shift was measured. The normal force was applied by pushing the optical fiber probe at the resonator end with a double cantilever beam. The normal force was obtained from the deflection of the double cantilever. The measured results showed the phase shift is proportional to the applied normal force, which indicates the feasibility of the method.

4. Conclusion

The normal force measurement method with accurate gap control is expected to be useful for clarifying the tribological phenomena in micro/nano mechanical devices.

5. Acknowledgements

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6. Reference

Influence of Adding Substrate Bias Voltage on Adhesive Strength and Frictional Characteristic of Ti-doped Amorphous Carbon Film for Applying to the Artificial Hip Joint

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1. Introduction

At present, lifetime of a common artificial hip joint is as short as about 10-15 years. We attempted to improve the life time of artificial hip joint by coating amorphous carbon films on a ball of the artificial hip joint. Amorphous carbon films have attractive properties, such as high hardness and excellent biocompatibility. However, amorphous carbon films are limited to the large-scale use because of weak adhesive strength to substrates. In addition, the friction coefficients of amorphous carbon films are rather high for biomedical applications such as the artificial hip joint.

Our attention was focused on improving the adhesion and frictional characteristics of amorphous carbon films for applying to the artificial hip joint. We tried out doping Ti as an active element, addition of the substrate bias, creation of Ti interlayer, and heat treatment in vacuum.

2. Experiments

We prepared Ti-doped DLC films on silicon substrates by DC magnetron sputtering in Ar plasma. Table 1 shows film creating conditions. In creation process, kinetic energy of sputtered particles in the plasma was controlled by adding substrate bias voltage. In addition, we created Ti interlayer with Ti target. Furthermore, the amorphous carbon films were heated at 500˚C for 10 minutes after the creation.

The adhesive strengths of films were measured by the scratch test with a loaded diamond stylus. The friction coefficients of the films were evaluated by the ball on disk tribometer in an ambient atmosphere.

3. Results

Figure 1 shows effects of the substrate bias voltage on friction coefficients. Friction coefficients of Ti-doped amorphous carbon film and the film added -50V bias voltage against a stainless steel ball were around 0.25. Though, the friction coefficient of the Ti-doped amorphous carbon film added -100V substrate bias fell to around 0.1.

Figure 2 shows effects of the Ti interlayer on critical loads. The critical load of the Ti-doped amorphous carbon film was around 60mN. On the contrary, that of the film with Ti interlayer was around 150mN. However, the critical load decreased from 150mN to 110mN by adding the bias voltage of -100V.

4. Conclusion

Friction coefficients of amorphous carbon films were reduced by adding substrate bias voltage. Moreover, adhesive strengths of amorphous carbon films were increased by creating Ti interlayer. However, these results are not enough for applying to artificial hip joint. We have to improve adhesion and frictional characteristics further.

5. Reference

Generalized Reynolds Equation and Its Perturbation Equations to Determine the Dynamic Coefficients of the Fluid Dynamic Bearings with Curved Surface

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1. Introduction

Fluid dynamic bearings (FDBs) have been applied to the spindle motor of a computer hard disk drive (HDD) to support the rotating disk-spindle system due to their outstanding low noise and vibration characteristics. FDBs in a HDD have a unique structure comprised of coupled journal, thrust, spherical and conical bearings. Fig. 1 shows a mechanical structure of FDBs with journal and conical bearings. Many researchers [1, 2] have studied the static and dynamic characteristics of the coupled journal and thrust bearings. However, prior researchers did not derive the Reynolds equation in the general curved surface so that their method cannot be applied to the FDBs with various curved surface. This paper derives the Reynolds equation in the generalized curved surface and its perturbation equations to calculate the characteristics of the various shapes of the FDBs such as journal, thrust, conical, and spherical bearings. They are transformed to the finite element equation to calculate the pressure, load, and stiffness and damping coefficients of FDBs.

2. Method of Analysis

The FDBs with curved surface can be determined by the radius from the axis of rotation to the bearing surface \( r(s) \), the angular coordinate \( \theta \), and the meridian coordinate \( s \). The radius from the axis of rotation to the bearing surface \( r(s) \) determines the various shapes of FDBs.

The Reynolds equation in \( s, \theta \) plane can be derived by substituting the equation of flow rate into the continuity equation.

\[
\frac{1}{r(s) \partial s} \left( \frac{h^3}{12 \mu} \frac{\partial p}{\partial s} \right) + \frac{1}{r(s) \partial \theta} \left( \frac{h^3}{12 \mu} \frac{1}{r(s)} \frac{\partial p}{\partial \theta} \right) = \frac{V_{\theta}}{2 \pi r(s) \partial \theta} + \frac{\partial h}{\partial t}
\]  

(1)

Perturbation equations are derived by substituting into the Reynolds equation a first-order expansion of the film thickness and the pressure with respect to small displacements and velocities.

The FEM was used to solve the generalized Reynolds equations in (1), as well as the generalized perturbation equations. The global matrix equation of the finite element equation of the general Reynolds equations can be obtained to calculate the pressure of the FDBs with curved surface in quasi-equilibrium. Once the pressure in the fluid film is determined, the global matrix equation of the finite element equation corresponding to each perturbation equation can be determined. The Reynolds boundary condition was applied as the internal boundary condition. It guarantees the continuity of pressure and the pressure gradient across the cavitated area using an iterative method.

3. Results and Discussion

The proposed method was applied to the coupled conical and journal FDBs of the spindle motor of a HDD in Fig. 1. Fig. 2 shows the static and dynamic characteristics of the coupled conical and journal FDBs due to the change of the conical angle. Only the conical bearing of the coupled conical and journal bearing generates the axial load to support the rotor so that the axial load and the axial stiffness increases with the increase of conical angle. The film thickness of journal bearing is independent of the change of flying height, so only the film thickness of conical bearing changes due to flying height. Also the friction torque and the damping coefficients increase with the increase of conical angle.

4. References

Grit Particle Effect on Frictional Characteristics and Grit Embedment of Automotive Braking System

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1. Introduction

Automotive disc braking is often linked to the presence of hard particle and water derived from the environment [1]. Hard particles of different sizes and shapes together with other contaminants from outside can enter the brake gap and cause an increase or decrease in average friction force and momentary peak values of friction force at the braking interface. The presence of these external contaminants is unavoidable and can indirectly affect the braking effectiveness [2, 3].

2. Methodology

The effect of grit particle from environment on friction coefficients (CoF) and grit embedment (GE) was investigated using a vertically oriented brake test rig. Silica sand particles of the size ranging from 50-180 µm, 180-355 µm and 355-500 µm were used in drag mode brake application. The presence of external grit particle significantly influences the friction response at the pad/disc interface. Once the particles enter the gap between the brake pad and the disc, the value and amplitude of friction coefficient tend to change with the speed and load applied. Normal force is applied via solid steel cylinder to the brake pad using a lever arm and mechanical weight loading system. Full bridge strain gauges are used to record the instantaneous normal force and friction force at the brake interface. A grit particle feeder tube is attached to the hopper to direct the grit particles to the brake gap. A 1 h.p. three phase induction motor with a controller is used to rotate the disc. Fig. 1 shows the schematic diagram of the test rig.

3. Results

Experimental results showed that grit particle size has a significant effect on the CoF values. The values of CoF decrease when grit particles of size below 355 µm are present, but their values increase with smaller grit sizes. Smaller grit particles were found to mix and change the effective contact area more rapidly with some smaller particles filling the cavities on the pad specimen. The CoF values also increase with sliding speed and applied load. Smaller particles assisted in the formation and growth of more secondary contact plateaus, thus lowering the percentage of grit embedment (GE), as shown in bar graph in Fig 2.

4. Conclusions

In general, the CoF values are reduced when grit particles are present, however, with smaller grit particle size they increase. Smaller grit particles give lower percentage of GE. Both CoF and GE results are closely related to the change and growth of the effective contact area i.e. primary and secondary contact plateaus.

5. References

Study on the Nanoparticle-Wafer Surface Contact in CMP Process Using Finite Element Analysis

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1. Introduction

Chemical-mechanical polishing (CMP) for wafer planarization is an abrasive machining process where the wafer surface to be polished is pressed against a polishing pad that is covered with liquid slurry containing abrasive nanoparticles1. In the process, the critical and difficult issues are the unwanted defects and in-wafer non-uniformity of material removal2. In this work, efforts using finite element (FE) analysis were made to observe the effects of the process parameters on the surface defect generation and material removal3,4. Strong interest in this research arises from the fact that many uncertainties still exist in how the process works and what wafer-particle-pad interactions take place. In this paper, we will report the first computational observation on the dependence of the material removal and surface scratch on the material and shape of the particle.

2. Experimental details

A FE model comprised of single nanoparticle, wafer, and pad was made, in which slurry fluid in a few micrometer-sized fluidic channel between the wafer and pad asperities was considered (Fig.1). As for the particle material, silica, ceria and alumina were chosen and various materials such as Cu, W, and Si3N4 were modeled as the wafer material. Impacts on the wafer surface with the particle were simulated under the slurry flow in a micrometer-scale pad-surface gap as well as the direct contact between wafer, pad and particle. The FE simulations were done under various conditions such as the changes in the shape (trapezoidal, rectangular, spherical, and hexagonal), and size of particle, wafer moving speed, pressure and slurry viscosity.

3. Results and discussion

Fig. 2 is an analysis result of the contact between ceria particle and Al surface in slurry flow. The result reveals that the stress of about 5 GPa is generated by the particle impact on Al surface and thus the plastic deformation and subsequent material removal occur. This result was obtained under the situation that the particle collided with the surface at the speed of about 60 m/s.

4. Conclusions

In this paper, the relationship between the material removal rate and the interfacial mechanical properties at particle-surface contact situation, which can be seen in an abrasive machining process using micro/nano-sized particles like chemical mechanical polishing (CMP), was discussed by using FE analysis. The FE analysis results according to various process parameters were presented and discussed together with the related experimental data.

5. References

Effect of surface hydrophilicity on the nanofretting behavior of Si(100) in atmosphere and vacuum

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1. Introduction

Because of the temperature variation and mechanical vibration, nanofretting of monocrystalline silicon potentially exists in the contact interfaces of micro/nanoelectromechanical systems (MEMS/NEMS). Therefore, with the development in MEMS/NEMS, the understanding and control of the nanofretting of monocrystalline silicon both in atmosphere and vacuum has been becoming an important issue of concern.

Since the nanofretting was proposed in 2003, only a few papers were found to discuss the nanofretting behaviors of monocrystalline silicon [1,2]. However, all the previous studies on nanofretting were carried out on the surface of silicon without any chemical treatment. Even though the surface hydrophilicity plays a very important role in adhesion, friction and wear behaviors of material in nanometer scale, the effect of surface hydrophilicity on the nanofretting behavior of Si(100) has not been well addressed [3].

2. Results and discussion

Three kinds of silicon samples with different surface hydrophilicity were prepared for nanofretting tests. They are hydrophilic silicon (OH/Si), original silicon (Si), and hydrophobic silicon (H/Si).

Fig. 1 The friction coefficient $\mu$ versus the displacement amplitude $D$ ($\mu$–$D$) curves of three Si(100) samples against SiO$_2$ microsphere under $F_w=5 \ \mu N$.

The effect of surface hydrophilicity on friction was not obvious in vacuum, but very prominent in atmosphere. In vacuum, since the number of water molecules on Si(100) surfaces was very limited, the contact between tip and sample surfaces was solid-solid contact. As a result, the friction coefficient $\mu$ was almost equal and in the low scale, as shown in Fig. 1. In atmosphere, since all the nanofretting tests in atmosphere were performed at a relative humidity of 35%–40%, the thickness of water film was estimated as 0.31 nm, 0.65 nm and 0.78 nm on H/Si, Si, and OH/Si, respectively. Here, the friction was dominated by solid-solid interaction, viscosity effect and capillarity on H/Si, Si and OH/Si, respectively [3]. Therefore, the initial friction coefficient $\mu$ was lowest on H/Si, medium on Si and highest on OH/Si.

Fig. 2 AFM images of the nanofretting scars on Si(100) surfaces after nanofretting. $F_w=5 \ \mu N$, $D=100$nm and $N=500$ in vacuum, $N=200$ in atmosphere.

Fig. 2 shows the AFM images of the nanofretting scars on Si(100) surfaces. The nanofretting damage of Si(100) surfaces in vacuum was very weak. The damage was characterized as the depression of 0.1–0.2 nm in depth on OH/Si and the hillocks of 0.8–0.9 nm in height on H/Si and Si. However, the nanofretting damage in atmosphere was much more serious, which was identified as the grooves of 8–11 nm in depth on Si(100) surfaces. The depth of grooves was deeper on a more hydrophilic surface. Analysis indicated that even if the nanofretting damage in atmosphere was the coupled results of mechanical interaction and tribochemical reaction, the tribochemical reaction played a dominated role.

3. Summary

(1) The increase in the hydrophilicity of Si(100) surface will expand the stick regime of Si(100)/SiO$_2$ pairs into a higher value of displacement amplitude.

(2) When the Si(100) surface was more hydrophilic, the nanofretting damage was more serious under the same conditions.

(3) Nanofretting damage is more serious in atmosphere than that in vacuum. Compared to the mechanical interaction, the tribochemical reaction played a dominated role in the nanowear of Si(100) in atmosphere.

4. References

Galling Prediction in Pin-Bush Joint Interface

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1. Introduction

In this paper, a galling prediction in pin-bush joint interface is investigated semi-empirically based on thermo-mechanical model undergoing oscillatory motion. Pin-bush joints used in heavy machinery are often heavily loaded with repeated oscillatory motion, thus generating a significant amount of heat at the interface. Traditionally, galling prediction is conducted barely by the extensive experiments to develop a wear map for the variable set of test conditions¹,². The need of analytical tool for the galling prediction has been recognized to save the time and cost for the successful development of pin-bush joint designs.

2. Analysis & Experiment

For predicting galling of Pin-Bush joint, mathematical prediction model of thermo-mechanical heat transfer model is used. Mathematical model of heat equation is expressed as the following equation.

\[ \rho \frac{\partial T}{\partial t} + \rho c \phi(t) \frac{\partial T}{\partial \theta} = k \nabla^2 T + g(r, \theta, t), \quad R_m, R_M, t > 0 \]  \hspace{1cm} (1)

Where, \( R(s) \) is a shaft of radius, \( R(b) \) is a bushing of inner radius, \( T \) is maximum temperature at contact surface and \( \Theta(t) \) is the bushing motion described by the time function. Contact mechanism and mathematical model are integrated to develop the computer program. The steady-state temperature, which is the key influence factor among many input condition, can be predicted. Through FE analysis between pin and bush, rise of temperature can be predicted as shown on the typical example of Fig.1. In this study, it is assumed that the temperature is the most influential factor which decides galling point. In order to compare the analysis results with the experiment, DOE (Design of Experiment) is conducted using steady-state temperature date and influence factor. Based on fact, actual oscillatory test condition in pin-bush interface is selected to link the reality galling phenomenon with thermo-mechanical behavior.

3. Galling Prediction

Using the data of measuring results, steady-state temperature selected at friction coefficient is fixed to 0.16. Also, a point at which abrupt friction rise appears is defined as galling start. Standard of galling prediction is implemented by using analysis of the wear at the galling point. With the given conditions, RSM (Response Surface Model) analysis is performed and galling prediction equation is obtained. Fig.2 shows the comparison of experimental results and results obtained from the galling prediction equation. With this it is confirmed to having data relationship through normality test.

Fig.1 Example of temperature rise in pin-bush joint

Fig.2 Comparison of experiment and life prediction

4. Conclusions

In order to develop improved galling predicting program, mathematical temperature prediction modeling is performed to predict the steady-state temperature at pin-push joint interface. Subsequently, DOE is carried out to relate the steady-state temperature with galling start point. In order to predict galling, life prediction equation was derived by using response surface model. Predictive value which is comparison of experiment and life prediction is confirmed with data normality by using graph applied trend line. Based on the research, henceforth wear phenomenon is predicted at the least number of experiments. By using proposed model, optimized design can be carried out in early stage of design.

5. References

Investigation of Frictional Behavior of CNTs with Respect to Orientation by MD Simulation

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1. Introduction

Over the past decade, Carbon nanotubes (CNTs) have been receiving much interest due to their outstanding mechanical and electrical properties such as high tensile strength, high modulus, and high conductivity\textsuperscript{1}. Various studies on applications and characteristics of CNTs have been reported. According to various tribological studies, carbon nanotubes show quite complicated frictional behavior\textsuperscript{2,3}. Their tribological characteristics often depend on experimental conditions such as orientation, environment, and normal load. In this study molecular dynamics simulations were performed to investigate the frictional behavior of CNTs deposited on an Ag substrate sliding against an Ag upper body under various contact and sliding conditions.

2. Simulation Methodology

Simulation model consisted of silver substrate, silver moving body, and CNTs. CNTs were placed between the silver substrate and the moving body. Brenner’s (Reactive Empirical Bond-order) and Morse potential function was used to model the CNT and Ag material, respectively. Set up of the model in the simulation is shown in figure 1. Initially, the CNTs were indented with the upper body in the load range of 2–30 nN followed by sliding motion of the upper body. Figure 2 shows deformation of CNT as normal load. The longitudinal axis of the CNTs aligned perpendicular to the direction of sliding was set to 0 degree. The frictional behavior was observed with respect to the applied load and sliding angles.

3. Simulation Results and Conclusion

The tribological behavior and friction coefficient of CNT with respect to normal load and sliding direction was obtained. Simulation results showed that the friction coefficient was 0.008 and 0.07 for sliding angle of 0 and 90 degrees, respectively at applied loads less than 5 nN. However, at higher applied loads the difference in the friction coefficient for the three sliding directions was not as significant. As expected, 0 degree sliding angle resulted in the lowest friction coefficient at low applied loads but this was not the case at high applied loads.

According to the simulation results, it was clear that frictional behavior of CNT was affected by normal load and its orientation. The frictional behavior of CNTs sliding against an Ag upper body was attributed to the contributions of sliding and rolling motions of the CNTs which were influenced strongly by the orientation of CNTs and the amount of CNT deformation under the applied load. Further work is being carried out to better understand the effects of each variable to the frictional behavior of CNTs.

Acknowledgement

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4. References

Performances of Needle Roller Bearings

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1. Introduction

Needle roller bearing has a small diameter and large length of rollers compared with other types of roller bearings\(^1\). Therefore, this bearing has a higher load carrying capacity and stiffness than other rolling bearings. In addition, this bearing is suitable for reciprocating motion applications because of the small inertia. Furthermore, since the needle bearing is separable, the outer and the inner rings can be mounted separately in the housing and on the shaft during assembly. However, it is considered that performance of this bearing under various operating conditions is not sufficiently clear.

In this study, the influences of radial load, number of revolutions and eccentricity of radial load on the performance, i.e. bearing torque and thrust, of needle roller bearing were investigated experimentally. In the experiments, single type and double type of bearings were used.

2. Test Bearing

Needle roller bearing used in experiment is schematically illustrated in Fig.1. The bearing has 32 rollers of the length of 13.8mm and roller diameter of 2.5mm. The bearing inner and outer diameter \(\phi d\) and \(\phi D\) are 40mm and 55mm, respectively.

![Fig.1 Test bearing (separation type)](image)

3. Results

Fig.2 shows the relationship between the number of revolution \(n\) and the torque \(T\) at various bearing loads of \(W = 500, 1000\) and \(2000\)N. As can be seen from this figure, bearing torque is increasing with the number of revolutions. The effect of radial load on the torque is also shown in Fig.2. It can be seen that bearing torque increases with load.

![Fig.2 Torque vs number of revolutions](image)

The relationship between the number of revolutions \(n\) and the thrust \(R\), at various bearing loads \(W = 500, 1000\) and \(2000\)N is shown in Fig. 3. Thrust \(R\) means the force in axial direction generated in the bearing. As can be seen from this figure, as the number of revolutions becomes large, the thrust decreases rapidly for each load. It is interesting to note that the thrusts measured for these bearings are bigger than those expected.

![Fig.3 Thrust vs number of revolutions](image)

4. References

Comprehensive Comparison of Mineral, Synthetic, White and Natural Gear Oils

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1. Introduction

Nowadays, in many areas due to a risk of lubricant leakage, biodegradable and/or non-toxic lubricants should be used. Biodegradable lubricants are required in agriculture, horticulture, forestry, quarries, hydrological buildings. Such lubricants are formulated using natural (mainly vegetable) base oils having the highest biodegradability (70-100%). Non-toxic lubricants based on highly refined mineral oils, known as white oils, are biodegradability (70-100%). Non-toxic lubricants based on highly refined mineral oils, known as white oils, are biodegradability (70-100%). Non-toxic lubricants based on highly refined mineral oils, known as white oils, are biodegradability (70-100%). Non-toxic lubricants based on highly refined mineral oils, known as white oils, are biodegradability (70-100%).

Biodegradability of white oils is assessed as 25-45% and is comparable to mineral oils (15-35%) and better that for polyalphaolefins - synthetic hydrocarbon oils (5-30%). In spite of numerous publications concerning white and natural oils, their influence on some important forms of wear (e.g. pitting and micropitting of gears, pitting of rolling bearings) and level of vibrations is rarely investigated. The aim of the work was to assess possibilities of using natural and white oils as potential bases of gear oils.

2. Experimental

Four model gear oils were tested - with mineral, synthetic hydrocarbon (PAO), white, and natural (rapeseed) base oil, all of similar viscosity. The oils contained additives typical of gear oils - EP additives, antifoam additives and antioxidants. Tribological four-ball tests (with sliding and rolling contact) and gear tests were performed. During gear tests the vibration level was measured. Also, physico-chemical analyses of the aged oils were carried out.

3. Results

The results obtained are presented in Table 1.

Table 1 Comparison of the tested oils

<table>
<thead>
<tr>
<th>Feature to scuffing</th>
<th>Test method</th>
<th>Type of oil</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Resistance of gears to pitting</td>
<td>Four-ball sliding</td>
<td>Mineral</td>
<td>0.12</td>
</tr>
<tr>
<td>50% Fatigue life</td>
<td>ASTM D 5183 (modified)</td>
<td>PAO</td>
<td>0.11</td>
</tr>
<tr>
<td>LC50 [cycles*10^6]</td>
<td></td>
<td>White</td>
<td>0.12</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Natural</td>
<td>0.13</td>
</tr>
<tr>
<td>Resistance of gears to micropitting</td>
<td>Four-ball sliding</td>
<td>Mineral</td>
<td>2173</td>
</tr>
<tr>
<td>Time until</td>
<td>Own method</td>
<td>PAO</td>
<td>2151</td>
</tr>
<tr>
<td>micropitting ≥50%</td>
<td></td>
<td>White</td>
<td>2078</td>
</tr>
<tr>
<td>of tooth area [hrs]</td>
<td></td>
<td>Natural</td>
<td>1579</td>
</tr>
<tr>
<td>Gear vibrations</td>
<td>Four-ball rolling</td>
<td>Mineral</td>
<td>186</td>
</tr>
<tr>
<td>Overall level of</td>
<td>IP 300</td>
<td>PAO</td>
<td>35</td>
</tr>
<tr>
<td>acceleration amplitude, RMS</td>
<td></td>
<td>White</td>
<td>87</td>
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<tr>
<td>[m/s²]</td>
<td></td>
<td>Natural</td>
<td>36</td>
</tr>
<tr>
<td>Thermo-oxidative stability in gear pitting tests</td>
<td>FZG gear test:</td>
<td>Mineral</td>
<td>9,0</td>
</tr>
<tr>
<td>TAN change [%/run]</td>
<td>PT C/10/90</td>
<td>PAO</td>
<td>-9,3</td>
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<tr>
<td></td>
<td>FVA 2/IV, 1997</td>
<td>White</td>
<td>-2,0</td>
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<tr>
<td></td>
<td>IP 300</td>
<td>Natural</td>
<td>6,1</td>
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<tr>
<td>Physico-chemical stability during 3-year storage</td>
<td>Determination of VI:</td>
<td>Mineral</td>
<td>2,8</td>
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<tr>
<td>VI change [%]</td>
<td>PN-C-04013:</td>
<td>PAO</td>
<td>0</td>
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<tr>
<td></td>
<td>1979</td>
<td>White</td>
<td>-3,0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Natural</td>
<td>8,2</td>
</tr>
</tbody>
</table>

4. Conclusions

The results show that in comparison with “classical” oils the white oil gives a better resistance of gears to micropitting, more stable physico-chemical characteristics during the long-lasting storage, almost the same antifriction properties, almost the same resistance of gears to scuffing under extreme conditions, similar resistance of gears to pitting, similar level of gear vibrations, and comparable thermo-oxidative stability in gear experiments. The white oil exhibits also a drawback - significant acceleration of pitting of bearing balls.

In comparison with “classical” oils the natural (rapeseed) oil reduces the level of gear vibrations, gives almost the same antifriction properties, similar resistance of gears to pitting, and almost the same stability of physico-chemical characteristics during the long-lasting storage. However, the natural oil exhibits numerous drawbacks such as: lower resistance of gears to scuffing under extreme conditions, much lower resistance of gears to micropitting, significant acceleration of pitting of bearing balls, and worse thermo-oxidative stability in gear experiments.

While removal of the drawbacks of the white and natural oils appears impossible, they can be used as bases of gear oils provided that they will lubricate only gears working under moderate conditions. Additionally, in case of natural oils it is recommended that the time of their exploitation should be shortened.

5. References
The Rolling Contact Fatigue of W-DLC Coated Spur Gears for Various Pinion/Wheel Material Combinations

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1. Introduction

The durability of heavily loaded machine components working in non-conformal contacts (such as gears) depends on two phenomena: scuffing of mating elements and rolling contact fatigue – pitting. Many types of thin hard coatings deposited on the surface of machine components improve scuffing resistance of non-conformal contacts\textsuperscript{1).}

Furthermore, the scuffing resistance of DLC-coated gears also depends on pinion/wheel material combinations\textsuperscript{2).} Both “mixed” combinations (steel pinion/W-DLC-wheel and W-DLC-pinion/steel-wheel) performed significantly worse than the self-mated combinations, they exhibited extremely high wear on the non-coated steel gears.

However, a factor still limiting the scope of application of some coatings is their poor performance under conditions of cyclic contact stress, which leads to accelerated fatigue failures (pitting). This concerns typical PVD coatings (like TiN) and radically limits the area of application of such coatings to machine components subjected only to scuffing\textsuperscript{3).} The aim of the study was to investigate the effects of various pinion/wheel material combinations on the rolling contact fatigue of spur gears.

2. Experimental

The W-DLC (also denoted as WC/C) coating is DLC type representing a-C:H:Me group. The coating consists of an elemental Cr adhesion layer adjacent to the steel substrate, followed by an intermediate transition region consisting of alternating lamellae of Cr and WC, and hydrocarbon layer doped with W. The coating was deposited using PVD process by reactive sputtering. DLC coatings belong to the class of solid lubricant coatings due to the presence of carbon in form of graphite and the graphitization of the initially sp\textsuperscript{3} bonded carbon.

W-DLC coatings were deposited onto carburized spur gears after final grinding.

The experiments were performed using the single-stage pitting test procedure (PT C/10/90) in an FZG type gear test rig, using C-PT gears\textsuperscript{4).} The result of the tests is the fatigue life \( LC_{50} \) related to 50\% probability of failure. Four pinion/wheel gear pairs were selected: steel/steel, steel/coating, coating/steel and coating/coating.

A commercial, mineral automotive gear oil of GL-5 performance level and viscosity grade 80W/90 was used for lubrication.

3. Results

The results indicate that for the coated/coated pair (pinion and wheel coated) and coated pinion/steel wheel pair a significant decrease in the fatigue life compared to the uncoated gears was obtained – Fig. 1.

![Fatigue life \( LC_{50} \) for various pinion/wheel gear material](image)

Fig.1 Fatigue life \( LC_{50} \) for various pinion/wheel gear material

The best results were obtained in the case of the steel pinion / W-DLC coated wheel – even fourfold increase in the fatigue life was observed. This shows a very high potential of application of DLC coatings for gears.

4. Conclusions

The analysis of the results shows that to increase the fatigue life of gears, one should deposit the coating on that gear that is less exposed to wear (i.e. the wheel).

Therefore, the main factor hampering application of thin coatings on heavily loaded elements for many years, i.e. their poor behavior under cyclic stress conditions, has been overcome. This new technology of manufacturing of steel, heavily-loaded machine components covered with coatings which increase the service life of components, allows for application of environmentally friendly oils. This will increase the reliability of machines and reduce pollution.

5. References

Friction and Wear Properties of PTFE Sliding Seals

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1. Introduction

Sealing devices are needed to prevent dust and water invading the positioning mechanisms of outdoor equipment. The frictional resistance generated by sliding seal frequently obstructs the smooth motion of the mechanisms. The problem becomes particularly severe for high-speed and precision control mechanisms. Therefore, the sealing device not only has to provide sealing performance, but also needs to provide low and stable friction properties.

In order to develop the usable sealing device, the influences of roughness and hardness of the seal gland surface on friction properties and durability were investigated using the seal test apparatus shown in Fig.1 with the specimen disks described in Table 1 and PTFE sliding seal. As a result, it was realized that disk F (Electroless nickel plating, 0.1μm Ra) exhibits better performance with respect to friction properties and durability than the other disks in the tests1).

In this report, to explain the cause of the good performance obtained using disk F, surface-profile measurements and surface analyses of the sliding tracks of the disks were performed and the results were compared with the reported test results1).

2. Results and discussion

Fig.2 shows the surface profiles of disks C, D and F measured in radial direction by a stylus-type profilometer before and after the tests. The surface damage of the sliding part in disks C and D was visible as shown in Fig.2 (a) and (b). On the other hand, in the other disks there were not significant differences between the surface profiles before and after the tests as shown in Fig.2 (c).

Fig.3 shows XPS spectra of disks B, D–F after the tests. The peaks of PTFE (689 eV) were recognized in the spectra of disks D and F. In addition, to investigate the distribution of PTFE on the surface of disks D and F, AES were performed. In the disk D, PTFE was only on the concave part of the disk surface damaged by sliding. In the disk F, it could be seen that the sliding surface was covered with PTFE.

It is considered that in the case of disk F, PTFE was transferred from the seal lip to the disk surface by sliding and remained attached on the disk surface because its surface hardness is sufficiently high to withstand the wear of sliding. The PTFE-transferred film facilitates sliding and sealing. That is considered as the cause of the good performance with respect to friction properties and durability obtained by using disk F.

3. Reference

Nanostripe Structure: Fabrication and Friction Characteristics

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1. Introduction

There are many types of top-down fabrication processes for nano-patterns, e.g. laser/ion beam-based methods and nanoimprinting lithography. The major drawback in using laser/ion beam-based methods to fabricate nanoscale patterns is that the longer processing time is required for larger fabrication areas and denser patterns. The limitation of nanoimprinting is in material selection, i.e. substrate materials must be softer than the mold or must be softened by heating, and thus polymer and low-melting glass should be used as the substrate. Thus, we developed a novel nano-patterning method for “nanostripe” fabrication that incorporates a multilayer film and a microscale slope array. We showed the nanostripe fabrication method and examined the friction characteristics of the surfaces with nano-patterns.

2. Basic Concept and Fabrication

Schematic illustration of nanostripe fabrication process is shown in Fig 1. First, a microscale pattern of a periodic slope array is fabricated on a substrate using photolithography. Then, multilayer films are deposited by a PVD process. Finally, the substrate is polished, resulting in the film materials appearing on the surface as parallel stripes. The width of a stripe $w$ is given by:

$$w = t / \tan \theta$$

where $\theta$ is the slope angle of the microscale pattern and $t$ is the thickness of a single layer in the multilayer film measured in the normal direction to the substrate.

Confocal microscope images of “micro-scale” patterns fabricated on silicon substrate are shown in Fig. 2. A (100) plane of silicon substrate was anisotropically etched in TMAH (tetramethyl ammonium hydroxide) solution resulting in a micro-scale surface pattern. The pitch of the ridge in the pattern was 4 $\mu$m. Multilayer films of Fe-Au were deposited on the surface and polished. This resulted in a nano-groove pattern on the flat ridge of the micro-scale pattern (Fig.3).

3. Friction Tests

Friction force was measured on nanostripe structure under lubricated conditions. A rotating friction tester was used for the measurements. A nano-groove pattern was pressed against 4-inch silicon wafer on the rotating base at applied load of 0.5 to 3 N. The rotational speed was varied from 61 to 650 rpm (0.1 to 1 m/s of sliding speed) and friction force was measured. Poly-Alpha-Olefin was used as a lubricant.

The resulting Stribeck curve, where the ordinate and the abscissa axes show the friction coefficient and the sliding speed divided by load, respectively is shown in Fig. 4. The nano-groove pattern survived the friction test at under measurement conditions.
A Study of Tribological Characteristics of Plasma CVD DLC Coatings in Sliding Contact under Oil Lubrication

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1. Introduction

In recent days industries utilize machine elements with engineered contact surfaces which provide desirable properties such as low friction, good abrasion resistance and scuffing resistance. Diamond-like Carbon (DLC) coating is mostly used in surface engineering owing to its superior tribological performance\(^1\). The tribological properties of DLC coatings depend on the coating method, inclusions, tribological system as well as working conditions\(^2\). In this study the tribological characteristics of high thickness DLC coatings of a-C:H type in sliding contact under oil lubrication are investigated.

2. Experimental details

Disk and ring specimens were used for evaluating the tribological characteristics. The specimens made of martensitic stainless steel (JIS SUS440C) were quenched and tempered, followed by grinding or lapping. DLC coating was deposited by using hot cathode Penning Ionization Gauge (PIG) plasma CVD equipment. DLC coating of thickness 3μm, 6μm and 10μm were made on the specimens with and without radical nitriding. The wear test was performed at room temperature with a sliding velocity of 0.5m/s. Tests were conducted using the step-loading method. Each loading step was increased at a mean surface pressure of 0.25 MPa upto 10 MPa, and 1MPa beyond 10 MPa until failure of coatings. The sliding distance in each step was 250 m. The lubricating oil was ISO VG 68 turbine oil. The temperature of oil was maintained at 313 ±2K and the oil was circulated at a rate of 300ml/min. At specific intervals of tests the specimens were cleaned ultrasonically and examined under microscope. The surface roughness and wear amount of both disk and ring were also calculated.

3. Results

Figure 1 shows the friction coefficient and wear amount of the lapped disk and ring specimens. The friction coefficient and wear amount of non-coated specimens increased sharply from the start of test. The 3μm thick nitrided DLC specimens have high frictional coefficient owing to the increased roughness of radical pretreatment. The initial wear amount was also found to be more in 3μm thick nitrided DLC specimen. However the coating showed better wear resistance compared to specimen without nitriding. The 10μm coated DLC specimens without nitriding showed low friction coefficient and less amount of wear than the 10μm thick nitrided specimen. The nitriding pretreatment as well as higher thickness of DLC might have increased the surface roughness which has resulted in more wear amount of the nitrided 10μm DLC specimens. Nitrided 6μm thick DLC specimen showed lesser amount of wear for the test conditions where as the friction coefficient was equivalent to other nitrided specimens.

![Fig.1 Results of lapped disk-ring specimens](image)

3. Summary

In 3μm thick lapped specimens the failure of coating occurred after gradual wearing as the test progressed. Nitrided specimens had better wear resistance. In the case of 10μm thick coated specimens the failure took place in the form of delamination at a shallow distance near the interface after gradual wear of coating. In 6μm thick specimen the DLC coating failed as a portion was removed by delamination. The friction coefficient of nitrided specimens was higher than the ones without nitriding due to increased surface roughness by coating process. The 10μm thick DLC coated specimens provided the best wear resistance under the performed test conditions.

The results of ground specimens will be detailed in the presentation.

4. Acknowledgements

The authors would like to thank SHINKO SEIKI CO., LTD. for providing the DLC coatings. One of the authors, M.Ananth Kumar would like to express gratitude to MEXT, Japan for the award of Monbukagakusho scholarship.

5. References

Friction and Wear Characteristics of Various Solid Lubricating Films for Machine Elements

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1. Introduction

Understanding the relationship between friction and wear characteristics of solid lubricating films and sliding velocity is important to select appropriate solid lubricating film under specific working condition1. In this study, tribological tests were performed to investigate friction and wear characteristics of DLC, CrN+DLC, WC/C and CrN+WC/C coatings under various sliding velocities using ball-on-disk tribometer.

2. Materials and Experiment

All solid lubricating films were deposited on AISI 52100 bearing steel ball. The 1~2 µm-thick DLC and 1~2 µm-thick WC/C layer were deposited using PACVD with acetylene and magnetron cathode sputtering with acetylene, respectively. For all solid lubricating films, the 0.5 µm-thick Cr layer deposited by sputtering was used as the intermediate layer to increase the adhesion between each solid lubricating film and the AISI 52100 bearing steel ball. In the case of CrN+DLC and CrN+WC/C, the 0.5 µm-thick CrN layer deposited by sputtering was placed between DLC or WC/C layer and Cr layer. The counter surface was carburized SCM 415 Cr-Mo steel disk.

The test conditions are shown in Table 1. Under each test condition, the kinetic friction coefficient was obtained through dividing time average value of the kinetic friction force by normal load. Wear rate was calculated by dividing wear volume by normal load and sliding distance.

3. Results

The kinetic friction coefficients and wear rates of solid lubricating films are shown in Figs. 1 and 2. When the sliding velocity was below 0.25 m/s, the kinetic friction coefficients of all solid lubricating films increased with increasing sliding velocity. Under the 0.5 and 1 m/s sliding velocity conditions, the kinetic friction coefficients of DLC, WC/C and CrN+DLC were lower than that obtained under the 0.25 m/s sliding velocity condition. On the other hand, the kinetic friction coefficients of CrN+WC/C were higher than that in the 0.25 m/s sliding velocity condition. Under the same test condition, the kinetic friction coefficients of DLC and WC/C were lower than those of CrN+DLC and CrN+WC/C, respectively.

The wear rates of all solid lubricating films decreased with the increase in sliding velocity when the sliding velocity was below 0.25 m/s. Under 0.5 and 1 m/s sliding velocity conditions, the wear rates of all solid lubricating films were higher than that under the 0.25 m/s sliding velocity condition. In the case of CrN+DLC and CrN+WC/C, the wear rates were lower than those of DLC and WC/C obtained under the same test condition, respectively.

4. Conclusions

Under the test conditions considered in this study, the friction and wear characteristics of solid lubricating films were affected by the variation in sliding velocity and the existence of CrN layer.

5. Acknowledgement

This work was supported by the Brain Korea 21 Project in 2010.

6. Reference


Table 1 Test conditions

| Ball specimen | DLC, CrN+DLC, WC/C, CrN+WC/C |
| Disk specimen | carburized SCM 415 Cr-Mo steel |
| Sliding distance | 200 m |
| Normal load | 2.63 N (contact pressure=365MPa) |
| Sliding velocity | 0.0313, 0.0625, 0.125, 0.25, 0.5, 1 m/s |
| Air temperature | 18~20 °C |
| Air humidity | 35~40 % R.H. |
Tribological Evaluation of a Trench Type Wear-Resistant Probe

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1. Introduction

Nanometer-scale processes utilizing a scanning probe system (probe-based nano lithography¹, probe-based data recording², etc.) are state-of-the-art technologies which are able to handle nm-sized tiny patterns. In order to bring these technologies from R&D stage to practical use, significant improvement of the wear-resistance of the probe tip is required for reliable and long-term operation of the system.

2. Concept

The authors have developed a trench type wear-resistant probe (Fig.1) which has potential for overcoming the above problem. The probe cantilever made from non-doped Si has a trench-shaped structure filled with a metal (Au) electrode at its end. The tip of a Si slider and the electrode contact simultaneously to the substrate so that the wear-progressive speed becomes slower because of the relatively larger contact area. Since the width of the electrode is uniform at its length direction, the size of its electric contact area to the substrate is expected to remain constant even after the long distance sliding.

3. Experiment

A sliding test of the fabricated probe is performed using an SPM system which can apply a bias voltage to the substrate holder. The probe tip is applied to the surface of conductive metal substrate (Ru) and both the contact AFM and current image are measured (Fig.2). The measurement is performed repeatedly without changing the scanning area.

Initially almost all the area of the current image is white. This indicates an excellent electric contact between the protruded Au electrode and the substrate. As the test progresses, the Au electrode is gradually worn out and a black area appears in the current image indicating lack of electric contact. Though the scanning area is fixed, the pattern of grooves on the Ru surface, shown in the AFM images, changes, depending on whether the current image is white or black. This implies the two tips of metal electrode and Si slider do not always contact simultaneously. Instead, they contact alternately and pick up the topography at each contact point one after another (Fig.3). However, it is considered that the simultaneous contact of these two tips is realized at the region where the current image is white but no clear grooves are observed in the AFM image as indicated by arrow in Fig. 2.

4. Conclusion

The wear-resistant probe has been developed and evaluated. Simultaneous contact of the conducting electrode and the insulating part has been observed.

5. Acknowledgement

This work was supported by New Energy and Industrial Technology Development Organization (NEDO).

6. References

The Significant Reduction of Wall Abrasion in the Direction Changing Devices for Two Phase Fluid Flow

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1. Introduction

The rate of erosion or abrasion wear in two phase pneumatic conveying pipes is proportional to the velocity, quantity of flow and shape of particles. By creating a vortex flow before a bend, a significant wear reduction, balanced with velocity and pressure loss was obtained.

2. Computational Fluid Dynamics (CFD) – two phase flow; Discrete Phase Model (DPM)

Figure 1 shows the control volume (no vortex flow) where, in the main impact point of the elbow the maximum erosion value in terms of mass loss was 3.46e-07 kg/m²-s.

![Fig. 1 Control volume wall erosion](image1)

The vortex flow was induced by subtracting a spiral volume, of different length, pitch and cross section from the inner volume of a 50 mm diameter pipe, just before the bend. From several options, the best results were obtained by a spiral shaped volume with a rectangular cross-section of 6 mm x 6 mm and a pitch of 100 mm, placed at the end of the horizontal portion just before the elbow (volume 2). Figure 2 shows the wall erosion for volume 2, where the maximum erosion value in terms of mass loss was 5.96e-08 kg/m²-s, as compared to 3.46e-07 kg/m²-s for the control volume. The volume 2 erosion represents only 17.22% of the control volume erosion. The wall erosion translated in mm – wall thickness loss per hour, is 0.158 mm/hour for the control volume against 0.027 mm/hour for volume 2. Furthermore as can be seen from figure 2 the maximum impact points are scattered in a spiral pattern avoiding the possibility of total wall perforation, for a longer time than the control volume. The wall thickness of the pipe used is 3.91 mm, consequently the time to perforation for the control volume and volume 2 will be 24.7 hours and 144.8 hours respectively. The CFD results show an increased life expectancy for volume 2, of 5.82 times more than that of the control volume.

Further practical validation experiments must be done with different types of solids and concentrations.

![Fig. 2 Volume 2 wall erosion](image2)

Figure 3 shows the particle traces defined by velocity magnitude. The maximum velocity is 34.8 m/s while the minimum velocity is 12.3 m/s. The majority of the particles move with velocities ranging from 22.5 m/s to 30.7 m/s well above the 12 m/s saltation velocity.

![Fig. 3 Particle traces defined by velocity magnitude](image3)

3. References


CFD Analysis of Spool Valve with Different Shaped Groove Considering Cavitation

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1. Introduction

In order to maintain the accurate and precise movement of the actuator working in a hydraulic systems, smooth function of the fluid flow control valves is required. The spool type directional control valve has locking problem¹. To reduce the possibility of hydraulic lock, peripheral grooves balancing uneven pressure distribution in the radial clearance are introduced onto the spool. To study the fluid film lubrication behavior of spool valve with deep groove the Navier-Stokes equation is more suitable than the Reynolds equation. This is because some of assumptions used in Reynolds equation are not valid when cavitation occurs or the fluid inertia is significant². In this study, the lubrication characteristics of a tilted spool valve with grooves of different shape was analyzed using computational fluid dynamics (CFD) software. The cavitation, which occurs at the groove region was taken into account.

2. Method & Numerical Model

Commercial available computational fluid dynamics (CFD) software, FLUENT has been used to analyze the lubrication characteristics of spool valve. Cavitation model takes under the account the bubbles dynamics and the effect of non-condensible gases. The schematic diagram of a tilted spool valve is shown in Fig.1 while cross sections of four types of grooves analyzed are shown in Fig. 2. Saturation pressure, boundary pressure condition and sliding speed are shown in Table 1 and the properties of working fluid and geometry of spool valve are listed in Table 2, 3.

Table 1 Pressure conditions and sliding speed

| Pressure | \( P_{a0} \), \( P_{a1} \) (Pa) | 0 |
| Sliding speed | \( U \) (m/s) | 1 |

Table 2 Properties of the working fluid

<table>
<thead>
<tr>
<th></th>
<th>Oil</th>
<th>Oil-Vapor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density (kg/m³)</td>
<td>962</td>
<td>0.02556</td>
</tr>
<tr>
<td>Viscosity (kg/m·s)</td>
<td>0.013468</td>
<td>1.256×10⁻⁵</td>
</tr>
<tr>
<td>Surface tension (N/m)</td>
<td>0.0378</td>
<td></td>
</tr>
</tbody>
</table>

Table 3 Geometrical parameters of a spool valve

| Height of the groove (mm) | 7.51 |
| Height of the valve (mm)  | 7.5  |
| Height of the clearance (mm) | 0.5  |
| Length of the clearance (mm) | 20   |
| Length of the groove (mm) | 1.5  |
| Length of the valve (mm)  | 0.5  |
| Angle of the groove (degree) | 0.0228 |
| Number of groove | 2 |

Fig.1 Schematic diagram of a tilted spool valve

Fig.2 Cross sections of the grooves analyzed

Fig.3 Lateral force and volume flow rate vs the groove shape

3. Results & Conclusions

The effect of groove shapes on the pressure distribution, lateral force and leakage flow rate in the clearance is presented.

U (type-2) groove shape gives lower lateral force and volume flow rate than the other groove shapes analyzed.

4. Acknowledgment

This work was supported by the Brain Korea Project in 2010.

5. References

Fabrication of Nano-Structured Surface Texture on Fluorocarbon Thin Film and Its Tribological Properties

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1. Introduction
Fluorocarbon is commonly used as a coating material for water and oil repellent surfaces. However, this coating’s durability under rubbing is low due to its poor mechanical properties. Since nano-texture might improve the friction and wear properties, a surface modification with atmospheric-pressure microplasma is employed to fabricate isotropic nano-structures on the fluorocarbon thin film. In this study, the effects of nano-structured surface texture on tribological properties of fluorocarbon thin film are investigated using sliding friction tests.

2. Materials and Methods
The ATR-FTIR spectra of fluorocarbon thin film after annealing treatment and polytetrafluoroethylene (PTFE) film are shown in Fig. 1 for comparison. The film, of 1 μm thickness deposited on the substrate, was produced using a fluorine coating agent (FS-1020 TH-0.5 from Fluoro Technology Co.). Nano-structures were generated on the surface by the atmospheric-pressure microplasma with argon gas. Friction experiments were conducted on a pin-on-plate tribometer consisting of a fluorocarbon thin film coated on an optical glass plate and a cast iron pin (120 mm radius). All of the tests were performed at a normal load of 0.30 N, sliding speed of 0.32 mm/s and stroke of ±1.0 mm. The lubricant used was hexadecane.

3. Results and Discussion
AFM images of fluorocarbon thin film before and after plasma treatment are shown in Fig. 2. Height of bumps on the plasma treated surface was about 40 nm. A fabrication mechanism of this texture was attributed to anisotropic etching of film surface with oxygen radicals. Figure 3 shows the variations in friction coefficient with sliding distance. Results showed that finely structured texture can provide a stable low friction over a longer period of time than the untreated surface. From microscopic analysis of wear scar (Fig. 4), it was found that this film durability enhancement depends on wear dispersal in the contact area.

4. Conclusion
Nano-structured texture can improve friction and wear characteristics of fluorocarbon thin films under lubrication conditions.

5. References
Adhesion Characteristics of UV Curable Resins for Nanoimprint Lithography (NIL)

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1. Introduction
Nanoimprint lithography (NIL) is one of the promising methods for micro/nano scale devices fabrication\(^1\). Ultra violet (UV) NIL as one of NIL methods has some advantages, such as lower imprint pressure, shorter time and lower temperature for process, compared with thermal NIL. In UV NIL, the patterns transferred by imprinting frequently experience fracture or deformation due to the adhesion between a resin and a stamp in a separation process. For this reason, the research about adhesion and friction between these materials becomes very important\(^2\). In this study, the adhesion phenomenon between UV curable resins and fused silica lens for UV NIL application was investigated.

2. Materials and Experiment
AMO-NIL MMS4 (AMO GmbH, Germany), PAK-01 (Toyo Gosei, Japan), mr-UVCur21 (micro resist tech., Germany) and NIP-SC58LV200 (ChemOptics. Korea) were used as UV curable resins. Each UV curable resin was spin-coated on Si wafer at 3000 rpm for 60 seconds. Before the spin coating, Si wafer was cleaned by piranha solution. Fused silica lens was used as the material contacting with UV curable resins. The lens was also cleaned by piranha solution before each test. Its radius of curvature was 11.502 mm.

Adhesion tests were conducted using the specially designed adhesion force measurement system as shown in Fig.1. In the adhesion test, the fused silica lens was brought into contact with UV curable resins, and UV was exposed to the contact surface. Adhesion force was measured when the fused silica lens was separated from the UV curable resin. The contact surface was observed during the test using a microscope and a CCD camera. The experimental conditions are listed in Table 1.

3. Results
Pull-off forces per unit area are shown in Fig.2 when the applied loads were 0.25 N and 0.5 N for each resin. Pull-off force per unit area means the measured pull-off force divided by the maximum contact area observed by a microscope during the test. In case of mr-UVCur21, pull-off force per unit area was larger than those for any other resins, and that for NIP-SC58LV200 was the smallest. Pull-off forces per unit area with the applied load were similar for AMO-NIL MMS4, PAK-01 and mr-UVCur21 when the applied loads were 0.25 N and 0.5 N. However, in case of NIP-SC58LV200, pull-off force per unit area when the applied load was 0.25 N was 40% larger compared with that in case of 0.5 N.

<table>
<thead>
<tr>
<th>Table 1 Experimental conditions for UV resins</th>
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<tbody>
<tr>
<td>AMO-NIL MMS4</td>
</tr>
<tr>
<td>----------------</td>
</tr>
<tr>
<td>Applied load</td>
</tr>
<tr>
<td>Approach velocity</td>
</tr>
<tr>
<td>Separation velocity</td>
</tr>
<tr>
<td>UV dose</td>
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</tbody>
</table>

4. Acknowledgements
This research was supported by a grant (2010K000197) from Center for Nanoscale Mechatronics & Manufacturing, one of the 21st Century Frontier Research Programs which are supported by the Ministry of Education, Science and Technology in Korea. This research was also supported by the Brain Korea 21 project in 2010.

5. References
Lubrication Characteristics under Transient Conditions in a Swash-Plate Axial Piston Pump

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1. Introduction

The integration of electronic control into hydraulics has led to more severe and variable operating conditions. This study examines lubrication characteristics in a variable-displacement swash-plate piston pump. A computer simulation was carried out for the dynamic cylinder pressure and the change of lubrication gap profile between the barrel and valve plate considering the variation of swash-plate angle. All developed algorithms in different physical domains were combined for the investigation of several transient operating conditions in the piston pump.

2. Analysis

Figure 1 and Table 1 represent general pump configuration and specification respectively. ith cylinder in the barrel has pressure \( P_i \) and volume \( V_i \). The swash-plate angle is \( \beta \) and its velocity and acceleration are \( \beta' \) and \( \beta'' \) respectively. Pressure \( P_i \) is calculated as followed:

\[
\frac{dP_i}{dt} = \frac{K}{V_i} \left( \frac{dV_i}{dt} + Q_{\text{Valve Port},i} + \sum Q_{\text{leakage},i} \right)
\]

(1)

where \( K \) is bulk modulus

\( dV/dt \) and \( Q_{\text{Valve Port}} \) are expressed with \( \beta' \) and rotation angle, and then we calculated at particular time if the pump rotating speed is constant. In order to calculate \( \sum Q_{\text{leakage}} \), partial differential and ordinary differential equations (Reynolds equation and equations of motion respectively) were established and solved simultaneously for only \( Q_{\text{Barrel/Valve}} \). For the other flow leakage \( Q_{\text{Piston/Cylinder}} \) and \( Q_{\text{Slipper/Swash}} \), their film thicknesses were assumed to be constant over the length and breadth of the gaps used in this study.

For the dynamic cylinder pressure, we consider orifice loss model and fluid momentum effect1. Applying the fluid momentum model to the pipe line, the flow is expressed as followed:

\[
\frac{dQ_{\text{Pipe}}}{dt} = \frac{A_{\text{Pipe}}}{\rho L_{\text{Pipe}}} \left( P_i - P_{\text{out}} - \frac{1}{2} \rho U_{\text{Pipe}}^2 \sum \xi \right)
\]

(2)

where \( \xi \) is hydraulic loss coefficient

Runge-Kutta method and finite difference method were implemented to solve the equations (1) and (2) simultaneously.

3. Result

Figure 2 shows the cylinder pressures obtained by the multi-piston model proposed in this paper and single-piston model which has been widely used in the literature. Swash-plate motion was assumed to be a simple trigonometric function at this stage. The multi-piston model can properly predict transient cylinder pressure while the single-piston model can not.

4. Conclusion

A method considering fluid momentum effect and swash-plate motion was proposed in this paper. More realistic fluid film history for transient operating conditions was obtained using the method.

5. Acknowledgement

This work was supported by the Brain Korea 21 Project in 2010.

6. References

Hydrodynamic Lubrication Analysis of a Piston Ring and a Micro-Dimpled Cylinder Liner

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1. Introduction

Approximately 25-40 percent of friction loss in an internal combustion engine occurs between a piston ring and a cylinder liner1,2. Recently, the surface texturing technique to generate micro-dimples on a cylinder liner has been used to increase the efficiency of an engine by decreasing friction loss between a piston ring and a cylinder liner.

In this study, the depth, radius and gap of micro-dimples are considered as design parameters, and the effect of those on hydrodynamic lubrication characteristics between a piston ring and a micro-dimpled cylinder liner is analyzed numerically.

2. Analysis

Because of the shape of a piston ring and micro-dimples generated on a cylinder liner, cavitation may occur in oil film. Elrod’s universal equation implementing JFO film rupture / reformation boundary conditions is adopted to predict the cavitation region properly and to calculate the pressure distribution between a piston ring and a micro-dimpled cylinder liner3. The multigrid algorithm is used to improve the rate of convergence in numerical analysis.

3. Results

Figure 1 shows the geometry of the piston ring and the micro-dimpled cylinder liner. Table 1 and 2 present the specification of the engine and parameter values of dimples. Varying the value of one parameter and keeping the others constant, the analysis is carried out in order to analyze the effect of each parameter on hydrodynamic lubrication characteristics between the piston ring and the micro-dimpled cylinder liner.

Figure 2 shows the average friction loss for the full engine cycle with variation of the radius and depth of dimples. As can be seen from Fig. 2, the average friction loss for the micro-dimpled cylinder liner is less than that for the non-textured cylinder liner and decreases with increasing the radius or depth of dimples.

4. Conclusion

In this study, the effect of micro-dimples on hydrodynamic lubrication characteristics between a piston ring and a micro-dimpled cylinder liner is analyzed numerically. The results show that micro-dimples can make friction loss decrease in comparison with a non-textured cylinder liner.

5. Acknowledgement

This work was supported by the Brain Korea 21 Project in 2010.

6. References

Comparison of Iterative Methods for the Solution of Compressible-Fluid Reynolds Equation

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1. Introduction

Numerical simulation in many tribological studies requires the solution of Reynolds equations for air- or gas-lubricated bearings¹. Due to the lack of analytical solution the Reynolds equations are usually solved numerically after the equations are discretized. The discretized equation forms a large sparse system of linear equations. To search an effective solution method for these simultaneous equations several factors may be required to consider, such as execution time, memory storage requirement, and perhaps more importantly, accumulated round-off errors due to frequent operation of floating-point numbers. Iterative methods are advantageous over direct methods in terms of aforementioned factors.

2. Methods

In this study, the air (lubricant) in the analysis is treated as Newtonian fluid and the variation of viscosity and density across the film direction is neglected. The condition of no-slip at the bearing surface interface is also assumed.

Two fixed-point methods, a direct fixed-point iterative (DFI) method and Newton’s method, are applied to transform the Reynolds equation to iterative form. The iterative methods examined are Gauss-Seidel (GS) method, successive-over-relaxation (SOR) method, a preconditioned conjugate gradient (PCG) method, and a multigrid method.

When an air bearing model² is solved numerically, two levels of iteration are involved. For instance, when the Newton’s method is applied the outer iteration assembles the system of equations, and the inner iteration solves the system of equations. Usually, the inner iteration is performed only up to a precision that conserves the convergence properties of the outer iteration.

The data of the journal air bearing: shaft speed = 30,000 rpm; eccentricity ratio = 0.9; air viscosity = $184.6 \times 10^{-7}$ N·s/m²; bearing width = 38.1 mm; journal diameter = 38.1 mm; and radial clearance = 50 µm.

3. Results

Fig. 1 shows the results of the bearing model linearized by the Newton’s method and solved by the PCG method with relaxation factor of 1.95. Except for the last case, the CPU time is decreased as the number of inner iteration number is increased.

Fig. 2 shows the results for the air bearing model, linearized by the Newton’s method and solved by the multigrid method, are shown in Fig. 2. Note that the CPU time shown for each outer iteration is the overall execution time from the first iteration to the respective iteration.

4. Conclusions

In this study, the Newton’s method demonstrates its effectiveness over DFI method if the same iterative method is applied. The PCG method is not widely applied in solving air bearing problems, which may be due to the method is difficult to program than the GS and SOR methods, and less efficient than the multigrid method. It is also noted that the SOR method can easily handle non-continuous boundary conditions in bearings, such as in a hydrostatic recess, by setting the pressure at specific grids in iteration. A compromise has to be made in terms of ease of use as well as programming effort for the solution of the Reynolds equation.

5. References

Tribological Surface Chemistry of Model Lubricant Additives Measured in Ultrahigh Vacuum

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1. Introduction

It has been found previously that lubricant additives that operate under extreme-pressure (EP) conditions react rapidly at the hot (~1000 K) sliding interface to decompose on the surface to deposit the constituent elements. A boundary layer film then forms by a thermally activated, electric-field assisted film growth mechanism.1,2 The following describes a boundary film formation process that occurs at room temperature due to a shear-induced intermixing between the surface and subsurface region caused by vortices that have been modeled using molecular dynamics (MD) simulations.3 Experiments are carried out using an ultrahigh vacuum tribometer that has been described elsewhere.4 This operates at a base pressure of ~1×10^-10 Torr and enables friction coefficients to be measured on clean, well-characterized substrates by sliding a pin on a surface with a well-defined load and sliding speed. The system allows lateral forces to be measured to obtain friction coefficients. The chamber contains a high-resolution electron gun with a channeltron secondary-electron detector that enables images of the surface to be obtained with a resolution of ~ 1 μm, in particular of the scars formed by rubbing. The electron gun can be configured to emit higher currents (~ 1 μA) but at a lower resolution (~ 70 μm) to allow Auger spectra of the wear scar to be obtained. Rubbing experiments are carried out using a tungsten carbide ball, which is 6.5 mm in diameter which can be cleaned in vacuo by electron-beam heating. The substrate used for these experiments is a copper foil that had been polished to 1 μm finish with diamond paste prior to insertion into the UHV chamber.

2. Results and Discussion

The results are illustrated here using dimethyl disulfide (DMDS) on a copper surface. It has been shown that DMDS reacts with copper by S-S bond scission to deposit methyl thiolate (CH₃-S) species on the surface.5 Rubbing a thiolate-covered copper surface causes the sulfur Auger signal within the wear scar to decrease to zero after ~20 rubbing cycles. Note that the temperature rise during rubbing was < 1 K. To determine the fate of the sulfur, whether it is worn from the surface or has been transported into the bulk, all sulfur was removed from the copper surface, in particular, that outside the wear scar, by Argon ion bombardment. The sample was then heated to ~780 K in ultrahigh vacuum. Auger analysis of the wear scar region showed that sulfur was present inside the wear scar, but no sulfur was detected outside. This indicates that the sulfur had penetrated the bulk to form a metastable subsurface layer and diffused to the surface once again on heating, and that shear had caused it to be transported to the subsurface region as predicted from MD simulations.3 In order to establish whether this mechanism is capable of forming a friction-reducing, boundary film, the friction coefficient was monitored while continually dosing the copper surface with DMDS. The results are summarized in Fig. 1, which shows that the friction coefficient decreases when the sample is exposed to DMDS (after a run-in period of 70 scans). A depth profile obtained by sequentially Argon ion bombarding the surface and collecting Auger spectra confirmed that sulfur had penetrated the subsurface region.

Fig.1 Plot of friction coefficient versus number of cycles measured in the ultrahigh vacuum tribometer at a sliding speed of 1×10^-3 m/s at a normal load of 0.44 N. The clean surface was initially scanned 70 times to reach a steady-state value of friction coefficient and then DMDS was introduced via a dosing tube with a background pressure of 5×10^-8 Torr.

3. Conclusions

Tribological experiments carried out in UHV reveal a novel, low-temperature boundary film formation pathway which is illustrated using dimethyl disulfide on copper surfaces. Other examples will be given in the presentation.

4. References

Effect of gear surface and lubricant interaction on mild wear

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1. Introduction

Standard rating procedures for gears1 take no account of the effects of varying lubricant and surface finish. Moreover, environmental awareness has led to the introduction of environmentally adapted lubricants (EALs) that differ from traditional mineral base oils, for example, in terms of additive response, as discussed by Bergseth et al.2. Increased knowledge in this area is needed in order for gear transmissions to compete with other efficient power transmissions, such as electric motors and hydraulic systems.

2. Experimental Details

A twin disc test machine was used to simulate the rolling/sliding gear contact for different surface finishes. The surfaces were produced by: transverse grinding followed by polishing; transverse grinding only; and transverse grinding followed by preheating. The surfaces are referred to as polished, ground, and preheated. The preheating means that the discs were covered and heated with a mineral lubricant containing one additive.

The disc specimens were made from case hardened gear steel and the contact was sparsely lubricated with an environmentally adapted lubricant and a commercial heavy truck gearbox lubricant. The six surface/lubricant combinations were compared at a slip ratio of 10% and a maximum Hertzian contact pressure 1.5 GPa (representative of a gear contact).

In order to obtain information about the composition of chemically reacted surface boundary layers, the worn disc specimens were analysed with a method called glow discharge - optical emission spectroscopy (GD-OES).

3. Results

The mean coefficients of friction for all test combinations (repeated once) are shown in Fig. 1. The commercial lubricant resulted in the lowest mean coefficient of friction for all surfaces. The mean coefficient of friction is not affected by the preheating for both lubricants.

The difference in wear (not shown) is significant between the polished and the non-polished surfaces, although manufacturing variations of the two surface specimens complicate a comparison. The preheating increased the mass loss for the ground surface with the commercial lubricant by a factor of 30.

The surface depth profiles for all surfaces show different behaviours with the two lubricants.

4. Conclusions

The results show that the interactions of different surface finishes and lubricants have different impacts on wear, friction behaviour and on the reacted surface boundary layer formed by the lubricant.

The commercial lubricant resulted in the lowest mean coefficient of friction, which is advantageous for higher gear transmission efficiency. However, the preheating (which was intended to enhance the build-up of easily sheared surface boundary layer) seems to have hindered the commercial lubricant from reacting properly with the surface or changed the working conditions for the commercial lubricant.

The polished surface interaction with both lubricants indicated that a correlation existed between the reacted surface layer depth (obtained from GD-OES measurements) and friction. That is, a slight thicker reacted boundary layer for the commercial lubricant seems to have lowered the coefficient of friction. This relation was not as clear for the ground surfaces.

5. References

Study of interactions between probes and ultrathin liquid lubricant films using atomic force microscope

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1. Introduction

In order to achieve a magnetic recording density of more than 1 Tb/in², the head-disk spacing needs to be less than 2–3 nm. It is important to clarify the interactive forces between the sliders and the disk surfaces in a nanometer head-disk interface (HDI). In particular, the adhesive and friction forces between the thermal fly-height control (TFC) sliders and the lubricant film are of great importance. Therefore, in this study, the interactions between the probes of an atomic force microscope (AFM) and ultrathin liquid lubricant films were studied. The force curve and the lateral force were evaluated and the effects of the main chemical chain structures of the lubricant molecules as well as the lubricant film thickness on the interactive forces were investigated and discussed.

2. Experimental

An environmentally controlled AFM was used in this study. The force curve that indicated the interactions between the probes and the ultrathin liquid lubricant films was measured and evaluated as a model experiment for the actual HDI in hard disk drives by varying the lubricant film thickness. The maximum attractive force in the force curve is called the adhesive force; this force is required to break the contact between the probe and the disk surface. The lateral force was also measured. The substrates used in these experiments were 1.8-in glass disks prepared with a magnetic alloy and a 3-nm amorphous carbon film. The average surface roughness was approximately 0.6 nm. The test lubricants used were Z-tetraol2000 and TA-30.

3. Experimental result and discussion

Figure 1 shows the effect of the lubricant film thickness on the adhesive force for TA-30 and Z-tetraol2000. The adhesive force decreases gradually with an increase in the lubricant film thickness in both cases. Furthermore, the adhesive force of TA-30 is considerably smaller than that of Z-tetraol2000 in the case of a thin lubricant film and tends to become almost equal to that of Z-tetraol2000 with an increase in the lubricant film thickness. This is because the TA-30 film exhibits higher coverage than Z-tetraol2000. In other words, the TA-30 film completely covers the diamond-like carbon (DLC) film in the case of a thin lubricant film, as compared to the Z-tetraol2000 film. Figure 2 shows the effect of the lubricant film thickness on the lateral force in both cases when the applied force was 45.4 nN. The lateral force, which is suggested to be equivalent to the friction force, decreases gradually with an increase in the lubricant film thickness in both cases. This is very similar to the trend observed in the case of the adhesive force. However, the lateral force of TA-30 is greater than that of Z-tetraol2000. A similar result is confirmed from the pin-on-disk friction test. Figure 3 shows the estimated conformation of the test lubricant molecules. The abovementioned contrast characteristics can be explained by using this conformation on the DLC film. Therefore, it was found that the interactive forces such as adhesive and lateral forces at a lubricant film thickness of approximately one monolayer strongly depend on the conformation of the lubricant molecules on the DLC thin films.

4. References

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Computer Modeling of Electrical-Frictional Interaction of Friction Couples

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1. Introduction

Computer modeling of electro-friction interaction of friction couples is carved out using the method of the Generalized Thermal Dynamics of Friction and Wear of Tribosystems (GTDFWT) [1-3].

2. Research details

The Informational-Tribotechnical System of Forecasting, Control and Managing Changes of Exploitation Indicators of Tribosystems (ITS FCME EIT) provides a base for the computer modeling of EFI[2].

The dynamics of electrical-frictional interaction in time are conditioned by availability of a gradient of temperatures on normal in contact (grad $\mathbf{g}$) [3]. The gradient is caused by difference in temperatures of conjugate surfaces of friction, i.e. jump of temperatures in contact. This was taken into account in the model of EFI of frictional couples.

Time factor is exhibited only through a gradient of temperatures on the normal in contact (grad $\mathbf{g}$), which is an argument used in the GTDFWT set of equations that models wear and its dynamics from the thermic point of view.

The designed computer model can be described as a circuit of the sequential closed acts of mutual changes of parameters of EFI in time, where the reached performances’ level of the completed act is a starting-point for modifications of the new act.

The computer model of EFI that reflects the latter’s time dynamics contains ratios and functional associations that form a closed system of functionals of mutual changes, analytically noted as follows:

\[
\begin{align*}
\mathbf{g}_{1,2} & = \mathbf{g}_{1,2} \ominus \mathbf{g}_{6,1,2} = 0 \\
\mathbf{g}_{\text{max},1,2} & = \mathbf{g}_{1,2} - \mathbf{g}_{\text{h},6,1,2} = 0 \\
\mathbf{g}_{\text{h},1,2} & = \mathbf{g}_{\text{h},1,2} (N = \{P, V, J, f, R\}, \mathbf{V}, \mathbf{M}, \mathbf{P}, \mathbf{K}, \mathbf{X}_{1,2}, \mathbf{T}, \mathbf{F}_{1,2}, \mathbf{I}_{1,2}) \\
\mathbf{g}_{\text{h},6,1,2} & = \mathbf{g}_{\text{h},6,1,2} (\mathbf{g}_{1,2}, J, \mathbf{F}_{1,2}, \mathbf{T}, \mathbf{X}_{1,2}, \mathbf{I}_{1,2}) \\
\mathbf{g}_{\text{h},6,1,2} & = \mathbf{g}_{\text{h},6,1,2} (N = \{P, V, J, f, R\}, \mathbf{V}, \mathbf{A}, \mathbf{A}_{\text{x}}, \mathbf{m}_{\text{m}}, \mathbf{F}_{\text{D}}, \mathbf{X}_{1,2}) \\
\mathbf{m}_{\text{m}} & = m_{\text{m}}(P, \mathbf{M}, \mathbf{K}, \mathbf{X}_{1,2}, \mathbf{F}, \mathbf{X}_{1,2}, \mathbf{X}_{1,2}, \mathbf{I}_{1,2}) \\
\mathbf{A}_{1,2} & = A_{1,2}(P, f, \mathbf{m}_{\text{m}}, \mathbf{F}) \\
\mathbf{A}_{\text{c}} & = A_{\text{c}}(R_{\text{e}}, A_{1,2}, \mathbf{F}) \\
\mathbf{F} & = f(\mathbf{g}_{\text{max},1,2}, \mathbf{g}_{\text{h},1,2}, \mathbf{g}_{\text{grad},1,2}) \\
\mathbf{R}_{\text{e}} & = R_{\text{e}}(\mathbf{g}_{\text{max},1,2}, \mathbf{g}_{\text{h},1,2}, \mathbf{g}_{\text{grad},1,2}) \\
\mathbf{I}_{1,2} & = I_{1,2}(\mathbf{g}_{\text{max},1,2}, \mathbf{g}_{\text{h},1,2}, \mathbf{g}_{\text{grad},1,2}) \\
\mathbf{g}_{\text{grad},1,2} & = \mathbf{g}_{\text{grad},1,2}(\mathbf{g}_{\text{max},1,2}, \mathbf{g}_{\text{h},1,2}, t)
\end{align*}
\]

where:

- $f$ – friction coefficient; $R_{\text{e}}$ – contact electrical resistance;
- $I_{1,2}$ – linear wear intensity; $t$ – friction time; $P$ – load; $V$ – sliding velocity; $J$ – current strength; $N$ – electrical-frictional interaction power; $A_{1,2}, A_{\text{c},a}$ – actual area of contact (frictional and electrical); $m_{\text{m}}$ – probable number of contacts on the actual area of contact; $\Phi_{\text{M}}$ – phys-mechanical characteristics; $\Phi_{\text{X}}$ – thermal-physical characteristics; $\Phi_{\text{D}}$ – electro-physical characteristics; $\mathbf{g}_{\text{grad},1,2}$ – mean surface temperature (absolute); $\mathbf{g}_{\text{h},1,2}$ – maximum friction surface temperature; $\mathbf{g}_{\text{grad},1,2}$ – current volumetrical temperature; $\mathbf{g}_{\text{h},1,2}$ – mean surface temperature (increment) – mean surface temperature increment caused by EFI; $\mathbf{g}_{\text{h},1,2}$ – temperature flash on the contact; $\mathbf{g}_{\text{grad},1,2}$ – maximum surface temperatures gradient on normal in the contact.

3. Discussion and conclusions

The computer method developed allows for:

1) an automation of the process of tribological research and its implementation in a mode of dialogue between the researcher-tribologist and a computer program. The program is used to calculate (analytical) prediction of the working (operational) electrical-frictional-wear (EFW) performances of tribosystems. This provides for the possibility to optimize the tribosystems’ parameters of durability or functionality of a friction knot;

2) a methodology of the controlled process of generating, on the surface of the construction material, of the nanocoating layer (i.e. tribolayer) made of the main material or its combination with the counterbody material, through the process of friction.

4. References

Tactility Experience, Information Given Through the Fingers

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1. Introduction

This abstract is concerned with information caused by three fingers touching and sliding over a surface. It reports how roughness and friction give emotional response to product design. The hypothesis is that the three fingers tactile information is of a more gentle kind than the information given by a one-finger touching and sliding test. The probes consisted of eight polymer test plates that were supplied by the company DuPont. A total of sixteen individuals participated in the test, seven females and nine males. Their age varied from 23 to 30 years. The result shows how tactile information is related to multi-touch sliding over a surface. This information is compared to previous tests made on single finger friction and perception tests.

2. Methodology

To be able to objectively measure the normal and frictional force generated between the fingertips of the test participants and the surface, a three dimensional Kistler sensor was used. It is a device that uses the piezoelectric effect to measure forces. The test plates were mounted on the sensor by using adhesive tape. This test set up has previously been used for single finger tests [1-5]. Each person tested was asked to slide ten times over the surface with the index finger in a way that felt natural. The same procedure was then repeated with three fingers, index, middle and ring fingers.

The contact area between the fingertip and the surface was measured by using a thin layer of blue chalk on the test plates that was adhered to the fingertips when touched. The area was calculated when the fingertips were scanned. The relation of the pressure being used by the three fingers was measured using pressure sensitive film. By dividing the measured normal force by the contact area between the fingertips and the surface, numerical values of the average contact pressure could be acquired.

3. Results

The results from the normal force measurements are presented in Figure 1. The left graph shows the results from one-finger and three-finger tests from one of the participants. In the right graph the results for the total population tested and also for gender are presented. The results show that the normal force didn’t differ much between the single-finger and the three-finger test. That means that the force applied by each fingertip was lower during the three-finger test. The difference in average force between the male and female participants was minimal. This is also shown in Figure 2 where the results from the contact pressure measurements are presented for two participants. The results show that the applied pressure was significantly lower when three fingers were used. This strengthens the hypothesis that three fingers tactile information is of a gentler kind than the tactile information given by only one finger.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig1.png}
\caption{Normal force vs. time (left graph) for one-finger and three finger tests from one of the participants. Normal force vs. gender and total population tested (right graph).}
\end{figure}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{fig2.png}
\caption{Pressure measurement The figures to the left and to the right show the results from two test participants with considerably different sliding techniques.}
\end{figure}

4. References

An Experimental Method for Evaluation of Tangential Contact Stiffness
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1. Introduction

Topography of contact interface is one of the most important factors that determine the dynamic behavior of contacts. Control of contact topography is therefore important in controlling this dynamic behavior. On the other hand, asperities interaction on contact interfaces can be quantified using a parameter called contact stiffness. Introducing surface texture to contact interface can therefore control the contact topography, thus controlling the contact stiffness.

In this paper, an experimental method for the estimation of tangential contact stiffness of surface textured contact interface is described.

Fig. 1 (a) Block specimens and (b) contact characteristic of disk specimens.

2. Method

Two groups of specimens are prepared, denoted as block specimens and disk specimens. Block specimens and their contact characteristics are shown in Fig. 1(a). Disk specimens and their contact characteristics are given in Fig. 1(b). The disk specimens are 50 mm in diameter and 20 mm in thickness.

In the case of block specimens, the quantitative value of tangential contact stiffness is related to the occurrence of peak in the frequency response function (FRF) of these blocks when subjected to a small amount of impact. For the case of the disk specimens, the same method is used together with a newly proposed experimental setup shown in Fig. 2. Such relationship between peak frequency and the normal contact stiffness has been reported in Ref. [1].

3. Results

The estimated values of tangential contact stiffness, $K$, of block specimens and disk specimens are summarized in Fig. 3. Those values are analyzed from the shift of peak frequencies of each system as the result of changes in contact condition. The increase of the contact stiffness values can be described using a power law given in Fig. 3. Here, it can be observed that the quantitative values of contact stiffness lie in the vicinity of a line showing the exponential increase of the contact stiffness to a theoretical maximum value, $K_{\text{max}}$, which is the case when contact occurs perfectly across all contact interface; the case of Block III model.

For the two kinds of analyzed contact conditions, the tangential contact stiffness ranges from 2.27 MN/m at normal load of 6 N to 150 MN/m at normal load of 121.5 N for texture T1 contact combination and 2.38 MN/m at normal load of 6 N to 183 MN/m at normal load of 147.1 N for texture T2 contact combination.

Fig. 2 Experiment setup for tangential contact stiffness evaluation

Fig. 3 The estimated values of tangential contact stiffness of contact interfaces with surface texture

4. Conclusions

The results can be concluded as follows:

1. An experimental method for evaluation of tangential contact stiffness of surface textured contact interface has been proposed.

5. References


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Analysis of Lubricants Adsorbed at Solid-liquid Interface Using Infrared-visible Sum-frequency Generation Vibrational Spectroscopy

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1. Introduction

Although boundary layer lubrication is of primary importance in most engineering applications, relatively few techniques are available that probe the surface molecular structure of lubricants in situ when the solid surface is in contact with solution.

The nonlinear optical techniques, such as second harmonic generation (SHG) and sum frequency generation (SFG), are particularly attractive tools to study nondestructively buried interfaces at submonolayer sensitivity.

In this study, infrared-visible sum frequency generation (SFG) vibrational spectroscopy was used to determine the molecular structures of oleic acid (OA) adsorbed on n-alkanethiolate self-assembled monolayers (SAMs) with various end groups.

2. Experiments

SAMs of hexadecanethiol (HDT) and 16-mercaptoundecanoic acid (MHDA) were used as substrate. OA was used as a model for lubricants. SAMs were prepared by immersing the gold substrates into 2.0 × 10⁻⁴ M ethanol solution containing HDT or MHDA for 24 hours or more at room temperature. Afterward, they were rinsed with fresh ethanol.

In SFG, two pulsed laser beams of visible frequency (ω_vis) and IR frequency (ω_IR) are overlapped at a surface. The nonlinear optical effect of SFG results in the emission of light at ω_SFG = ω_vis + ω_IR. The intensity of the SFG light (I_SFG) is proportional to |χ(2)|², where χ(2) is the second-order nonlinear susceptibility. χ(2) is zero in centrosymmetric environments. Therefore, SFG is a surface-specific technique. In this study, SFG spectra were recorded in the C-H and C=O stretching regions.

3. Results and Discussion

Figure 1 shows the SFG spectra observed on air/HDT and OA/HDT interfaces. For HDT-SAM, vibrational modes of the methyl symmetric stretch (r'), Fermi resonance of the methyl symmetric stretch (r'Fermi), and methyl asymmetric stretch (r) appeared strongly in air. This result indicates that the HDT formed closed-packed well-ordered structure on gold substrate. When SFG spectra were recorded at the interface between oleic acid and HDT-SAM, vibrational modes of methylene symmetric (d''') and asymmetric stretch (d') were observed in addition to the resonance related to the terminal methyl group. This indicates the existence of gauche defects and hence a lack of centrosymmetry in the oleic acid hydrocarbon chains.

Figure 2 shows the SFG spectra observed on air/MHDA and OA/MHDA interfaces. In the case of MHDA-SAM, the intensity decreased compared to the SFG observed on HDT-SAM. However, the resonance related to the methylene group (d'' and d') was observed. Therefore, high degree of disorder in this film was indicated because of the hydrogen-bond formation between neighboring carboxylic end group. When SFG spectra were recorded at the interface between oleic acid and MHDA-SAM, the intensity of the resonance related to the methylene group increased. In addition, the intensity of the vibrational mode of single hydrogen-bonded C=O stretching was significantly increased. This result indicates that oleic acid molecules were adsorbed on the MHDA-SAM surface through hydrogen-bonding between their carboxyl groups.

4. Conclusions

Adsorption of OA on SAM surfaces with different chemical properties was studied using SFG. The results indicate that the difference in the chemical properties affected the adsorption of OA.

5. References

A Method to Increase Machining Accuracy in Cutting Process by Using Active Controlled Hydrostatic Bearing

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1. Introduction

Machining inaccuracies are often caused by the alteration of the relative position between the tool and workpiece. This alteration can be produced by the spindle eccentric displacements under the cutting forces during the cutting process by heavy machine tool. The spindle is often supported by hydrostatic bearings in order to rotate smoothly, and its rotation performances are directly affected by the bearing characteristics. So, it is feasible to improve the spindle performances by modifying the bearing characteristics. Some researchers found that the dynamic characteristics of the hydrostatic bearing were improved by changing the hydrostatic pressure\(^1,3\) which is changed by means of active throttle control technology and such bearings can be called Active Controlled Hydrostatic Bearings (ACHB).

2. Mathematical Model

A method using ACHB to control shaft positions to increase machining accuracy is presented in this paper. Active throttle control technology is implemented by hydraulic control system which consists of electro hydraulic servo valves and controllers. Operational principle of ACHB is shown in Fig.1. \(X_i, Y_i\) and \(Y_j\) are restrictors, \(S_x\) and \(S_y\) are servovalves in X and Y directions. The servovalves are controlled by electrical signals, \(I_x\) and \(I_y\), which are generated by neural network adaptive controller according to the journal displacement measurement signals in X and Y directions. The fluid that flows through the servovalves is used to modify the pressure differences between pockets so that significant modification of forces can be obtained.

![Fig.1 Operational principle of active hydrostatic bearing](image)

The dynamic model of ACHB is established with oil-film forces instead of the stiffness and damping coefficients. The dynamic model is as follows:

\[
\begin{align*}
\dot{m}\ddot{e}_x &= \int_{0}^{L} \int_{0}^{\ell}(p(\phi, z, t) \sin \Phi) \Phi d\phi dz + F_x \\
\dot{m}\ddot{e}_y &= \int_{0}^{L} \int_{0}^{\ell}(p(\phi, z, t) \cos \Phi) \Phi d\phi dz + F_y
\end{align*}
\]

where \(m\) is the journal mass, \(L\) is bearing axial length, \(F_x\) and \(F_y\) are loads acting on the journal. The pressure \(p\) is obtained by solving the Reynolds equation as follows:

\[
\frac{1}{R^2} \frac{\partial}{\partial \Phi} \left( \frac{h^3}{12\mu} \frac{\partial p}{\partial \Phi} \right) + \frac{\partial}{\partial z} \left( \frac{h^3}{12\mu} \frac{\partial p}{\partial z} \right) = \frac{\Omega}{2} \frac{\partial \Phi}{\partial \Phi} + V_x \sin \Phi + V_y \cos \Phi
\]

where \(h\) is oil-film thickness, \(\mu\) is the viscosity, \(\Omega\) is the rotational speed. \(V_x\) and \(V_y\) are the velocity of journal in X and Y directions. The fluid pressures in the pockets are calculated from the continuity equation considering the flow rate of the servovalves.

Then the dynamic shaft positions and eccentric displacements under the cutting forces can be calculated according to the dynamic model given in Eq. (1)

3. Results

The numerical simulation results of shaft positions under the action of cutting forces are shown in Fig.2. The spindle supported on conventional hydrostatic bearing has great eccentric displacements, and then the relative position between the tool and workpiece is greatly changed. However, the one supported on ACHB stabilizes near the bearing center, keeps the relative position, and the adjustment process occurs rapidly.

![Fig.2 the dynamic response of spindle](image)

4. Conclusions

In this paper, the ACHB is used to control shaft positions of machine tools. The simulation results show that the ACHB are quick and effective in controlling the shaft positions and can effectively maintain the relative position between the tool and workpiece. Hence this method is feasible for improving machining accuracy.

5. Acknowledgement

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6. References

Transient Processes of Stick-Slip at Dry/Wet Interfaces

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1. Introduction

Real-time observations have been performed using a longitudinal line contact between transparent blocks of polymethyl methacrylate (PMMA). In a previous paper1, the mechanism of precursor events, observed before the global slip, was examined by considering non-uniform loading. This paper describes the effect of lubricant on the transient process from the stick to the slip phase.

2. Experimental details

The longitudinal line contact between a PMMA base (length: 120 mm, width: 30 mm, and height: 40 mm) and a PMMA slider (length: 100 mm, width: 10 mm, and height: 20 mm) in non-lubricated (dry) and lubricated (wet) conditions was employed. The base had a contact surface with RMS roughness of 5 µm. The bottom edges of the slider were cut off along the longitudinal direction, resulting in an apparent contact area with a length of 100 mm and a width of 0.8 mm under a normal load of 400 N.

For the test under a wet condition, polyalphaolefin (PAO) with a viscosity of 150 cSt at 40 °C was applied to the contact surface of the base. After applying the normal load of 400 N, a tangential load was applied to the trailing edge of the slider at a loading rate of 80 N/s. The intensity of the light transmitted across the contact interface was imaged at a frame rate of 92 kHz.

3. Results and discussion

The experimental results under the dry and wet conditions are shown, respectively, in Figs. 1 and 2. The upper graph shows the temporal change of the image of the contact interface \( I(x,t) \); it was normalized using the values of \( I \) at \( t = 95 \) ms and represented using 256 gray-scale levels. The lower graph shows the temporal changes of the total intensity \( I_{\text{total}}(t) \) (i.e., the sum of \( I \) for all \( x \) at \( t \)) and its time derivative \( dI_{\text{total}}/dt(t) \); they were normalized using the values of \( I_{\text{total}} \) and \( dI_{\text{total}}/dt \) at \( t = 0 \) ms, respectively. Note that the origin of the abscissa (i.e., \( t = 0 \) ms) represents the time when \( dI_{\text{total}}/dt \) takes a maximum value caused by the first transition from the stick to the slip phase.

In the dry condition (Fig. 1), a detachment front2 propagating from the trailing edge (\( x = 0 \) mm) to the leading edge (\( x = 100 \) mm) is found. Note that the interface rupture is always initiated at the trailing edge to which the compressive tangential stress was applied. The detachment front propagates with a mean velocity of 800 m/s and reaches the leading edge before the total intensity takes the minimal value at 0 ms.

In contrast, in the wet condition (Fig. 2), the propagating detachment front is not observed, but the rupture nucleation is found within the contact area. It is believed that the lubricant (PAO) creates a number of real contact areas with low shear strength that are ruptured by the shear stress transmitted inside the slider. It should be noted that the growth rate of the nucleation is two or three orders of magnitudes lower than the propagation velocity of the detachment front. However, the nucleation is initiated at a lower tangential load, resulting in smaller static friction by lubrication.

References

Microstructure of Cu Transfer Layers Produced on Carbon Steel by Sliding

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1. Introduction

Material transfer from a counter surface is a well known phenomenon frequently observed for two surfaces rubbing against each other1. In this work, the microstructure of Cu transfer layers produced on carbon steel by sliding was investigated, and the mechanisms of formation of the transfer layers are discussed.

2. Experimental

A common pin-on-disc method was used for the sliding friction tests, and a pure copper pin was rubbed against a carbon steel (0.45 mass% carbon) disc in vacuum (8x10^-4 Pa) at sliding speeds of 0.1-5 m/s with loads in the range from 4.9 to 49.1 N. The number of turns of the rotating disc was varied in the range of 1 to 5000. The rubbed surfaces were observed in a digital microscope and SEM, and the microstructures of Cu transfer layers formed on the disc were examined by optical microscopy and STEM (Scanning Transmission Electron Microscopy) equipped with EDS (Energy Dispersive X-ray Spectroscopy).

3. Results and discussion

Microscopy surface observations revealed that material transfer from a Cu pin to the surface of a steel disc and material transfer from the steel disc to the surface of the Cu pin occurred at the same time during only one turn of the disc. The sizes of the area of the material transfer on the rubbed surfaces ranged from 0.1 to 200 μm. After several turns of the disc, patchy transfer layers of Cu with maximum thickness of 50 μm were formed on the steel disc surfaces.

Figure 1 shows STEM bright-field images of transverse section of the steel disc rubbed against a Cu pin. Nanocrystalline structures with grain sizes of 30-180 nm were identified in the Cu transfer layer. Very fine lamella structures with interlayer distance of 30-50 nm were also observed in the steel disc near the interface of the transfer layer. The ultra-fine structure formation is considered to be due to the very large plastic shear strain caused by sliding friction2. Moreover steel particles with sizes of 0.1-1 μm were observed in the Cu transfer layer, as shown in Fig.2. This suggests that the steel particles that were transferred from the disc to the pin surface were back-transferred together with the transferred Cu during the continued sliding.

4. Conclusions

Nanocrystalline structures with grain sizes of 30-180 nm were produced in the Cu transfer layers, and back-transferred steel particles with sizes of 0.1-1 μm were observed in the Cu transfer layers.

5. References

Friction and Energy Dissipation of Multilayer Graphene
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1. Introduction

The excellent mechanical properties of graphene make it an ideal candidate for NEMS applications where tribological properties are of essential importance. Molecular Dynamics (MD) simulations have been used to study the tribological properties of multilayer graphene. The dependence of friction force on the layer number is investigated under different conditions. The results can be guidelines for the design of NEMS devices based on graphene.

2. Methods

Molecular Dynamics (MD) simulation is performed to investigate the multi-layer effect of perfect AB stacking graphene on the friction force. Each graphene layer has a dimension of about 9.7nm \times 8.4nm, containing 3200 atoms. The bottom layer is held fixed, and part of the upmost layer is driven with a constant translational velocity along x axis. Langevin thermostat is imposed on a few of the edge atoms to control the temperature of the system. A smooth wall pressing down along Z axis applies load to the graphene layer. The intralayer interaction is described with REBO potential, while the interlayer interaction is described with LJ6-12 potential.

In the simulation, a timestep of 1fs is used. The system is relaxed first, then are dynamics run for about 50ps to equilibrate the system. After that, a constant velocity is imposed on the upmost layer to shear the system.

Cases with different velocities, pressures, temperatures are simulated.

3. Results

The displacement curves for each single layers and the lateral force curves exhibit evident stick-slip behavior when the layer spacing is small (Fig.1(a,b)). However, with the spacing increasing, the stick-slip gradually disappears. The main source of friction then is from the momentum exchange between the adjacent layers.

The average friction force of multilayer graphene with different layer number is investigated (Fig.1(c)). It is interesting to find that in all cases, with the increasing layer number, the average friction force increases and converges to a certain value. This is different from some earlier experiment works\textsuperscript{1,2}, which indicate a reverse trend. However, in such experiments the shear sliding does not occur between graphene layers but between the tip and the graphene surface which causes this difference. We speculated that the increase trend is due to the decreasing stiffness of the sliding body with increasing layer number, which also reveals that bulk atoms can affect surface friction\textsuperscript{3}.

It is found that the sliding can occur between different layers which is sensitive to the initial condition. In some of the cases, two sliding interfaces can exist, which causes the average friction force to increase.

The lateral force curve for multilayer graphene with a small layer number (2 or 3) shows some symmetry, but this symmetry is broken with more layers. The symmetry is the reason why the average friction force is ultra low in such graphene, which can imply a superlubricity state.

Fig.1 (a) Displacement of each layer along shear direction of 6-layer grapheme (layer1 is fixed); (b) Lateral force curve; (c) Average friction force curve with different layer spacing (2.8-3.2 Angstrom)

The in-plane stress distribution of graphene layers was investigated. It was found that “stick” is accompanied with a strong in-plane stress imbalance within some layers, which is released when a sudden “slip” takes place, exciting longitudinal vibrations, followed by energy dissipation as the vibration modes rapidly dissipate into thermal phonons.

4. Conclusions

Molecular Dynamics is adopted to investigate the tribological properties of multilayer graphene. Stick-slip is evident with small layer spacing, the average friction force increases as the layer number is increasing for all cases studied.

5. References

Novel Near-dry Machining of Aluminum Alloys by Combination of Atmosphere Control and Lubricant Treatment

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1. Introduction

Near-dry machining has been playing a significant role in environmentally friendly manufacturing. In particular, minimal quantity lubrication (MQL) machining has been recognized as a representative near-dry machining operation. This has been demonstrated through a great number of successful results and has already become common technology in machining of steels.

On the other hand, in high performance machining of aluminum and its alloys, effective lubrication is often necessary to prevent material build ups on the tool and the resultant chip jamming. Such situations are particularly true in environmentally friendly operations with a very small amount of cutting fluids. In order to provide successful lubrication for near-dry machining of aluminum, a novel method, using a combination of atmosphere control and lubricant treatment utilizing water, is introduced.

2. Experimental

As atmosphere control, a specific humidifier was installed en route on the compressed air line, and dry air (DA) or humid air (HA) was supplied to a MQL mist generator. Furthermore, as lubricant treatment, using the vapor bubble condensation method, an optimal MQL lubricant of a synthetic polyol ester with stably dispersed micro-water droplets (MWD) was prepared.

In order to investigate the combination of the atmospheric humidity control and the MWD treatment, end milling of an aluminum alloy was carried out under the cutting conditions presented in Table 1. Because of the side length of the workpiece, one pass of end milling is 160mm long and each cutting test is conducted up to ten passes, i.e. the maximum cutting distance is 1600mm.

Table 1 Cutting conditions of MQL end milling

| Cutting speed | 110 m/min |
| Depth of cut | Axial: 10 mm, radial: 5 mm |
| Feed | 0.1 - 0.3 mm/tooth |
| End mill tool | High speed steel (no coating) |
| Diameter | 10.0 mm |
| Workpiece | JIS* AC8A aluminum alloy casting |

*Japanese Industrial Standard

Based on the inspection of end mill appearance after each pass of cutting, evaluation criteria are defined as follows:
- Excellent (E): no adhesive Al was observed on the tool after the tenth pass of cutting.
- Not good (NG): accumulation of adhesive Al was observed on the tool after the first pass of cutting.
- Good (G): accumulation of adhesive Al was observed on the tool after the second or higher pass of cutting.

3. Results

Table 2 shows the evaluation results of the end milling with various air and MQL supply. As the feed increases, cutting conditions become more sever and, if lubrication is not satisfactory, the more accumulation of adhesive aluminum on the tool is observed. The combination of HA and the MWD treatment has provided the best result in MQL end milling of aluminum.

Table 2 Evaluation results of end milling with various air and MQL supply

| Feed, mm/tooth | 0.10 | 0.15 | 0.20 | 0.25 | 0.30 |
| Dry Air (DA) | E | NG | NG | NG | NG |
| Humid Air (HA) | E | NG | NG | NG | NG |
| MQL+DA | E | E | NG | NG | NG |
| MQL+HA | E | E | E | G |
| MQL+HA+MWD | E | E | E | E | G |

4. Conclusions

The combination of the atmospheric humidity control and the MWD treatment improved the cutting performance of the MQL lubricant. The method could become a useful tool in solving the problems encountered in near-dry machining of aluminum alloys.

5. Acknowledgments

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6. References

Tribological Characteristics of Steel-Brass Couples at Different Humidity in Air and Oxygen Atmospheres

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1. Introduction
A large number of experimental results have demonstrated that ambient humidity has an important influence on the friction and wear behavior of metal materials. In this paper design, installation and debugging of a humidity control device are described. Wear tests of CrNiMo steel against H59 brass were carried out at different humidity in air and oxygen atmospheres.

2. Experimental
The friction and wear tests were carried out on high temperature and controlled atmosphere pin-on-disk tribo-tester (QG-700). The tests were performed at sliding velocity of 5.5m/s, and at a contact pressure of 0.2MPa. The tribo-pairs were placed in semi-closed cavity filled with air and oxygen with different relative humidity in the range of 10% to 90%.

3. Results
The improved atmosphere humidity control device is shown in Fig 1.

Fig. 1 Improved atmosphere humidity control device

The pin on disk model and its dimensions were shown in Fig 2.

Fig.2 pin-disc contact model

Fig. 3 shows the tribological characteristics of the tribo-pairs at different humidity in air and oxygen atmospheres.

4. Conclusion
1. The friction coefficient and friction temperature, the wear rate of CrNiMo steel pin and the brass disk show a decreasing trend with the relative humidity of the surrounding air. The tribological performance is improved. Comparisons with the atmosphere of air, the friction and wear properties of the tribo-pairs in oxygen have shown relatively lower friction coefficient and higher wear rate or friction temperature.

2. The main wear mechanism between the CrNiMo steel/brass tribo-pair is the adhesive wear, decreasing with the environmental humidity increase. It can be concluded that the mild wear occurring at high humidity levels might be due to the inhibition of this wear mechanism by the formation of interfacial layers such as tribo- and oxidation films.

5. References
1. Introduction

The operation of highly loaded machine components is significantly influenced by the topography of rubbing surfaces. Roughness features bring the risk of surface failures due to the local fluctuations in film thickness and pressure or even asperities interactions within elastohydrodynamically (EHD) or mixed lubricated contacts.

Two main approaches have been developed in the effort to understand the changes of initial surface topography within highly loaded contacts. The first approach considers the real surface topography \( e.g. 1,2 \) while the other uses the simplified topography features consisting of sinusoidal features of various wavelength and amplitudes to explain the behaviour of real roughness. \( e.g. 3,4 \)

This study combines two optical measurement techniques – phase shifting interferometry and thin film colorimetric interferometry, to provide initial (undeformed) surface topography and film thickness data within EHD contact, respectively. It enables to obtain amplitude reduction from measured film thickness data and compare it with the theoretical prediction.

2. Experimental method

Experimental study of real rough surface attenuation requires the evaluation of both the undeformed surface roughness and lubrication film thickness. Lubrication film thickness that carries the information about in-contact deformation is measured using EHD optical test rig. The same rig is also used for the undeformed surface roughness measurements to ensure that both measurements are performed on the same ball surface position. A five fringes algorithm and quality guided path following phase unwrapping method were used to obtain surface topography maps. The accuracy of this application was checked by measuring standard height step and was found to be below 1 nm.

Two AISI 52100 steel balls of 25.4 mm in diameter with original isotropic roughness \( (R_q \text{ about 15 nm}) \) and no additional surface treatment were used. The glass disk had 150 mm in diameter and its surface was optically smooth. The elastic modulus of the steel balls was 212 GPa and that of the glass disk was 81 GPa. Maximum contact pressure was about 0.5 GPa. Mineral base oils were used as lubricants.

3. Results and discussion

The summary of the results obtained during the experiments is shown in Fig. 1. The x-axis was divided into segments having width of 0.1. Then the average value was obtained within each segment and non-zero values are plotted. Obtained results are plotted in the form of amplitude reduction curve that gives \( A_d/A_i \) against dimensionless wavelength parameter \( \nabla \) \([4]\),

\[
\nabla = \frac{\lambda}{b L^{0.5}},
\]

where \( \lambda \) is wavelength; \( b \) is half width of Hertzian contact and \( M, L \) are Moes dimensionless parameters. Experimental results are compared with theoretical attenuation curve which for point contacts and isotropic roughness is defined as \([4]\)

\[
\frac{A_d}{A_i} = \frac{1}{1 + 0.15\nabla + 0.015\nabla^2}.
\]

Even though further measurements could provide more details in long wavelength region, the overall agreement with numerically predicted attenuation curve is very good taking into account the fact that lubrication film thickness was measured within real rough contact.

4. Conclusion

Measured data were used for the experimental verification of the attenuation of low-amplitude surface roughness passing through an EHD contact. Obtained results showed not only the qualitative agreement with the general principle described in the previous studies as to differences in the attenuation of short and long wavelengths, but also the quantitative correlation between measured data and numerically calculated attenuation curve.

5. References

1. Introduction
For the past few years, the research on tribological characteristics of spherical plain bearings has aroused the widespread interest. In this paper a bearing with copper grid composite liners and a bearing with PTFE woven liners were tested using a home-made test rig, SEM and EDS, and the wear mechanism under high frequency and heavy load was identified.

2. Experimental
Tests were conducted at room temperature and atmospheric pressure. The oscillation frequencies were 1.2Hz, 1.9Hz, 2.9Hz and 4.8Hz and the oscillating angle was ±10°. Four different loads 3kN, 6kN, 9kN and 12kN were separately applied 2.5×10⁴ times.

3. Results
The friction and wear curves of spherical plain bearings at four loads and two oscillating frequencies are shown in Fig. 1.

![Fig. 1 Variation of friction coefficient with load for different spherical plain bearings](image)

The friction and wear curves of spherical plain bearings at four oscillating frequencies and two loads are shown in Fig. 2.

![Fig. 2 Variation of friction coefficient with load for different spherical plain bearings](image)

4. Conclusion
It was found that there is PTFE film lubrication for the lower oscillating frequency and smaller contact load, and the adhesive wear is the main wear mechanism. As the oscillating frequency and contact load increase, the wear mechanism of bearing changes from the adhesive to abrasive wear. At high oscillating frequency and high contact load, there is extrusion and evidence of both abrasive and adhesive wear is present.

The change of the wear mechanism for the bearing is related to the PV value. When the PV value increases the friction temperature rises. This will affect the mechanical properties and tribological performance of the liner material of the spherical plain bearing, resulting in increase of bearing wear.

5. References
Effect of Surface Texturing of Bearing Materials on Tribological Characteristics of Ultra-High Molecular Weight Polyethylene Used in Joint Prostheses

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1. Introduction

Wear debris from polyethylene insert of joint prostheses can induce tissue reaction and joint loosening. Newly-developed surface texturing on hard bearing material made of Co-Cr-Mo alloy was proposed in order to extend the service life of the joint prostheses. Submicron grooves and dimples on superfine surface of Co-Cr-Mo alloy disc could trap the lubricating liquid and separate effectively the two mating surfaces.

2. Experimental

Pin-on-disc friction tester capable of multidirectional motion was used to investigate the influence of the textured surface profile of a Co-Mo-Mo alloy disc on the tribological behavior of an ultra-high molecular weight polyethylene pin (Fig.1). Six types of surface profiles of the disc were prepared (Fig.2). Artificial synovial fluid was used as a lubricating liquid, and contact pressure of 1.2 MPa and sliding speed of 12.12 mm/s were applied for total sliding distance of 15 km.

3. Results and Discussion

The influence of disc surface roughness on the wear was not observed (Figs.2 and 3: A-C). The finished surface might increase the real contact area between the pin and disc, so that the adhesion wear mechanism might be activated. The submicron grooves or dimples increased wear, because abrasive wear mechanism could be activated by the increase in the apparent surface roughness (Figs.2-3: D-E). However, submicron size groves and dimples on superfine surface resulted in the reduced wear. It seems that the disc textures might trap the lubricating liquid, improving the lubrication and the separation between two bearing surfaces.

References

Surface Nanostructuring of Aluminium Alloys for Enhanced Wear Resistance

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1. Introduction

Ultra-fine grained or nanocrystalline metals with enhanced mechanical properties can be readily produced by severe plastic deformation (SPD) techniques. Nanocrystalline layers have been reportedly formed on the surface of bulk metals by a family of impact-based techniques (for example [1]). Such treatments have been performed on a number of alloys and have been reported to improve surface mechanical properties and fatigue resistance. This paper reports progress into the surface nanocrystallisation of aluminium alloys to enhance their tribological properties.

2. Procedure

Surface impacts were generated by oscillating an evacuated cylinder along its longitudinal axis so that balls therein impact the workpiece(s) that make up the flat ends of the cylinder, Fig 1. The frequency range studied was from 10 to 50 Hz.

The investigation comprises three stages. Firstly a model of a single ball between two oscillating plates was developed to study the effects of key experimental parameters on the impact velocity and frequency. Secondly a miniature apparatus was constructed to verify the results of the model and to study the effects of impact velocity and frequency on the surface microstructure of small workpieces. Finally a larger apparatus will be built to create surface modified layers on larger samples for tribological testing.

3. Results

The impacts of the ball on the top plate, as revealed by the model showed rich dynamic behaviour, Fig 2. For the system studied here however a wide range of parameters gave rise to an impact frequency equal to the oscillation frequency (other regimes included one third of the oscillation frequency and twice the oscillation frequency). Within this range the impact velocity varied linearly with oscillation frequency (fixed amplitude and inter-plate distance). Increasing the inter-plate distance also increased the impact velocity but decreased the frequency range over which the linear relationship was observed. Increasing oscillation amplitude also increased impact velocity but for a mechanically-driven system this parameter is more difficult to vary than oscillation frequency.

4. References

Sliding Friction by Liquid Meniscus Bridge Under Shear

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1. Introduction

As micro machines and magnetic storage devices, devices get smaller their performance expectations become very demanding. Control of friction, capillary and viscous forces becomes a critical issue in these devices. A liquid meniscus bridge can form between two bodies when a liquid film, droplet or condensed water from humid air is present on a surface. The meniscus bridge causes a strong interaction that heavily affects the operation of micro/nano devices1. Intense studies have been carried out to investigate the attractive force produced by the meniscus bridge between two surfaces. In these studies effect of the normal component of capillary force generated by the meniscus bridge was investigated. We focused on the lateral component of this force2,3. The deformation process of the meniscus bridge was observed experimentally while the lateral component of capillary force was measured as a frictional force.

2. Experiment

The liquid meniscus bridge forms between an optically flat plate and a hemisphere of 103.8mm radius of curvature. The plate and the hemisphere are made of BK7 glass. The flat plate can be moved using an automatically operated micrometer stage. The frictional force acting on the hemisphere is measured from a deflection of a parallel leaf spring via a capacitive displacement sensor. The spring constant is 408N/m. The sensor resolution is 5nm. A CCD camera observes the deformation process of the meniscus bridge.

3. Result

The deformation process of the meniscus bridge under shear is shown in Fig. 1. Dashed lines represent positions of a leading and a trailing edge on the flat plate under shear (t=90s). These lines showing the edges move with the movement of the flat plate for about 30s. By using an image-processing technique, movements of edges can be tracked, as shown in Fig. 2. TR and BR in the legend of Fig. 2 mean the edge positioned Top-Right and Bottom-Left of the meniscus bridge respectively. From this figure, edges on the flat plate move with the plate for 20s. After that, the movement of edges is terminated. In other words, an interfacial slip between the bridge and the flat plate occurs. Fig. 3 clearly shows that the frictional force linearly increases before the slip occurs.

4. References

The Flowing Behavior and Lubricating Mechanism of Emulsion

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1. Introduction

Oil-in-water emulsions have been widely used as lubricant in manufacturing processes. To understand the emulsion lubrication mechanism it is necessary to understand the oil droplets behavior on the solid surface and flow behavior of the emulsion first [1]. Until now, only few results, based on the direct observation of emulsions in contacts, have been reported while the droplet behavior has been studied mostly in line contacts [2, 3].

In this work, the flow behavior of the emulsion droplets under point contact was studied. Spreading of emulsion droplets on solid surface with the oil pooling before the contact region was confirmed, and significant side flow of droplets was observed.

2. Experimental

Deionized water was used as the continuous phase liquid. The paraffin oil was used as the to-be-dispersed phase. The mixture of Span 80 and Tween 80 with hydrophile lipophyllic balance (HLB) value of 9.65 was used as emulsifier. An EHL interferometry test rig was used for the experiments. A contact was formed between a highly polished steel ball and a glass disc covered with semi reflective layer of chromium. The light interference microscope with vertical incidence and a monochrome camera were attached to the test rig. The load on the ball was kept a constant level of 30N. Experiments were conducted at a room temperature of 25°C.

3. Results and Discussion

Under a rolling speed of 5mm/s, two kinds of droplets in the emulsion flow, i.e. merging droplets and bypassing droplets, can be distinguished, as shown in Fig. 1. Only a few droplets contribute to the formation of the oil pool at the point contact, i.e. most of the droplets escape from the contact area.

Effect of different rolling speeds on the formation of oil pool was investigated by measuring oil pool spreads as shown in Fig. 2. The spread of oil pool decreased with the rolling speed. Based on the concept that the oil pool contributes to the film formation in the contact area, the decline of the film thickness measured in the contact area would be caused by the disappearance of the oil pool ahead of the contact.

Effect of droplet size has also been investigated. The emulsion with good stability results in an oil pool smaller than 60μm even at low speeds. However, the emulsion with poor stability results in much bigger oil pool over the range of speeds tested, which well agrees with the film formation. The extents (spreads) of oil pools were observed to increase with the concentration oil in the emulsion.

4. Conclusions

The emulsion flow in point contact has been investigated. Two types of droplets have been observed. The extent of oil pool was found to be strongly affected by the droplet size, oil concentration and the rolling speed. The size of oil pool is related to the film formation ability.

5. References

Formation and Wear Mechanism of ZDDP-Derived Reaction Layers in Rolling-Sliding Contacts

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1. Introduction

Extreme-pressure (EP) and antiwear additives (AW) control the performance of lubricants in the mixed and boundary lubrication regimes. The additives form protective reaction layers on the wear track, however, base oil and operating conditions influence the topography and properties of the layers [1]. The formation of ZDDP-derived reaction layers and their wear mechanisms determine the tribological properties exhibited by those layers.

2. Materials and Methods

As base oil, a fully synthetic non-polar oil, poly-α-olefin (PAO) was used. A fully formulated primary zinc dialkyldithiophosphate (ZDDP), with 99% purity, is employed in simple solution in both base oils without other additives present.

Macro-tribological tests

The tribotests were carried out using ball-on-disc test rig at an applied load of 300 N which resulted in a maximum Hertz contact pressure of 1.9 GPa (contact diameter 540 µm) at a SRR = -10%. The temperature was set constant at 90°C for all the tests. The specific film thickness or lambda ratio (the ratio of the central film thickness to the composite surface roughness of the two surfaces in contact) was set constant at 0.4 and the entrainment speed is set accordingly to 0.25 m/s. The system is operating in the boundary lubrication regime.

Spacer layer interferometry was used to monitor reaction layer formation.

Nano-tribological tests

AFM and LFM images of the wear track after the tribotests were obtained with AFM MFP-3D (Asylum Research, Santa Barbara, CA) in ambient condition. Silicon nitride Veeco tips on a triangular cantilever V shaped with low spring constant were used for nanotribological tests (friction measurements). Silicon tips on aluminium coated cantilever (OLYMPUS OMCL-HA) were used to perform nanowear tests.

3. Results and Discussion

The results of the layer characterization after the tribological tests are summarized in Table 1.

<table>
<thead>
<tr>
<th>Sample</th>
<th>Rubbing time [s]</th>
<th>Friction coefficient, µ</th>
<th>Wear track width [µm]</th>
<th>Reaction layer thickness [nm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>ZDDP1</td>
<td>5 min</td>
<td>0.086±0.002</td>
<td>540±14.4</td>
<td>13.3±2.3</td>
</tr>
<tr>
<td>ZDDP2</td>
<td>15 min</td>
<td>0.092±0.001</td>
<td>503±7.2</td>
<td>53.3±1.2</td>
</tr>
<tr>
<td>ZDDP3</td>
<td>30 min</td>
<td>0.089±0.002</td>
<td>508.3±2.6</td>
<td>70.8±2.1</td>
</tr>
<tr>
<td>ZDDP4</td>
<td>60 min</td>
<td>0.083±0.005</td>
<td>574.3±2.5</td>
<td>79.4±1.2</td>
</tr>
</tbody>
</table>

Wear measurement were performed in the reaction layers formed by scratching 40 times using a silicon tips on aluminium coated cantilever having very high spring constant. Wear amount was estimated by determining the average depth and width of the scratched area. The results indicate that the ZDDP-derived layer is initially softer, undergoing a hardening process with rubbing time.

4. Conclusions

ZDDP-derived reaction layer thickness measurements performed using Spacer layer interferometry method in rolling-sliding contacts, show how initially a thick layer quickly develops on the rubbing surfaces, and then stabilizes at a ‘limiting thickness’ of approximately 70 nm.

The topography of the layer at different rubbing times, analyzed using AFM, evolves from an initial slight coverage of the surface, to the growth of a thick layer over the wear track. The ZDDP-derived reaction layer after 15 and 30 min test is relatively soft hence pull off forces are high and wear test shows high wear volume. Higher adhesive properties of the soft layer are caused by the attachment of the ZDDP molecules to the tip. When rubbing progress further, the additive-derived layer experiences a constant roughening and hardening with rubbing time. These processes may be responsible for the observed increase in friction and wear protection respectively with rubbing time.

5. References

Nanocomposite Coatings for Contact Brush Applications

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1. Introduction

Metal-graphite composite brushes sliding against Cu or brass metal surfaces are commonly used in current- and signal transmission applications. Graphite is used as it is a good conductor and a well known solid lubricant. Metals are added for better conductivity. The accuracy of the signal transmission is of highest priority but the wear of the system is also of great importance as less wear will increase the life time of brushes.

According to Shobert1 the wear of a brush system is mainly due to the sliding motion. As shown by Rabinowicz and Chan in 19802, the wear rate of a metal-graphite brush system largely depends on the counter material at high sliding speeds. They concluded that to obtain low friction a graphite film on the counter surface is needed, but this would wear severely the brush itself. They suggested that to get low wear, limited solid solubility between graphite and counter material is desirable.

In this study we investigate the possibility to lower the wear by using a Ti-Ni-C coating as a counter material against a metal-graphite composite.

2. Experimental

The test apparatus used has a rotating cylinder of metal-graphite composite with tracks for loaded springs and a diameter of 0.1 m. During this study no current was carried as only the influence of spring pressure on mechanical wear was of interest. The sliding speed was set to 10 m/s and the counter materials investigated were spring steel, brass and a coating of Ti-Ni-C on spring steel. All experiments were run in ambient temperature and humidity.

The total number of revolutions was 20 million. Every 5 million revolutions the test was stopped and 3D replicas of the tracks in the cylinder were made. After 20 million revolutions the springs were removed and analyzed using SEM and EDS.

3. Results

At a spring load of 2 g no wear of the metal-graphite was measurable. Fig. 1 shows an SEM image of the brass spring after 20 million revolutions at the load of 2 g. It can be seen that the brass has been worn.

On the contrary, there is no detectable wear of the steel spring and Ti-Ni-C coated spring at this load. EDS analysis showed adhered metal from the brush on both the steel spring and the Ti-Ni-C coated spring. This also showed that the coating was no longer intact.

At 20 g there was considerable wear of the metal-graphite that had run against both the steel spring and the Ti-Ni-C coated spring, but still no wear was measurable against brass, see Table 1.

<table>
<thead>
<tr>
<th>No. of rev.</th>
<th>Brass</th>
<th>Steel</th>
<th>Ti-Ni-C</th>
</tr>
</thead>
<tbody>
<tr>
<td>5·10^6</td>
<td>N.M.*</td>
<td>30 mm^3</td>
<td>40 mm^3</td>
</tr>
<tr>
<td>10·10^6</td>
<td>N.M.*</td>
<td>40 mm^3</td>
<td>70 mm^3</td>
</tr>
<tr>
<td>15·10^6</td>
<td>N.M.*</td>
<td>60 mm^3</td>
<td>110 mm^3</td>
</tr>
<tr>
<td>20·10^6</td>
<td>N.M.*</td>
<td>80 mm^3</td>
<td>170 mm^3</td>
</tr>
</tbody>
</table>

*Not measurable

The wear rate seems to be higher for the Ti-Ni-C coated spring than for the steel spring. When imaged using SEM and analyzed using EDS again, no visible wear but adhered material was found. Fig. 2 shows adhered material on the steel spring at the load of 20 g.

4. Conclusions

The brass is sacrificed against the metal-graphite while for the steel spring and Ti-Ni-C coated spring the metal-graphite wears and adhered material can be seen on the springs. It seems that too much adhered material promotes adhesive failure of the coating which might be a reason for the high wear rates against Ti-Ni-C.

5. References

Experimental Apparatus for Film Thickness and Friction Measurements on Gear Teeth and Cam-Follower Contacts

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1. Introduction

Non-conformal contacts, such as those between gear teeth or cam and its follower, work under transient elastohydrodynamic lubrication conditions. As load, speed and geometry can vary both numerically and experimental simulation of these contacts are difficult. Experiments can validate numerical results and furnish useful indications for design optimization of gears and cam systems.

In this work, a new test rig, purposely designed for the studies of non-conformal, lubricated contacts, is presented.

2. Experimental Apparatus

From a literature review on theoretical/numerical and experimental investigations directed towards the studies of non-conformal contacts¹, several new different designs of a new experimental apparatus for the simulation of gear teeth and cam-follower contacts have been considered². The final version of the apparatus has been developed based on the simulation results performed to investigate the dynamic behavior of the test rig using different cam profiles.

The test rig is able to measure instantaneous contact forces and film thickness between a specimen of suitable shape (usually a cam) and a counterface (usually a flat surface of a disc, simulating a follower). Special attention has been paid to the versatility of the apparatus. Different configurations are possible with this rig, depending on the motion of the cam's axis. The fundamental parts of the apparatus in its basic configuration, with moving cam axis and fixed follower, are shown in Fig.1.

![Fig.1 3D view of the test rig.](image)

3. Instrumentation

Film thickness and contact forces are the main quantities measured by this apparatus. Film thickness and its shape are estimated using optical interferometry. This measurement is possible when the basic configuration shown in Fig.1 is used; the cam with moving axis is in contact with a fixed glass follower (Fig.1-A). Interference images are recorded using a microscope connected to a computer controlled high-speed camera. A computer controlled guide system is used to move the microscope along xyz axes. All components of dynamic contact forces are measured by the system of load cells. An air-cooled ceramic support is fitted between the lubricant’s chamber and the load cells to provide thermal insulation. A real-time controller and data acquisition system are used for managing the operating conditions and data recording.

4. Power Transmission System

The specimen is driven by an electric motor. Motion of its axis is allowed by a rocker arm, where a planetary gear system is mounted. The system is able to provide a 1:1 transmission ratio between the motor and the cam-shaft (Fig.1-B). Special attention has been paid to limit the dimension of the moving components in order to keep inertia forces as low as possible. The velocity is measured by the motor encoder and controlled by a VFD (Variable Frequency Driver). An additional optical encoder, placed after the elastic joint, is used for a more precise speed measurement. The load is applied through a lever system (rocker-arm). Calibrated springs can be inserted between the rocker and a structure designed. A visco-elastic block connected to another structure can limit undesirable excessive oscillations of the rocker.

5. Conclusions

An experimental apparatus for the simulation of gear teeth and cam-follower contacts, able to measure film thickness and contact forces, has been designed and developed.

Versatility and modularity are the main aspects of the test rig described in this work. The apparatus will be used for the analysis of non-conformal contacts working under transient conditions. It could be used in cam profile optimization and investigations of materials and surface characteristics of cams and gear teeth in order to limit energy losses and reduce wear.

6. References

Scratch Testing of WC/Co Hardmetals

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\textsuperscript{2)} National Physical Laboratory, UK
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1. Introduction

WC-Co hardmetals are often used in applications where wear resistance is important. One simple method of determining wear resistance of WC-Co hardmetals is through scratch testing wherein a spherical diamond indenter with known radius is pressed and dragged across the surface of the hardmetal using a known load. In this paper, two grades of WC-Co hardmetals were tested. Sample A had 7 wt\% cobalt content and 1.1 $\mu$m average WC grain size and sample B had 12 wt\% cobalt content and 6.4 $\mu$m average WC grain size. Both materials subjected to macro-scratch and micro-scratch tests. These tests provided information on the friction coefficient of the sample at different loads. Images of the scratched surfaces were analysed using SEM and Optical Confocal 3D microscopy to give information on the damage mechanisms involved.

2. Experimental

Two scratch test facilities were used in this study: 1) a macro-scratch test system using a load of 30 N with a 200 $\mu$m radius diamond indenter, and 2) a micro-scratch test system using a load of 50 mN with a 10 $\mu$m radius diamond indenter. The scratch loads were held constant with a scratch length of 5 mm for all tests. Samples of 8 mm x 6 mm x 5 mm were prepared, polished and annealed in a vacuum furnace at 800 $^\circ$C for 30 minutes. Analysis of the scratched surfaces was carried out using both Supra SEM and an Olympus LEXT confocal microscope.

3. Results

Figure 1 shows the friction coefficient results obtained from the macro-scratch and micro-scratch tests, respectively. The friction coefficient at higher loads was between 0.15-0.17 whereas at lower loads, it was between 0.20-0.25. Fluctuations were also observed which are attributed to a different response from hard WC grains and soft binder phase. These fluctuations are more prominent in sample B for both tests. Confocal microscopy 3D images of the macro-scratches (Fig. 2) show clear detail of the mechanisms that occurred during scratching. Transverse cracks on the scratched surface are visible on both samples with grain cracking and formation of slip bands being more prevalent around the scratch mark for sample B. Details of the material pile up at the edges and the relationship of damage to microstructure are also seen. The 3D images also enabled measurement of the scratch width and depth which are summarized in Table 1.

![Sample A, macro scratch](image1) ![Sample B, macro scratch](image2)

![Sample A, micro scratch](image3) ![Sample B, micro scratch](image4)

Table 1. Scratch width and depth measurements summary.

<table>
<thead>
<tr>
<th>Sample</th>
<th>Width, $\mu$m</th>
<th>Height, $\mu$m</th>
</tr>
</thead>
<tbody>
<tr>
<td>A at macro-scratch</td>
<td>66</td>
<td>1.2</td>
</tr>
<tr>
<td>B at macro-scratch</td>
<td>84</td>
<td>2.1</td>
</tr>
<tr>
<td>A at micro-scratch</td>
<td>2.56</td>
<td>0.068</td>
</tr>
<tr>
<td>B at micro-scratch</td>
<td>3.28</td>
<td>0.19</td>
</tr>
</tbody>
</table>

4. Conclusions

Scratch tests are a fast and reliable way to obtain information on the mechanical properties, the coefficient of friction, the damage mechanisms and microstructural features of a scratched surface. SEM and Confocal 3D images of the scratched surface show that damage is more evident on sample B – the measured scratch width and scratch depth were larger and there was greater pile up near the edges of the scratch. Intergranular as well as intragranular cracking and slip bands on the WC grains were also visible on both samples but were more evident on sample B. The coefficient of friction versus distance also showed higher fluctuations on sample B (with larger grain size) due to a difference in the friction response of the hard WC grains and soft cobalt binder phase.
Stochastic Modeling of Wear Surface from In-situ Monitoring Results

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1. Introduction

In-situ monitoring results of wear and friction surfaces give us quite interesting information to understand the surface fracture mechanisms. Usually it is difficult to achieve the observation and analysis because friction occurs between solid surfaces, often even in the presence of a lubricant. To control a tribological system, it is important to determine how the wear surfaces change over time. In many methods, the surfaces are observed only after tribological tests are completed and the test piece is removed from the tribo-tester.

2. Experiments and analysis

To overcome these limitations, we have developed an observation tool using laser-strobe technique that records surface photographs of the same region in the wear surface during each cycle for metal-metal contact with lubricant at several hundreds Hz reciprocation.

The recorded wear surface photographs, shown in Fig. 1, were analyzed by pattern matching methods and correlations between the images were determined. Characteristic patterns corresponding to friction track sizes and shapes were identified and used to perform the correlation calculations.

3. Modeling

A model was also proposed to understand the results of the friction and wear tests. The model assumed that the surface consists of small domains and that the friction patterns are generated by the interaction of these domains. The domains were defined to be removed by a certain probability, $P_1$, and adhered on the surface by the other probability, $P_2$. Finally the domains were removed from the wear surface with neighborhoods. This probability was defined as $P_3$. The removing probability, $P_1$, takes into account all bonding forces of the domains, Young’s modulus, and the other crystal properties of the material.

The adhesion probability, $P_2$, includes wear particle adhesion and surface transformation corresponding to the domains. Large scale defect generation probability, $P_3$, includes the number of neighborhoods, friction time, removed times, and adhered times. These probabilities and number of neighborhoods domains were the parameters of the proposed model. The typical results of the calculation are shown in Fig. 2.

![Fig. 2 Calculated wear surface transformation. The number indicates the percentage of test time.](image)

4. Summary

Friction and wear model concept was proposed based on the reversible tile which was considered from the results of in-situ monitoring of the friction and wear surface. This proposed model describes the surface transformation by three types of probabilities. The model structure was evaluated using certain probabilities and calculated for the surface patterns. The results of the calculations of the wear surfaces were similar to the observed surface patterns. These results indicated that the proposed model might be suitable to describe the real wear surface transformations.

5. References

Effect of Surface Texturing Dimensions on Tribological Properties under Plane Contact

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1. Introduction

The surface texturing technique has been reported for the purpose of modification of tribological properties for several kinds of machine elements1, and the optimal dimensions of texturing have been discussed2. However, evaluation of texturing dimensions is difficult using only conventional tribotesters because each machine element has a different contact area. The objective of our research is to develop the evaluation method of tribological properties on textured surface adaptable to various contact areas. This paper describes the effect of dimensions of specimen and surface texture on tribological properties by experimental approach.

2. Experimental

Figure 1 shows the experimental apparatus to evaluate the tribological properties using wide specimen (82 x 82 mm) of parallel plane surfaces. The friction coefficient and the film thickness were measured under varying the sliding velocity. Table 1 shows the dimension variation of texturing.

3. Results and discussion

Figure 2 shows the effect of aspect ratio (depth /diameter) on Stribeck curve. When the aspect ratio was higher than 0.02 (Fig.2a), the textured surface reduced friction coefficient compared with the flat surface in fluid lubrication regime. On the other hand when the aspect ratio was higher than 0.02(Fig.2b), textured surface reduced friction coefficient in fluid lubrication regime and mixed lubrication regime. These results indicate the oil retention effect of dimple pattern in mixed lubrication regime is affected by not only capacity of dimple but also the dimensions. The friction reduction in fluid lubrication regime was caused by the increasing film thickness.

We also characterized the friction of textured surfaces with different contact areas. The dimple pattern reduced friction in mixed lubrication regime both in a wide specimen and in a small (15x15 mm) one. However, the groove pattern reduced friction only a wide specimen. These results indicate that the dimple pattern is more effective in various contact areas.

4. Conclusions

The sliding tests of textured surfaces were carried out and the results lead to the following conclusions:
1. The oil retention effect is affected by the dimensions.
2. The dimple pattern was adaptive to various contact areas better than the groove pattern.

5. References

Stabilization of Hydrogen Bonding in Polypropylene Glycol at EHL Contact Region

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1. Introduction

At a lubricating contact, pressure and temperature increase, and a shearing is imposed to lubricant. Then, the molecular response under these extreme and dynamic conditions relates to the lubrication performance. In-situ observation with an infrared spectroscopy has been used to detect chemical information on the lubricant molecule directly at the contact, such as concentration of lubricant components, molecular orientation and so on. Polyalkylene glycols are known as water-soluble lubricants, which contain ether and hydroxyl groups. These polar functional groups interact with each other to form hydrogen bonding. This molecular interaction relates to the viscosity index and rheological properties of aqueous solutions of polymers.

In this study, we investigated the hydrogen bonding of polypropylene glycol under EHL contact by means of in-situ observation with micro-FTIR, and discussed the effect of pressure on stabilization of hydrogen bonding.

2. Experimental

Sample oils were polypropylene glycols (PPG) with molecular weight of 1000, 2000 and 3000 (As received). In-situ observation was performed to obtain IR spectra of oil film under EHL conditions. This apparatus consists of micro-FTIR and a disk on ball type lubrication tester. A steel ball was rolling against an optical disk made of sapphire. The applied loads were set at 4.9N, 9.8N and 14.7N, corresponding to a maximum Hertzian contact pressure of 0.51GPa, 0.64GPa and 0.74GPa. The entrainment speed was 0.27 m/s under a pure rolling condition with zero of slide/roll ratio.

For the measurement of infrared spectra under high pressure and temperature conditions, a diamond-anvil cell (DAC) with a thermal cell was used. In this study, test conditions were: pressure range from ambient pressure to 4GPa and temperature up to 100 degrees.

3. Results and discussion

Figure 1 shows the profile of the peak position of C-H and O-H stretching modes under EHL condition. From the result, it is clear that the peak position of the C-H stretching mode shifts slightly to a higher wavenumber at the center of Hertzian contact, compared to that obtained outside of Hertzian contact. The frequency shift was explained by a steric effect of neighboring molecules induced by formation of closer packing with lower volume under high pressure. In contrast, it was found that the O-H stretching mode shifted clearly to a lower wavenumber at the Hertzian contact. These peak shifts show a tendency to increase with increasing applied load. The result suggests that the stabilization of hydrogen bonding of PPG was caused by the effect of contact pressure at the Hertzian contact region.

The effect of pressure on the hydrogen bonding was confirmed using DAC. It is clear that the pressure effect is dominant on the stabilization of hydrogen bonding under EHL condition. The peak shift of the O-H stretching mode is dependent on the molecular weight. The average slope of the red-shift of the O-H stretching mode by the pressure was steeper in the case of lower molecular weight. This dependence of the slope with molecular weight implies that a short chain molecule has a higher concentration of hydroxyl groups.

4. Summary

1. Hydrogen bonding of polypropylene glycols was stabilized dynamically by high pressure of the Hertzian contact region.
2. The stabilization of hydrogen bonding by pressure was dependent on the molecular weight of polypropylene glycol.

5. References


Acknowledgements

We are grateful to Prof. Eiji Ohtani and Dr. Takeshi Sakai of Tohoku University for technical support of diamond-anvil cell.
Experimental Study on Preventing Web Defects under Transportation by Concaved Roller

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1. Introduction

In a production line, long flexible materials, such as paper, photographic film, thin magnetic tapes and optical film for liquid crystal displays, are called webs, and are transferred by traction between the web and a roller. To improve the production efficiency, high-performance webs that are thinner and wider without losing the established high-quality functions are desired. Thus, it is significant to prevent scratches caused by slippage between the web and the roller under the condition of low tension, and wrinkles on the roller caused by high tension. To solve this trade-off-relation problem, micro-groove rollers are used in the production line. However, the wrinkling of the thin web on the roller occurred easily even when transported under very low tension. To avoid wrinkles, it is well known, through experience, that the use of concave rollers is effective. In this paper, experiments on preventing wrinkles by using concave rollers compared with cylindrical rollers are described. Wrinkle generation is examined using the following four test rollers: a cylindrical roller with micro-grooves, a concave roller with a diameter that increases towards the two ends, and a concave roller with micro-grooves. Moreover, the relationship between the use of a concave roller and wrinkle generation is clarified by comparing all experimental results.

2. Experiments

Figures 1(a) and 1(b) show the dimensions of the cylindrical and concave rollers used in the experiment, respectively.

![Fig.1 Dimensions of the cylindrical and concave rollers](Image)

Figure 2 shows the transport test apparatus used in this experiment. In this apparatus, one end of the web was spliced to the other end after passing along five rollers. This web loop enables the endless transport of the web by mean of friction between the web and driving rollers. The roller surface velocity was controlled with a motor drive controller. The web tension was changed by moving the tension control roller up or down and was detected by load cells attached at the bottom of the roller. Its value was displayed by a tension pickup. In the experiment, the roller surface velocity and web tension were intermittently adjusted. The misalignment angle of the test roller was gradually increased using a micro-screw, and the web tension and misalignment angle were recorded when shear wrinkling occurred.

In this study, the following four types of rollers were used: a cylindrical roller with a uniform axial diameter without micro-grooves, a cylindrical roller with micro-grooves, a concave roller with micro-grooves, and a concave roller with micro-grooves. Figure 3 shows the result when the roller surface velocities were 0.15 m/s. In these figures, the solid lines represent the theoretical wrinkling onset tension the dashed lines represent the trend obtained in the experiment, and the dot-dash lines represent the slip onset tension in the theoretical estimation. As shown in Fig. 3, the wrinkling onset tension when using the concave roller with micro-grooves was higher than that when using the micro-grooved cylindrical roller for both roller surface velocities. The extension of the web in the cross machine direction owing to the shape of the concave roller with micro-grooves increases the wrinkle onset tension, similarly to the case of the concave roller without micro-grooves.

Therefore, the use of the concave roller with micro-grooves prevents slippage and wrinkling during transport of the web, enabling a marked improvement in productivity.

![Fig. 2 Experimental apparatus](Image)

![Fig. 3 Wrinkle region for PET12 with cylindrical and concave rollers with micro-grooves (U_r=0.15[m/s])](Image)
Low Friction Sliding by Various Kinds of Metal-Doped DLC/DLC Contact

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1. Introduction
DLC films have attracted attentions for use as modified surfaces with low friction and high durability against wear. They have been widely applied to many types of machine components. However, DLC films still have some characteristics that need to be improved, including low heat resistance, poor adhesion, etc. A few types of metal-doped DLC films have been developed in response to these problems. For example, silicon-doped DLC and titanium-doped DLC are superior to pure DLC from the viewpoint of adhesion and heat oxidative durability. To obtain lower friction coefficients, we investigated the tribological properties of DLC/DLC sliding using combinations of pure DLC film, silicon-doped DLC film (Si-DLC), and titanium-doped DLC film (Ti-DLC) produced by plasma-based ion implantation and deposition (PBIID).

We analyzed the film structures by Raman spectroscopy and evaluated the tribological properties of the DLC/DLC during sliding. Friction coefficients were measured using a ball-on-disk friction tester.

2. Experimental Methods and Results
2.1 Raman Spectroscopic Analysis
We analyzed the structures of DLC films (pure-DLC, Si-DLC, and Ti-DLC) by Raman spectroscopy. Raman spectra of the films are shown in Fig. 1. The pure DLC and Ti DLC spectra exhibited a D-band peak around 1380 cm\(^{-1}\) and a G-band peak around 1530 cm\(^{-1}\) meaning that these DLCs were typical amorphous. The Si-DLC film, on the other hand, had only a single peak around 1450 cm\(^{-1}\), called a T-band, attributed to the bond of sp\(^3\) hybrid orbital bond. It is a typical Si-DLC spectral pattern.

2.2 Ball-on-Disk Friction Test
We evaluated the tribological properties of the DLC/DLC sliding from the viewpoints of friction coefficients using a ball-on-disk friction tester. Pure-DLC, Si-DLC and Ti-DLC were deposited on both the disk and the ball. The friction coefficients of the DLC/DLC in sliding are shown in Fig. 2. The Si-DLC/Ti-DLC sliding had the lowest friction coefficient (0.023), while the Ti-DLC/Ti-DLC sliding had the highest (0.048). We found that using the same type of DLC made the friction coefficient the highest in all of the combinations.

3. Discussion
The high friction coefficient between the same types of DLC films sliding was due to doped metals adhesion. It is well known that the same types of metal are more adhesive than combinations of different metals. We demonstrated that this rule also applies to the DLC/DLC sliding. In the sliding of different types of DLCs, adhesion did not easily occur, and thus the friction coefficients became comparatively low. Further detailed surface analyses are necessary to clarify the friction phenomenon in near future.

4. Conclusion
(1) The pure-DLC/Si-DLC sliding had the lowest friction coefficient (0.023), while the Ti-DLC/Ti-DLC had the highest (0.048).
(2) Applying different types of metal-doped DLC films to each sliding surface was considerably effective in achieving lower friction.
Molecular Dynamics Simulations of Alkanethiol Self Assembled Monolayers on Au(111): Effects of Loading and Chain Size on Tribology and Film Structure

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1. Introduction

SAMs are self organized molecular assemblies that can be adsorbed onto a substrate via their head group and produce many different film characteristics. One of their applications is as surface modifiers, which reduce the friction and adhesion of moving mechanical parts in micro/nano-electromechanical (MEMS/NEMS) devices. Alkanethiols self assembled on gold surfaces have attracted great deal of interest as model systems due to their fairly easy preparation in experiments, and also in modelling and simulation studies, due to their fairly simple molecular structure. There is a big body of literature on the characterization of self-assembled monolayers (SAMs) of alkanethiols adsorbed onto gold surfaces by friction force microscopy (FFM). Along with the development of AFM techniques there have been extensive studies to characterize the nano-tribological properties of self-assembled monolayers. Our simulation study aims to gain insight into the behavior of alkanethiols under changing loading, sliding velocity and composition conditions and to understand the effects on their tribological properties.

2. Simulation method

Our system consists of two parallel gold substrates with a layer of SAM attached to the lower substrate. The substrate is a model of Au(111) surface. The lattice constant and the nearest neighbor distance of gold fcc crystal respectively are 0.408 nm and 0.288 nm. Periodic boundaries in lateral directions are applied and each surface consists of 4 layers of (111) gold atoms.

The thiol based chains (CH₃(CH₂)ₙSH) are simulated with a united atom model in which CH₂, SH and CH₃ groups are treated as single interaction sites. We have shown the capability of this united atom model in the studies of n-alkane systems crystallization [1], confinement induced phase transitions [2] and lubrication with gold surfaces [3]. The realistic molecular model includes bond stretching, dihedral and angle potentials. The monolayers are attached to the gold surface where sulphur atoms are positioned in a hexagonal configuration resulting in a surface density of 21.6 Å²/chain, in line with experimental observations.

3. Results

Our simulations are conducted in three stages, which involve equilibration of the SAM layer on the gold surface, the loading stage and equilibration under load, and the final stage where the SAM is subjected to shear by moving the two confining surfaces in opposite directions at a certain velocity (see Fig 1). The friction coefficient is then calculated using the normal load and lateral force. Our loading simulations show that the dodecane SAMs, yield under the normal pressure of 700 MPa. At normal loadings higher than this, the SAM molecules show significant deformation, although structural order still remains within the film. We find that the friction coefficient increases with the shear rate, although this dependence is much stronger at lower normal loads. The friction coefficient decreases with increasing the normal load where more pronounced effects are observed at larger sliding velocities. The stick slip during the sliding can be clearly seen in the friction force behaviour with strong correlations with the SAM tilt and orientation structure. Transient structural changes in the film structure are often accompanied by significant changes in the friction force. Dependence of friction coefficient on the alkanethiols molecular size is also explored. We find strong odd-even effects where the friction coefficient varies depending on the number of carbon atoms being odd or even.

Fig.1 A dodecanethiol monolayer attached on a Au(111) surface, and confined by another Au(111) surface, under shear.

ACKNOWLEDGMENTS

We gratefully acknowledge the support of this study by a University of Sydney Bridging Grant, and an APA scholarship for Leyla Ramin. We also thank the Australian Centre of Advanced Computing and Communications and also Australian National Computational Infrastructure Facility for the generous time allocated for computing.

4. References

Effect of the Substrate Temperature on Photoelectric Properties and Corrosion Resistance of Ti/TiN Multilayer Films

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1. Introduction

Nano-multilayer film is a multi-layer structure with two or more different nano-metre thickness layers of materials produced by alternating deposition. The aim of this study is to produce low resistivity and high corrosion resistance Ti/TiN multilayer films.

2. Experimental

Films were manufactured using CKJ-500D multi-target magnetron sputtering equipment with a Ti target (99.995% purity) and Si(111) single crystals as the substrate. The substrate temperatures were 600, 550, 500, 450 and 400°C, pressure of 5×10⁻⁴ Pa, argon gas flow 30sccm, nitrogen flow 2 ml/s, sputtering current of 0.4A and sputtering pressure of 0.5Pa. The immersion experiments were conducted in HCl with the concentration of 36%.

3. Results

As can be seen from Fig. 1, the (101) peak occurred as a Ti peak whereas the (200) peak occurred as a diffraction peak for TiN with the highest intensity.

Fig. 1 XRD patterns for the Ti/TiN multilayer film prepared at different temperatures

From Figs. 2 and 3, it can be seen that with decrease of the sheet resistance of the thin-film, its infrared reflectance increases.

Fig. 2 Sputtering temperature on the Ti / TiN periodic structure of thin-film conductivity

Fig. 3 Reflection spectra as a function of the temperature of Ti/TiN multilayer films

The corrosion resistance of the Ti/TiN multilayer film is higher than that of the single-layer TiN film. The corrosion resistance of the film is improved as the substrate temperature increases.

Fig. 4 The weight loss of the Ti/TiN multilayer films and single TiN film with different time

Fig. 5 The weight loss of the Ti/TiN multilayer films with different temperature

4. Conclusions

The sheet resistance and surface roughness of the periodic structure thin film decrease with the increase of substrate temperature. Furthermore, with decrease of the sheet resistance of the thin-film, its infrared reflectance increases. The corrosion resistance of Ti/TiN multilayer films reaches the maximum value when the substrate temperature is 600°C.
Performance of a New Aqueous Synthetic Lubricant in Cold Rolling of Stainless Steel

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1. Introduction

Polymer-based aqueous fluids have received considerable attention in metal forming\textsuperscript{1,2}. A new aqueous lubricant containing water-soluble synthetic polymer has been developed which can potentially improve strip surface cleanliness, minimize environmental impact and extend life of rolling lubricant compared with current oil-in-water (O/W) emulsion technology. The polymer is thermo-sensitive as the separation of polymer from water is activated by temperature changes above the cloud point.

The paper aims to investigate systematically the characteristics of new aqueous polymer-based lubricant in terms of friction, lubricity and surface roughness in cold rolling of stainless steel.

2. Experiment, material and lubricant

Experiments were implemented on a 2-Hi reversible rolling mill. The rolling fluid at controlled temperature was sprayed into the roll bite at the entry. Work rolls of 255mm in diameter, surface finish (Ra) 0.15µm in the axial direction and 0.33µm in the circumferential direction were employed.

Samples of 1.909x100x450mm were cut from 304 stainless steel coil, which have a true stress-strain relationship \(\sigma_{\varepsilon} = 310(15.2 \varepsilon + 1)^{0.5}\) MPa. The initial surface roughness (Ra) of the samples is 0.04µm.

The lubricant consists of water-soluble polymer, formulated package for anti-corrosion, phosphorus additive for extreme pressure and distilled water. The cloud point of polymer is around 33-38°C. The lubricant is pre-separated from the fluid by heating above the cloud point.

The paper aims to investigate systematically the characteristics of new aqueous polymer-based lubricant in terms of friction, lubricity and surface roughness in cold rolling of stainless steel.

3. Results and discussion

The experiments were carried out at a constant speed 0.27m/s with a nominal reduction 20%. To evaluate thermal performance of aqueous lubricant, strip samples were pre-heated to 30-100°C, while bulk fluid temperature was maintained at 30 and 40°C below and above the cloud point. Three aqueous lubricants were employed to determine the effect of compositional parameters on its performance, and compared with O/W emulsion. An inverse technique was introduced to calculate the friction coefficient\textsuperscript{3}.

The friction coefficient and strip surface roughness, as a function of strip entry temperature at various bulk fluid temperatures are shown in Figures 1 and 2. It is evident that polymer concentration plays an important role in lubrication performance. The lubricant containing 6% polymer displays the lower friction coefficient than that of 2% polymer. The lubricity can be improved significantly by incorporating 0.5% phosphorus additive for extreme pressure. Due to thermal characteristics of the polymer, the performance of aqueous lubricant is sensitive to strip entry temperature and bulk fluid temperature. The friction coefficient reduces as strip entry temperature is increased. The lubrication effect of the fluid composed of 6% polymer, 6% anticorrosion package and 0.5% phosphorus additive is more pronounced when the polymer is pre-separated from the fluid by heating above the cloud point.

4. Conclusions

The aqueous polymer lubricant has good lubrication capability in cold rolling of stainless steel. The temperatures of strip and bulk fluid as well as the compositional parameter of aqueous lubricant have the significant effects on its lubrication performance.

5. References

Characteristics of Lubricant Pick-up due to Breakage of Liquid Meniscus Bridge

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1. Introduction

In recent magnetic storage systems, the spacing between the flying head and the disk has been dramatically decreased to less than 10 nm to facilitate ultra-high density recording. Under the ultra-small spacing condition, liquid lubricant on the disk is picked up by the flying head slider due to the intermittent contact between the slider and the disk or due to the condensation of the lubricant vapor. The small amount of the lubricant picked-up by the flying head slider will affect the flying characteristics and the read/write performance. In this study, characteristics of the lubricant pick-up due to the breakage of a liquid meniscus bridge are investigated, both experimentally and theoretically.

2. Experiments

The experimental setup is shown in Fig. 1. A liquid meniscus bridge is formed between a glass semispherical surface (radius of curvature $R = 10$ mm) and a glass plate. The glass plate is moved linearly by a piezoelectric stage.

The liquid used as test samples were ethylene glycol, n-hexadecane, Fomblin Z03 and Zdol. The contact angle was controlled by an oil repellent or LB film.

3. Theory

The geometry of a liquid meniscus bridge between a glass sphere and a plane at break point is shown in Fig. 2, where $V_2$ is the liquid volume that remains on the glass surface and $V_3$ is the liquid volume that is picked-up or transferred to the upper spherical surface. These liquid volumes are calculated from:

$$V_2 = \pi \left( \frac{r \sin \phi}{1 - \sin (\theta_2 + \phi)} \right)^2 \left( 2 \cos \theta_2 - \frac{1}{3} \cos^3 \theta_2 - 2 \left( \frac{\pi}{4} - \frac{\theta_2}{2} + \frac{1}{4} \sin 2 \theta_2 \right) \right)$$

$$V_3 = \pi \left( \frac{r \sin \phi}{1 - \sin (\theta_2 + \phi)} \right)^2 \left( 2 \cos \left( \theta_2 - \phi \right) - \frac{1}{3} \cos^3 \left( \theta_2 - \phi \right) \right)$$

The volume fraction of the lubricant pick-up $\alpha_t$ is:

$$\alpha_t = \frac{V_3}{V_2 + V_3} \times 100 \quad (1)$$

4. Results

Figure 3 shows experimental and theoretical results obtained for ethylene glycol. The theoretical results are in good agreement with the experimental results.

5. Conclusions

Characteristics of lubricant pick-up due to breakage of a liquid meniscus bridge have been investigated experimentally and theoretically. It was found that the picked-up volume was affected significantly by the difference in the receding contact angles of two solid surfaces. Theoretical results obtained from a simple model were in good agreements with the experimental results. The model presented is considered to be useful in prediction of the lubricant pick-up characteristics.

6. References


Fig. 2 Geometry of a liquid meniscus bridge between a sphere and a plane at breakage

Fig. 3 Experimental and theoretical results
Effect of Bulk Temperature on the Tribological Performance of Aqueous Symmetric Tri-block Copolymers (PPO-PEO-PPO and PEO-PPO-PEO) Based Lubricant

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1. Introduction

This work investigates the effect of bulk temperature and the tribological performance of a number of symmetric tri-block copolymers on MS1020 commonly in metal forming. It is to determine the best bulk temperature for the lubricants in order to apply in metal forming application.

The tri-block polymers are temperature dependant. The polymers can be completely dissolved in solution since of hydrogen bonding in certain temperature. As temperature increases, the bond is weakened and the polymer break from the water and form dispersion in the water.

2. Materials and method

Four commercial tri-block copolymer from two different structure of polymer, the first are ABA (PPO-PEO-PPO) polymer with molecular weight of 2150 (20% by weight of PEO) and 2650 (40% of PEO); BAB (PEO-PPO-PEO) polymer with molecular weight of 2500 (20% of PEO) and 2900 (40% of PEO) were tested. Phospholan PE65, which is actually extreme pressure and corrosion inhibitor, is added to all of the tri-block for one serial experiment to see the effect of adding additive into aqueous polymer.

Pin-on-disks experiments were implemented under controlled bulk temperature lubricants to evaluate friction and wear loss of lubricated contact of mild steel 1020 (MS) against case-hardened ball as the pin in fully flooded lubrication conditions. Friction and wear depth are monitored online. The bulk temperature is set up at room temperature, which is below the cloud point, and 50°C, which is above the cloud point, for aqueous polymer; 50°C and 60°C for aqueous polymer with phospholan. As for BAB 2900, it is tested at 60°C instead of 50°C because of the cloud point of this polymer is at 60°C.

Sliding speed and duration of the tests were maintained at 0.25ms⁻¹ and 60min. Normal load is set on 12.5 N, the mean effective pressure is 1Gpa with max. effective pressure is 1.5Gpa for the entire test. Surface roughness of all of disk is maintained approximately 0.2–0.3 µm.

3. Result and discussion

Figure 1 and 2 show the coefficient of friction and volume loss from all of experiment. The result with the addition of phospholan shows that the COF and volume loss are significantly low compare to the other lubricant. However, increasing temperature until 60°C is slightly reducing the performance of the lubricants that can shows from increasing the COF and volume loss as well.

At 25°C (room temperature) and 50°C, ABA2150 shows better performance compared to the other lubricants. The COF of ABA2650 is nearly triple at 50°C. This result might be correlated with the adsorbed layer of those polymers. The higher the number of PEO, which is hydrophilic, seems like to push the adsorbed layer of polymer away from the surface.

4. Conclusions

It has been shown by the experiments that at above cloud point, the ABA2150 perform best either with or without additive. Increasing of PEO side reduce the performance of lubricant due to decreasing of the adsorbed layer of polymer at the surface.

5. References

A Comparative Study on Wear Behaviour of As-Quenched and Over-Aged Bearing Steels

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1. Introduction
Cu bearing steel has been utilised in the automotive industry since it exhibits excellent tensile strength and hardness, due to a precipitation strengthening by Cu particles. Precipitation strengthening can be obtained by air cooling after solution treatment, followed by aging. However, it has also been found that the solute Cu particles can improve the mechanical properties of ferritic steel, since the hardness of ferritic steel has a linear square root relationship with Cu content (26 Hv). Moreover, solid solution of Cu can decrease a ductile-to-brittle transition temperature (DBBT) of ferritic iron. Therefore, the authors have attempted to use Cu bearing steel in tribological applications. In this study, the effect of solute Cu particles and precipitated Cu particles on dry sliding wear behaviour of ferritic iron was investigated. Two materials were studied: as-quenched (solute Cu) and over-aged (precipitated Cu particles) Cu bearing steel.

2. Experimental
The material used in this study was Fe-3mass%Cu alloy. Steel ingot was produced in an induction furnace and was hot rolled to a 20 mm thick plate. Samples were subjected to two different types of heat treatment; (i) solution treatment at 1273K for 1.8ks followed by water quenching, (ii) aging at 873K for 150s to 6 ks. Hardness was measured by a Vickers Hardness tester with an applied load of 29.42N. Dry sliding wear tests were carried out against an EN-31 steel (698 HV) using a pin-on-disc configuration. The tests were performed at 3 different loads: 19.61N, 98.07N, and 196.14N, with a fixed sliding speed of 1ms⁻¹ at room temperature. Volume loss was used to calculate the specific wear rate (K') for the samples. Worn surfaces of each sample were examined using Infinite Focus Microscopy and Scanning Electron Microscopy (SEM).

3. Results & Discussion
In general, the Fe-3mass%Cu alloy underwent both age hardening and over-ageing (softening) at 6ks. The hardness of alloys increased proportionally with aging time, and experienced a decrease as the aging time reached an over-aged condition. It was also found that the hardness of as-quenched samples was almost the same as that of the over-aged samples. A comparison between the current findings and established literature is shown in Fig. 1. It was reported that the hardening mechanism of as-quenched alloys was caused by solid solution hardening of Cu atoms, whereas the hardness of the over-aged Cu bearing steel was due to precipitation hardening of incoherent Cu particles.

4. Conclusion
Specific wear rates decreased with the increasing normal load. Although, the specific wear rates of both samples were almost the same at high loads, the as-quenched sample possessed a higher wear resistance than the over-aged sample. It was caused by the effect of solid solution strengthening of Cu atoms that enhanced the wear properties of the ferritic iron.

5. References
Structural Changes and Tribological Characteristics of FOTS SAMs by UV Irradiation Treatment

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1. Introduction

Self-assembled monolayers (SAMs) have attracted an interest for the applications in MEMS lubrication under external field action, such as electric field, magnetic field and ultraviolet (UV) irradiation. Structural changes to SAMs under UV irradiation have been reported by several researchers. However, tribological characteristics corresponding to structural changes have not yet been studied. In this paper, UV irradiation treatment (IT) to self-assembled monolayers in air was conducted and tribological characteristics were analyzed.

2. Experimental

1H,1H,2H,2H-Perfluorooctyltrichlorosilane (FOTS) SAMs were prepared on a silicon base by self-assembled technique. Structural changes of SAMs after UV IT were analyzed by atomic force microscopy (AFM), X-ray photoelectron spectroscopy (XPS) and FT-IR spectroscopy. Water contact angle (CA) measurements were performed and surface energies of FOTS SAMs were calculated. Tribological characteristics of FOTS SAMs were evaluated using CETR Universal Micro-Tribometer in milli-Newton scale.

3. Results

The results indicate that water CA of FOTS SAMs decrease gradually while FOTS SAMs surfaces change from hydrophobic to hydrophilic. The thickness of SAMs reduces with increasing UV IT time because C-C bond energy in the FOTS molecule chain is the lowest and C-C bond cleavage occurs first. Configuration of FOTS SAMs resembles an island-shaped structure because Si-O bonds rupture last. Surface roughnesses of FOTS SAMs increase with UV IT time. The friction coefficients increase significantly when SAMs change from hydrophobic to hydrophilic. For FOTS SAMs without UV IT, the friction coefficients between SAMs and Si$_3$N$_4$ ball increase with increasing load and sliding velocity. However, for FOTS SAMs with UV IT, the friction coefficients change randomly due to unordered structure.

4. Conclusions

Strongly oxidizing ozone changes the group -CH$_2$ into hydrophilic groups -COOH. In the mean time, ruptures of C-C, C-H and C-F bonds occur, making surface uneven and changing from hydrophobic to hydrophilic. Friction coefficient of FOTS SAMs without UV irradiation increases with increasing load and sliding velocity. After UV irradiation the friction coefficient becomes irregular.

5. References

Evaluation of Long Term Oxidation Stability and Its Effect on the Performance of Few Vegetable Oils

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1. Introduction

For the last three decades there has been an increased interest in various natural lubricants as alternative to mineral oils because of environmental related issues1. But, one of the major concerns with the vegetable oils are their poor oxidative properties, the majority of vegetable oils break down rapidly at temperatures above 150°C. Understanding the long term oxidation of vegetable base oils will be very useful while formulating the biodegradable lubricants, as it can have great influence on the product quality.

2. Experimental

This study focused on the influence of lubricant ageing on the boundary lubrication performance of Castor, Sunflower and Rice bran oils. Accelerated ageing conditions were simulated by storage in a dark oven at 60±1°C and was based AOCS recommended practice Cg 5-97. In the present test, about 600 ml of Castor, Sunflower and Rice bran oils was poured in a 1- litre beaker and kept in an oven and removed after the prescribed ageing time. Samples were kept for 14, 28, 42 days of storage at 60°C. A fresh sample was used as reference. The samples collected at regular interval were evaluated in terms of Percentage of free fatty acid and Peroxide Number. The tribological evaluation of oil samples was carried out using four-ball tester. The tests were carried out immediately after the samples were removed from the oven.

3. Results

The results show that the peroxide content and the amount of free fatty acid increased with ageing as shown in Fig.1.

Fig.1 Variation of free fatty acid content with ageing

Further tribological studies have shown an increase in wear scar diameter with ageing for all the vegetable oils tested. It was caused by the presence of unstable peroxides resulting in forming, during ageing, of various products by reacting with metal surface during the ageing leading to the increased wear as shown in the Fig. 2.

Fig.2 Variation of Wear scar diameter with ageing for different oils.

The presence of moisture in the system resulted in the formation of free fatty acids which in turn reacted during sliding with the metal surface forming metallic soaps. These soaps exhibit low-shear strength resulting in the reduced coefficient of friction with ageing as shown in Fig. 3.

Fig.3 Variation of coefficient of friction with ageing.

4. Conclusions

Oxidation of vegetable oils resulted in increased wear due to the formation of corrosive products, free Peroxides which being unstable forming various products on breakdown resulted in increased wear. The free fatty acids formed during the ageing react with metallic surface forming low shear film resulting in reduced friction with aging.

5. References

Performance of commercial and palm oil based bio-lubricants in turning FCD700 ductile cast iron using carbide tools

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1. Introduction

Coolants and lubricants function differently in machining processes. Coolants control the heat generated in the machining process by quenching with water or oil based coolants, usually containing sulphur, chlorine, or other additives. Lubricants reduce friction in machining, thus minimizing heat build-up [1].

Cost of using coolant is increasing as the number and the extensiveness of environmental protection laws and regulations increase. In this study the performance of both commercial and the proposed palm oil based bio-lubricants was compared. The proposed palm oil based bio-lubricants is easier and cost less to dispose due to its biodegradable property as compared with the commercial mineral based lubricant.

2. Experimental details

The machining experiments were carried out on a Colchester Tornado 600 CNC lathe. The workpiece material used in the test, selected to represent the major group of workpiece materials used in the automotive industry, was FCD 700 cast iron. The performance of both commercial and the proposed palm oil based bio-lubricants was evaluated using the turning parameters shown in Table 1. These parameters were found to give the longest tool life when turning in dry chilled air [2-3].

Table 1 Turning parameters used in the experiment

<table>
<thead>
<tr>
<th>Exp no.</th>
<th>v (m/min)</th>
<th>f (mm/rev)</th>
<th>d.o.c (mm)</th>
<th>Lubricant</th>
</tr>
</thead>
<tbody>
<tr>
<td>D1</td>
<td>120</td>
<td>0.30</td>
<td>0.6</td>
<td>Commercial oil</td>
</tr>
<tr>
<td>P1</td>
<td>120</td>
<td>0.30</td>
<td>0.6</td>
<td>Palm oil</td>
</tr>
<tr>
<td>D2</td>
<td>220</td>
<td>0.20</td>
<td>2.0</td>
<td>Commercial oil</td>
</tr>
<tr>
<td>P2</td>
<td>220</td>
<td>0.20</td>
<td>2.0</td>
<td>Palm oil</td>
</tr>
</tbody>
</table>

3. Results and Discussions

In this study the tool life was limited by the flank wear of (VB) = 0.3 mm. The tool life and surface finish obtained for both commercial and palm oil based lubricants are shown in Table 2. It can be seen that longer tool lives were achieved when turning cast iron using commercial oil lubricant rather than the palm oil bio-lubricant. The tool life was shortened significantly at a high cutting speed of 220 m/min and a high depth of cut of 2 mm. At low turning parameters of 120 m/min, 0.3 mm/rev and 0.6 mm depth of cut, the tool life for the commercial oil lubricant was about 1.7 times longer than that of palm oil bio-lubricant. But as the cutting speed increased to 220 m/min and at 2 mm depth of cut, the tool life for the commercial oil lubricant was up to 3 times longer than that for the palm oil bio-lubricant.

Flank wear land vs. cutting time is shown in Fig. 1.

Table 4 Tool life and surface finish obtained for both commercial and palm oil based lubricants.

<table>
<thead>
<tr>
<th>Exp no.</th>
<th>Tool Life (min)</th>
<th>Final VB (mm)</th>
<th>Roughness (μm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>D1</td>
<td>122.0</td>
<td>0.326</td>
<td>4.010</td>
</tr>
<tr>
<td>P1</td>
<td>71.75</td>
<td>0.30</td>
<td>3.187</td>
</tr>
<tr>
<td>D2</td>
<td>28.33</td>
<td>0.318</td>
<td>2.329</td>
</tr>
<tr>
<td>P2</td>
<td>9.61</td>
<td>0.304</td>
<td>1.287</td>
</tr>
</tbody>
</table>

Fig.1 Flank wear land, VB (mm) vs. cutting time (min)

3. Conclusion

The results show that for both turning operating conditions, the tool life for the commercial lubricant was more than twice longer than that for the palm oil based lubricant.

References
Machined Surface of FCD 700 Ductile Cast Iron When Dry Turning Using Carbide Tool

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1. Introduction

Machinability can be defined as a combination of optimum machining parameters such as low cutting force, high metal removal rate, good surface integrity, accurate and consistent work piece geometry characteristics, low wear rate, and acceptable chip formation [1]. Surface roughness and dimensional accuracy play an important role in the performance of a machined component. In actual machining processes, however, the quality of the workpiece (either roughness or dimension) are greatly influenced by the cutting conditions, tool geometry, tool material, machining process, chip formation, workpiece material, tool wear and vibration during cutting [2].

2. Experimental Work

The machining trials were carried out on a Colchester model Tornado 600 CNC turning machine in dry condition. The FCD700 (JIS) grade ductile cast iron with spherical graphite and ferrite with Brinell hardness and tensile strength in the range of 241 HB and 845 MPa respectively with elongation of 6%. For the purpose of detailed study on the effect of turning parameter, especially the effect of cutting environment (normal air, chilled air, and dry cutting) on the surface roughness of the machined part, ten machining tests were performed for the cutting speed (120-300 m/min), feed rate (0.15-0.4 mm/rev), and depth of cut (0.6-2.0 mm).

The surface roughness was measured using a portable roughness tester Mahr perthometer.

3. Surface Roughness

The cutting speed vs. surface roughness at constant feed rate of 0.15 mm/rev, various depth of cut and cutting environment is shown Fig. 1. High cutting speed is believed to produce better surface finish, especially at cooler temperature of 10 deg C. At much cooler temperature of -2 deg C, even though at lower cutting speed and depth of cut, coarser Ra value of 4.48 micron is produced. Similar result was obtained by Davim et al. [3] and Gusri et al. [4], which shows Ra is sensitive to cutting speed and feed rate. On the other hand, depth of cut has the least effect. Depth of cut produces no noticeable improvement on surface roughness for high cutting speeds, except when operating within the built-up edge range [5]. In addition, a decrease of depth of cut improves surface roughness when operating at a low cutting speed and high feed rate. Ghani et al [6] found that the role of depth of cut is minimum in obtaining good surface finish. Furthermore, low values of surface roughness and cutting force were obtained when the feed rate and depth of cut were kept at low values [7].

4. Conclusions

Fig. 1 Cutting speed vs. surface roughness

High cutting speed is believed to produce better surface finish, especially at cooler temperature of 10 deg C. Ra measured is mainly depend on the feed rate and not on the depth of cut used in the turning operation. Generally, Ra value is increased to double, when doubled the feed rate.

References

Non-Newtonian Piston EHL of Two Different Viscosity Oils in Initial Engine Start-up

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1. Introduction
Critical challenges during low speed engine start-up are the absence of elastohydrodynamic lubrication (EHL) on piston skirts, larger piston-to-bore radial clearance and inappropriate oil viscosity, which fails to assist engine starting and prevent skirts wear. In this study 2-D Maxwell viscoelastic piston skirts EHL, at 600 rpm engine start-up speed, using 0.30187 Pas low viscosity Oil A at 30 µm radial clearance and 0.08571 Pas viscous Oil B at 40 µm clearance, neglecting thermal and surface roughness effects, is modeled.

2. Mathematical Model and Numerical Solution
The model incorporates mathematical relationships of axial and transverse piston motion to calculate eccentric displacements of skirts, oil film thickness and hydrodynamic pressure fields by adopting approach described in\(^1\). To incorporate viscoelastic effects, modified pressure is calculated by adding leading pressure term \(p\) from the solution of Reynolds equation and two perturbation correction terms \(De\) and \(\varepsilon\) from regular perturbation method, as\(^2\):

\[
\frac{dp}{dx} = \frac{\rho(\theta)}{\rho(s)} + \frac{\rho(\theta)}{\rho(s)} + \frac{\rho(s)}{\rho(s)}
\]

Numerical solution shows rigid hydrodynamic skirts lubrication at the respective time steps or crank angles for 720° cycle and includes viscoelastic solution by computing pressure correction terms. For EHL solution, elastic deformations and specified pressure distribution are calculated from geometry yielded by inverse solution of Reynolds equation\(^2\).

3. Results and Discussion
In Figure 1 the dimensionless skirts top and bottom eccentricity curves \(E_t\) and \(E_b\) for oils A and B are plotted between three horizontal lines, with upper line at \(1.0\) is skirts touching line with liner’s non-thrust side, lower line at \(-1.0\) is thrust side’s touching line, (both contacts invite wear) and mid-line at zero shows concentric axial piston motion. \(E_t\), \(E_b\) curve shows generally similar pattern as initial concentric piston motion in induction stroke is followed by transverse eccentricities towards non-thrust side during induction and compression strokes and major shift from non-thrust to thrust side occurs in first half of expansion stroke. Real difference in eccentric displacement is seen when \(E_t\) curve for oil A shows improvement in expansion and exhaust strokes as it drifts away from touching line compared to that of oil B. Corresponding hydrodynamic pressure fields, affected by piston eccentricities show that low pressures generated over 75° wide skirts surface with gentle slopes during intake stroke, gradually build up during the compression stroke, to rise very high during combustion with steep slopes and finally settle down to low values during the exhaust stroke. Film thickness reduction from 1.25 µm for oil A and 1.7 µm for oil B in hydrodynamic lubrication regime to a fraction of µ in EHL regime occurs. In Figure 2, \(h_{\text{bar}}\) is the film thickness at maximum pressure before deformation and EHL film, after deformation. EHL film profile shows less than 0.2 µm thickness for oil A and less than 0.4 µm thickness for oil B during most of the 720° crank rotation cycle. In Figure 3, dimensionless EHL pressure field over film thickness show different rising pressure intensities. For oil A, after gradual build up high intense pressure approaches maximum value of around 25% compared to relatively low intensity 40% pressure rise without its significant gradual buildup for oil B.

4. Conclusions
Low viscosity non-Newtonian oil A and viscous oil B show less chances of skirts physical contact with the liner but with oil A, piston concentric displacement visibly improves. Despite sharp reduction of the film thickness, it remains within the EHL regime for both oils A and B due to viscoelasticity effects. For oil B, EHL film thickness is of relatively thick due to which low EHL pressures are generated over elastically deformed surfaces, reducing the chances of their conversion to Hertzian pressure and subsequent damage to interacting surfaces. In conclusion more in depth would need to be conducted.

5. References
Shear Heating in Piston Skirts EHL Modeling in High Speed Initial Engine Start-up

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1. Introduction

A few initial engine start-up cycles cause piston and liner wear in the absence of elastohydrodynamic lubrication (EHL) film and normal engine operating conditions. High engine start-up speed is necessary for early engine warm-up but viscous shear heating dominates and temperature rise decreases viscosity to reduce load carrying capacity of originally high viscosity oil and increases engine start up wear. In this study Newtonian 2-D piston skirts hydrodynamic and EHL using the 2-D heat equation to incorporate adiabatic conduction and convective heat transfer for 0.1891 Pas viscosity oil at 313 K ambient temperature, 1600 rpm engine speed and by neglecting combustion heat and surface roughness effects is modeled.

2. Mathematical Model and Numerical Solution

The axial and transverse piston motion is modeled to calculate piston skirts eccentricities, oil film thickness and pressure fields in similar way to that defined by Zhu et al. 1. To incorporate shear heating effects the following form of heat equation is solved as given by 2:

\[ \rho_l \frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} = \left( \frac{\partial}{\partial x} \left( \alpha \frac{\partial T}{\partial x} \right) \right) + \varnothing \]

where, \( \alpha \) = Specific heat of oil; \( T \) = oil temperature; \( u \) = piston velocity; \( \rho_l \) = oil density; \( \varnothing \) = viscos term.

The numerical solution incorporates 2-D Reynolds, heat, Vogel, Roeland and film thickness equations to simulate the second-order non-linear differential equations to show the skirts lubrication at respective time steps or crank angles using finite difference method. The EHL solution uses the inverse solution technique 3 by integrating Reynolds equation for known skirts geometry and calculating the elastic deformations.

3. Results and Discussion

The results obtained show that in hydrodynamic lubrication, transverse displacements of piston skirts at high initial engine start-up speed and viscosity drop due to shear heating establishes physical contact between skirts and liner in expansion stroke resulting in engine start-up wear. In EHL regime, physical contact is barely avoided when high pressure increase elastically deforms piston and liner surfaces and despite temperature rise oil viscosity increases many orders of magnitude from that in hydrodynamic regime. The maximum and minimum film thickness profiles for 720°, 4-stroke crank rotation cycle in hydrodynamic lubrication regime are shown in Fig. 1(a). After reaching apex, film thickness becomes very low in power stroke due to combustion gas force effect. In EHL regime, film thickness reduces from a few \( \mu \text{m} \) to a fraction of a \( \mu \text{m} \) (Fig. 1b). Shear heating significantly affects oil viscosity at high start-up speed. In hydrodynamic regime, it drops to low values during compression and power strokes (Fig. 2a). In EHL regime, pressure dependent viscosity increase is four times the viscosity at ambient conditions to compensate for earlier viscosity drop (Fig. 2b). At high start-up speed, high temperature due to excessive shearing is anticipated. In Fig. 3(a), temperature field over surface of skirts in hydrodynamic regime shows maximum temperature increase up to 319 K, biased to the middle and right side. In EHL regime, temperature field shifts towards skirts top surface and maximum increase is up to 355 K and bias shifts to the right and left sides. An increase to 355 K is due to excessive shear heating because of significant viscosity increase in EHL regime.

4. Conclusions

Shear heating critically affects the efforts to minimize initial engine start up wear. High start up speed is detrimental to the efforts for wear prevention as high viscosity oil and a few micrometers thick film cannot compensate for large viscosity reduction in hydrodynamic regime. High temperature due to shearing in EHL regime partially restricts maximum viscosity increase to a little more than four times the original value, generating a fraction of a micron thick film, barely sufficient to prevent possible skirts and liner contact during the expansion stroke of piston.

5. References


Raman Spectroscopy Study of Impacted DLC Coatings

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1. Introduction

Raman scattering is an excellent tool to characterize the structure of carbon atoms in diamond-like carbon (DLC). The study of DLC coatings in the sliding conditions has been conducted for a decade using Raman spectroscopy analysis. However, there is still insufficient information about how the structure of DLC coatings changes during repetitive impact. In this paper changes in the structure under 90° repetitive impact at a large number of cycles are presented.

2. Experimental

DLC films were deposited by physical vapor deposition (PVD) method on tungsten high speed steel (SKH2) disc. The impact test was performed using a custom made impact tester, where a DLC coated disc was impacted by a chromium molybdenum steel (SCM420) pin at several impact loads under repetitive impact up to 100,000 cycles. This 90° inclination of impact was run under lubricated conditions. The diameter of disc and pin are 10mm and 2mm respectively. In the present work, the frequency of the impacts was selected at 10Hz. The bonding structures of wear debris as well as transfer layer on the pin were studied using Raman spectroscopy. In order to analyze the bonding structure of impacted DLC disc taken from the surface of impact crater, the region from 800-2000 cm⁻¹ in Raman spectrum was fitted with two Gaussian curves.

3. Results and Discussion

Series of Raman spectra of transferred DLC on pin under different impact cycles, with two predominant peaks at approximately 1317 cm⁻¹ and 680 cm⁻¹ are shown in Fig. 1. In the previous paper, these two peaks originate from a hematite (α-Fe₂O₃) and magnetite (Fe₃O₄) phases respectively. It is clearly revealed that the tribochemical reaction of DLC coatings occurred after several impact cycles due to the oxidation of ferrum (Fe) in the transfer layer. Additionally, the G peak appears to be significantly shifted towards higher frequencies, when compared to the as-received after 100,000 impact cycles. According to other researchers, this implies that DLC coatings are gradually transformed into graphite-like carbon associated with increase in impact cycles. This phenomenon also occurred in wear debris, where the Raman spectra were obtained from the debris on the edge of impact crater.

From the results of Gaussian fit analysis, decreasing ID/IG ratio with full-width at half maximum (FWHM) of G-line and impact cycles evidently correlated with higher sp³ fractions of impacted DLC coatings. In addition, its hardness also increased as reported by other researchers. From the observation of surface roughness using atomic force microscopy (AFM), it is believed that increasing sp³ fractions and the hardness of the impacted DLC coatings is associated with the reduction of surface roughness during impacting.

4. Conclusions

From the Raman spectroscopic analysis, the results show that the tribochemical reaction occurred at the transfer layer and wear debris after several impact cycles under 90° impact angle. This is due to the oxidation of Fe to α-Fe₂O₃ and Fe₃O₄ phases. In addition, the shift of G peak towards higher frequencies indicates that the DLC coatings gradually transformed into graphitic phases during impacting. As a result of decreased ID/IG ratio with FWHM of G line and impact cycles, the impacted DLC coatings tend to have higher sp³ fractions and are harder after numerous impacts. This can be correlated to the reduction of surface roughness of the surface layer of impact crater after the impact.

5. References

Tribological Study of the Interaction Between Surfaces of Polymer/Metal Friction Pairs

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1. Introduction

The authors present relationship between tribological properties of metal-polymer friction pairs and physical interaction of metal and polymer surfaces.

The authors present a new approach to optimization of the metal-polymer friction pair technology based on correlation between the total sum of polar component of the values of surface free energy and tribological characteristics (wear and friction) of rubbing materials.

2. Results and discussion

The current efforts put in the improvement of the mechanical properties of polymers do not take into consideration the energy state of the surface of polymer composites. Furthermore, the aspect of presence of oxides on metal (usually steel) surfaces, which create bonds with hydrogen, also is not usually taken into consideration. It is well known, that the process of oxidation promotes the tendency to creation of adhesive bonds between metal and polymer. The authors proved that the PVD coatings covering steel may reduce the tendency to such bonds creation and transfer polymer to the metal surface (Fig 1).

Fig. 1. The schemes of the hydrogen bonds in cases of polymer-steel and polymer - Ti(CN) interactions

3. Experiment

The increase of the durability of polymer/metal friction pairs can be obtained by the protection of the steel surface using engineering coatings (especially PVD/CVD) – Fig. 2. These coatings, properly selected, limit the adhesive interactions with the surface of polymer composite and create the barrier for destructive action of hydrogen. The additional benefit is the limiting of area of the contacts between the polymer and iron oxides located on the surface of the steel part.

Fig. 2. Wear and friction force for steel and polymer composite friction pair

These oxides are the nuclei for creation of hydrogen bonds - main mechanism responsible for the polymer transfer and destruction of the polymer material.

4. Conclusion

The investigation of the processes occurring in polymer/steel friction pairs proves, that the interaction of polymer composite (or thin polymer coating on steel) with steel surface covered by PVD coating enables to eliminate the numerous, mentioned above, drawbacks and by this significantly reduces the negative consequences of the friction process. The aspects of presence of the oxides on steel surfaces, which create bonds with hydrogen, also must be taken into consideration because this process promotes the tendency to creation of adhesive bonds between metal and polymer.

5. References

Molecular Gas-film Lubrication Analyses of a Slider over a BPM Disk
(Static and Dynamic Flying Characteristics of a 3-DOF Slider)

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1. Introduction

Recording media with grooves, such as discrete track media (DTM) and bit-patterned media (BPM), are considered to be some of the most promising media for achieving ultrahigh recording densities [1]. Thus, it is becoming increasingly important to analyze the static and dynamic characteristics of flying head sliders over DTM and BPM media using the molecular gas-film lubrication (MGL) equation [2-4]. In this study, a numerical scheme for analyzing the dynamic responses of a slider over BPM media is clarified.

2. Analysis

We considered the BPM disk configuration as the sum of sinusoidal disk waviness (Fourier series) and obtained the static slider attitude for the time-independent film thickness. For dynamic responses, we utilized a perturbation method in the frequency domain and obtained the stiffness and damping produced by the air film and also the negative stiffness caused by the attractive van der Waals (vdW) force.

2.1 Basic equations

When analyzing the surface pressures acting on a slider with a nanometer-order spacing, the pressures produced by both the air bearing (P_{MGL}) and the attractive vdW force (P_{vdW}) should be considered.

\[ P_{\text{total}} = P_{\text{MGL}} + P_{\text{vdW}} \]  

The molecular gas pressure \( P_{\text{MGL}} \) is calculated by the MGL equation [2, 3] and the vdW pressure \( P_{\text{vdW}} \) between the slider and the disk is

\[ P_{\text{vdW}} = -\bar{A}_{132}/60\pi H^3, \]  

where \( \bar{A}_{132} \) is the non-dimensional Hamaker constant, which is determined by the refractive indices of the solids. When the disk and the slider have diamond-like carbon (DLC) coatings, \( A_{132} \) will be 1.8×10^{-19} J [5].

2.2 Perturbation method

We used the perturbation method and obtained an equation for the static pressure \( P_0 \) and an equation for infinitesimal dynamic pressure \( \psi \).

\[ P = P_0 + \psi, \quad H = H_0 + \eta \quad (\psi < P_0, \eta < H_0) \]  

3. Numerical results

We calculated the dynamics of a plane-inclined slider with three degrees of freedom (3-DOF) over an asymmetric groove–slider configuration and numerically determined the dependence of the dynamic characteristics (such as spacing fluctuations) on the groove depth and the BPM wavelength (or frequency).

Figure 2 shows the relationship between static slider attitudes and the groove depth \( h_{\text{groove}} \). The decrease of the flying height due to \( h_{\text{groove}} \) is significant. Figure 3 shows the relationship between dynamic spacing fluctuations at the trailing edge (X=1) and \( h_{\text{groove}} \). The influence of the dynamic behavior on \( h_{\text{groove}} \) is insignificant. This is because the slider dynamics depend on the static spring load which is constant not only in the calculation but in the realistic HDI system.

4. References


Fig. 2 Static slider attitude vs. groove depth
Fig. 3 Spacing fluctuation vs. groove depth
The Importance of Tool Surface Roughness on the Galling Tendencies in Cold Forming of Aluminium

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1. Introduction

Aluminium alloys are shaped in cold condition in a large number of industrial applications and simple geometries can be produced to net shape in a single operation. The tool life, number of processing steps and complexity of the shapes are limited due to high stresses occurring when forming complex shapes and galling, i.e. transfer of aluminium work material to the tool surface, followed by hardening of the aluminium, that leads to problems in the next forming operation. Both stresses and galling are closely related to friction and adhesion at the tool to workpiece interface.

Earlier investigations proved that a selection of PVD coatings of DLC type have positive influence on galling prevention, even without lubricant, as long as the coating surface is very smooth\(^1\). With rough coating surfaces the DLC coatings showed no significant improvement compared to uncoated tool steel. The present investigation concentrates on the characteristic of the surface roughness and its influence on galling.

2. Experimental

The laboratory testing is performed in a load-scanner test set-up\(^2,3\). The test comprises two crossed cylinders in sliding contact, one made of coated tool material and one of aluminium work material, imitating the galling and friction situation in the forming process. The test scans over a wide load range, 20-1000 N, while monitoring the coefficient of friction. The test is focused on the friction level and the amount of adhered aluminium on the tool.

The tool is made of AISI H13 with a two layer coating; hydrated DLC on top of a Me-doped DLC (1.0 \(\mu\)m and 0.8 \(\mu\)m respectively, hardness 1500 Hv). The coating has proven to give very high resistance to galling in previous testing\(^1\). The deposited coatings are treated to show different degrees of roughness and surface irregularities. Starting with a polished surface, Ra 0.03 \(\mu\)m, the equally well polished tool surfaces are given controlled surface defects ranging from a few nm to several \(\mu\)m. The defects can be made in a controlled manner using a nanoindenter with a Berkovich diamond tip and a micro indenter with a Vickers diamond tip. The shape of the surface defects introduced is characterised using optical microscopy and SEM. The work material is an industrially important AlMgSi alloy.

3. Results

Indents were made along the coated tool rods with a separation of 8 mm and to six different depths; 8 nm, 65 nm, 160 nm, 650 nm, 1.5 \(\mu\)m and 5.6 \(\mu\)m, one depth on each rod. The first three indents were made with the use of a nanoindenter and the last three with the use of a microindenter. A reference rod without indents was kept finely polished. The geometry and size of the indents are visualized using optical microscopy and SEM, see Fig. 1 with examples.

![Fig.1 Image of indentation with depth (a) 160 nm (optical microscopy) and (b) 5.6 \(\mu\)m (SEM).](image1)

After testing in the load scanning equipment aluminium has adhered to the coated surfaces. The largest indents have been studied in the SEM and it is clearly visible that aluminium has filled the indents (Fig. 2). The surface roughness then causes galling in the next forming step.

![Fig.2 SEM images of a 1.5 \(\mu\)m indent at the position with load 100 N (a) before and (b) after testing.](image2)

The friction coefficient does not differ significantly with the indent depth during one stroke, but since aluminium adheres due to the indents, the friction coefficient increases faster with higher roughness in continuous testing.

4. Conclusions

The size and depth of the surface roughness are important for galling resistance. Indents, simulating occasional dents in the coating, pick up more aluminium than the surrounding coating. This implies that the forming tools need fine polishing and gentle handling to avoid dents.

5. References

A Comparison Between the Performance of Ferro and Magnetorheological Fluids in a Hydrodynamic Bearing

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1. Introduction

This paper focuses on the possibility of applying ferro- (FR) and magnetorheological (MR) fluids as lubricants for an actively controlled hydrodynamic bearing (ACHB). Both FR and MR contain magnetic particles suspended in the carrier fluid, but differ through their particle size as well as how they respond when subjected to a magnetic field. The FR fluids contain nano-scale particles whose Brownian motion keeps them from settling or clumping together1. MR fluids contain micro-size particles which under the influence of a magnetic field clump together causing the fluid to become extremely viscous. This magnetically induced additional viscous resistance induces a high yield stress which must be overcome in order to resume fluid motion2. The key elements to be studied are: the fluid flow patterns, pressures, load carrying capacity, stability, stiffness, and damping.

2. Numerical Analysis

The journal bearing used in this study has a radius of 12.7mm, an axial length of 12.7mm, a concentric clearance of .0254mm, an eccentricity range of \(\varepsilon=0.5-0.9\) and an angular velocity range of 2000-4000 rpm. A pressure outlet was used for one opening while a symmetry boundary condition was used for the other. This allowed using only one-half of the bearing effective length in the numerical implementation. The pressure outlet value was set to zero gauge. The magnetic field was generated using a current carrying wire.

To introduce the effects of the magnetic field on the fluids, a user defined function (UDF) was generated for each fluid and coupled with the three-dimensional ANSYS-FLUENT algorithm. According to Rosensweig1, the FR fluids experience a Kelvin body force in the presence of a magnetic field, which is then introduced as a source term in the Navier-Stokes equations. For the MR fluid the rigidity occurring in it, can be simulated by calculating the local viscosity using an initial viscosity that subsequently changes due to the effects of the magnetically induced local yield stress2.

The cavitation model accounting for the film rupture in the diverging region of the bearing has been proposed by Singhal et al.3. It has been inbuilt into the ANSYS FLUENT commercial algorithm which solves the three dimensional Navier-Stokes equations with a variable density and viscosity.

3. Numerical Results

ANSYS FLUENT4 was used in tandem with these authors’ UDFs in order to solve and evaluate the proposed ACHB concept. Figure 1 presents the pressure curves along the circumference at the axial line of symmetry of the bearing, for FR and MR cases when different currents are applied in order to induce the needed magnetic field.

![Figure 1. Pressure curve for a hydrodynamic journal bearing: (a) without magnetic field (dashed); (b) FR with 500 amp wire (solid); (c) MR fluid with 10 amp wire (dotted)](image)

While both fluids show a change in the pressure curve, the MR fluid shows a higher magnitude for a comparatively smaller magnetic field when compared to a FR. The FR (curve (b)) shows a spike in pressure in the proximity of the wire, with little change elsewhere. Similarly, the MR fluid shows a change in pressures also only in the proximity of the wire, yet the change affects a much larger area with a consequent higher load carrying capacity. The yield stress generated by the magnetic field significantly affects the apparent viscosity in that area.

4. Conclusions

FR and MR fluids differ in particle size and react quite differently in the presence of a magnetic field. This paper discusses the possible application of these fluids for lubrication and control of an ACHB.

5. References

4. ANSYS FLUENT Theory Guide, ANSYS 12.1, Canonsburg, PA, 2009
Microstructure and Porosity Effects on the Load Carrying Capacity of a Variable Geometry Thrust Bearing (Hybrid Rayleigh Step and Sinusoidal Geometries)

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1. Introduction

This paper studies the effect of microstructure profiles and porosity effects on the development of flow patterns, pressure profiles, load carrying capacity and dynamics characteristics (stiffness and damping) of a hybrid hydrostatic thrust bearing subject to: (i) Variable depth Rayleigh (VDRS) step, Figure 1a and (ii) Sinusoidal wave (SW) type geometries, Figure 1b. The hydrostatic effect is reached either through (a) a single central fluid injection feed or (b) a distributed feed through the porous stationary plate. This parametric study focuses also on the influence of the etched microstructure on the bearings’ overall performance.

2. Numerical Analysis

Additional geometric parameters characterizing these profiles include: (a) the total number of circumferential periodic microstructures (b) the aspect ratio \(AR\) \([AR=\frac{D}{L}\) (\(D\)=pocket depth and \(L\)=wavelength for SW (Rayleigh step circumferential chord for VDRS)], and (c) the pocket bottom slope angle \(\theta\), (d) porosity(\(\phi\))/permeability(\(\kappa\)) as characterized by Darcy law, (e) the upper plate angular velocity \(U\) and (f) the inlet restrictor supply feed line pressure \(P_{axial}\).

The nature of the microstructure imposed the usage of a very fine grid which in turn forced the establishment of periodic characteristic repetitive regions, Fig.2, of a more limited grid size. The region shown in Fig.2 is isolated from the rest of the bearing by means of periodic boundary conditions. The numerical results are obtained using the commercially available software package ANSYS CFX1, which employs an element-based finite volume method where mass, momentum and energy are conserved at every grid point.

3. Results

Present space limitations impose showing only some partial results. Figure 2 presents numerical results for the radial and circumferential distribution of pressure in a SW, characterized by an angular velocity of 3000rpm, a working fluid density of 888 kg/m\(^3\) and dynamic viscosity of 20cP (kg/m-s). One can see the distribution of pressure throughout the maximum and minimum clearance of each cyclic period of the SW profile. In the divergent portion of the pocket, pressure decreases due to the depth increase in the cross sectional area; this in turn causes the inertial effects in the flow that cause a pressure drop. Mass conservation combined with added resistance due the local wave converging geometry cause now a sharp pressure spike similar to the one seen in a hybrid Rayleigh step geometry, Braun and Dzodzi.2

4. Conclusion

The paper presents a numerical study of flow and pressure patterns in two types of hybrid hydrostatic thrust bearings (i) VDRS and (ii) SW. The analysis uses periodic boundary conditions and uniformly imposed inlet pressure condition for the fluid feed, whether the feed is central or through the porous bottom. \(Re_{calt}\), \(P_{axial}\), \(AR\), inclination angle \(\theta\), porosity \(\phi\) and permeability \(\kappa\) greatly influence the size and location of the vortical cells as well as the pressure profiles and dynamic characteristics of these microstructured type thrust bearing geometries.

5. References

Friction Behaviour of MoDTC on Various Surfaces

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1. Introduction

Molybdenum dialkyldithiocarbamate (MoDTC) is widely used as a friction modifier in engine and other lubricants. Its friction-reducing properties have been widely explored and the additive is now known to form nanocrystals of molybdenum disulphide on the load-bearing asperities within a rubbing contact\textsuperscript{1,2}. Most research work published to date has examined the behavior of MoDTC friction modifier on ferrous surfaces such as steel. Relatively little is known on how MoDTC behaves on surfaces other than ferrous. In an engine, it is most likely that friction modifiers would have to interact also with surfaces, such as antiwear additive deposited films. It is well known that ZDDP significantly alters rubbed steel surfaces to form a zinc phosphate glass-like film on the wear track. As well, overbased calcium detergents are also found to deposit thick, rough, CaCO\textsubscript{3} antiwear films\textsuperscript{3}. Furthermore, DLC coatings, now widely used in the industry, impose new challenges and interactions to friction modifying additives. Thus, MoDTC has to interact with surfaces that are very different from unrubbed ferrous surfaces.

In this paper, the film forming and friction-reducing properties of MoDTC on a range of different types of surfaces are measured. The effect of surface roughness, nature of the surface and rubbing conditions on effectiveness and durability of the MoDTC are explored.

2. Experimental

A rolling-sliding Mini Traction Machine (MTM), (PCS Instruments, London), was used to form additive reaction films and monitor their friction properties. Prolonged rolling-sliding was carried at a low entrainment speed of 0.1 m/s, a slide-roll ratio of 50 \%, and a load of 30 N. Periodically, motion was halted and Stribeck friction curves obtained in which friction was measured over a range of entrainment speed. Herewith, both, the initial effectiveness and friction performance of the MoDTC, as well as its durability and friction development over prolonged rubbing time could be obtained. Before resuming low speed rubbing, \textit{in situ} optical interference technique was used to monitor the tribo-film thickness formed on steel surface of the ball. Once an anti-wear reaction film was formed, the anti-wear additive solution was replaced by one consisting of just MoDTC in base oil and rubbing was continued while monitoring friction and film formation.

3. Results

Figure 1 shows a series of Stribeck curves measured during prolonged rubbing of MoDTC-oil solution on OBCaSu-derived film. For the MoDTC oil solution a remarkable reduction of coefficient of friction is observed in the boundary lubrication regime. Interestingly, from the corresponding interference images (Fig. 2), despite the dramatic effect in friction, little alteration of the previously-formed OBCaSu film can be seen.

4. Conclusions

It was found that MoDTC is effective in producing low friction films on the thick deposited CaCO\textsubscript{3} films formed by overbased detergents, on the reaction films formed by the antiwear additive ZDDP and also on DLC coatings. In addition, Atomic Force Microscopy (AFM) could further reveal similarities and differences of the MoDTC action and add to the understanding of its friction performances on different surfaces.

5. References

Heat Dissipation in MD Simulation of Nano-Scale Lubrication

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1. Introduction

When a lubricant is confined to less than ten molecular diameters, its static and dynamic properties start changing, both quantitatively and qualitatively, from the bulk ones [1]. For moderate contact sliding speeds, the encountered shear rates can be very elevated, i.e. $10^8 - 10^{10}$s$^{-1}$, causing significant viscous heating and largely influencing the lubricant dynamic behavior.

Numerical simulations by Molecular Dynamics (MD) have been extensively used for investigating nano-scale lubrication. Nevertheless, some artifacts related to the energy dissipation mechanisms are still employed.

Energy is dissipated in MD by using a thermostat. The choice of a thermostat in a friction simulation should be very careful especially if the encountered shear rates are high.

2. Heating onset due to shear

MD simulations were conducted with a confined system of 2.5nm thick hexadecane film between two gold surfaces. The sliding velocities were varied between 1 and 200m.s$^{-1}$, covering a wide range of shear rates. A Sliding Boundary Thermostat (SBT) [2] was implemented with a Langevin algorithm to control the surface temperature at 300K. Finally, a normal load of 500MPa was applied. A snapshot of the system with the boundary conditions is given in Fig.1.

The temperature rise inside the film as a function of the apparent shear rate is shown in Fig.2. For shear rates lower than $\dot{\gamma}_c = 6 \times 10^6$s$^{-1}$, the maximum dissipation rate of the thermostat is larger than that of its generation inside the film. As a result no lubricant heating is observed. On the other hand, for shear rates higher than this critical value, the lubricant heats up. In this regime, the dissipation methods are expected to significantly influence the measured friction in the simulations [2].

Table 1 Temperature rise and the total shear stress for three different dissipation methods.

<table>
<thead>
<tr>
<th>Dissipation Method</th>
<th>Temperature rise (K)</th>
<th>Shear Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Surface</td>
<td>Lubricant</td>
</tr>
<tr>
<td>SBT</td>
<td>57.8</td>
<td>55.4</td>
</tr>
<tr>
<td>PM</td>
<td>84.7</td>
<td>60.6</td>
</tr>
<tr>
<td>VBT</td>
<td>82.3</td>
<td>60.8</td>
</tr>
</tbody>
</table>

The results show that the choice of the dissipation method becomes crucial at elevated shear rates. Classical SBT method with constant surface temperature over-dissipated energy and eventually over-estimated the friction in the contact. The results using advanced dissipation methods (i.e. PM and VBT) show that the physical temperature rise at the surfaces level acts in reducing energy dissipation and results in a lower friction.

3. Comparison between different thermostating methods

MD simulations were run on the molecular system of Fig.1 with a sliding velocity of $v = \pm 100$m.s$^{-1}$. The apparent shear rate is $8 \times 10^{10}$s$^{-1}$. Three boundary dissipation methods were tested. The first method is the SBT. The second is the Phantom Molecules (PM) method. The third is the Variable Boundary Temperature (VBT), a novel dissipation method in which the boundary temperature evolves during the simulation according to the heat flux generated in the contact [3].

4. References

A Universal Model for Calculating the Lubricant Churning Loss of Gears

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1. Introduction

Nowadays higher operating powers and speeds are used in the gearboxes and greater power losses are generated by lubricant churning, tooth and bearing friction losses.

The higher power losses generate higher tooth surface temperatures decreasing the viscosity of the lubricant between the mating teeth. This causes higher frictional losses and lower load carrying capacity of the gears, limited by scuffing and scoring on the teeth surfaces.

There are methods for calculating the tooth and bearing friction losses but calculating the lubricant churning losses is more difficult. In this paper the lubricant churning loss of planetary gears and a new universal model for calculating the churning loss is discussed and the calculated values are compared with measured experimental results.

This new algorithm for calculating the oil churning loss takes into consideration the loss of oil expelling, the power loss of splashing and the disc churning loss.

2. Oil churning losses of gears

Lubricants are used mainly for decreasing the frictional losses and for cooling by dissipating the heat. However using too much oil in the contact zone of the gears causes higher power losses. In order to use the best lubrication for a given application it is important to understand, to determine and to minimize the lubricant churning losses in gear drives.

A new model was developed by Attila Csoóbán [1] for calculation of the oil churning loss containing the loss of oil expelling (Fig. 1.), the power loss of splashing and the disc churning loss of gears. This model is able to take into consideration all the special features of the oil churning losses of planetary gears.

3. Power loss of oil expelling

As the teeth of the gear turn into the tooth valleys of the gear in to a meshing cycle, the injected redundant oil volume is expelled by the teeth from the tooth valleys. The average power losses generated by the oil expelling can be calculated (Fig. 2.) with the following equations:

If \( \xi_i \geq \Omega \):

\[
\bar{P}_{\text{ex}} = \frac{P}{\xi_i} \int_{\phi_1}^{\phi_2} V_2^3(\phi_1) d\phi_1
\]

If \( \xi_i < \Omega \):

\[
\bar{P}_{\text{ex}} = \frac{P}{\Omega} \left( \int_{\phi_1}^{\phi_2} V_2^3(\phi_1) d\phi_1 + \int_{\phi_3}^{\phi_4} V_2^3(\phi_1) d\phi_1 \right)
\]

In the equations the following parameters were used: \( A_{\text{ex}_i} \) – cross sections of the oil flux \([m^2]\), \( \phi_2 \) - pinion or gear turn angle, parameter for oil expelling calculations \([\text{rad}]\), \( \xi \) – tooth distribution angle \([\text{rad}]\), \( V_{\text{ex}_i} \) – expelled oil flux \([m^3/s]\), \( \Omega \) – tooth turn angle \([\text{rad}]\), \( \bar{P}_{\text{ex}} \) – average power loss of oil expelling \([\text{W}]\), \( \rho \) – oil density \([\text{kg/m}^3]\), \( n_i \) – gear or pinion feet radius \([\text{m}]\), \( h_w \) – tooth whole depth \([\text{m}]\), \( \xi_{\text{ok}} \) – oil expelling factors \([-]\), \( \alpha_0 \) – angle velocity of gear or pinion \([\text{rad/s}]\).

4. References

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Temperature influence on dry clutch mechanical, thermal and tribological properties


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1. Introduction:

During dry clutch engagement, a transient sliding contact situation occurs between the pair of clutch facings mounted on the friction disk and the counter faces belonging to the flywheel and the pressure plate. Throughout this engagement process, heat is generated at the contact surface yielding temperature to rise [1].

Temperature rise may be very important in case of repetitive engagements until involving thermal fatigue on cast iron disk.

The dry clutch's tribological behaviour is closely linked to the characteristics of the friction material, such as stiffness, chemical composition and structure.

Good performances of the clutch facings material such as a stable and sufficiently high coefficient of friction are required in order to operate engagements in an efficient and regular way [2].

Fading phenomenon, which is due to a sudden decrease of the friction coefficient with temperature, must be avoided. Evolution of the friction coefficient under working conditions or due to material wear must also remain limited [2].

This paper presents the temperature effects on the behaviour of the clutch's friction material. The thermal degradation of the material is first studied using the Differential Scanning Calorimetry technique (DSC). Then, friction tests and Dynamical Mechanical Analysis (DMA) allow to characterize the mechanical and tribological behaviour as a function of temperature.

2. Materials and methods:

Most of the clutch facing’s discs, available on the market on personal cars, are based on special woven wire’s use (copper wires, glass fiber...). Thus, this kind of friction material is particularly composite and complex.

These wires are soaked in an organic binder and shaped into a mould, with a precise geometry. Then, this assembly is annealed in presses and size-adjusted.

In this study, three complementary experimental techniques are used:

- Pin on disk tests have been realized on a high velocity tribometer and are considered as screening tests. The type of contact is plane on plane. The pin is a friction material and the disk is cast iron. Three levels of normal load are used at different speeds and temperatures. - DSC allows to measure the heat required to increase the temperature of a sample/reference according to a defined thermal profile.

- DMA is a technique used to characterize the mechanical properties of viscoelastic materials having elastic modulus and phase angle greatly varying with temperature and frequency.

The dynamic mechanical test consists of measuring the stress resulting in the sample on which a sinusoidal stress is applied with regard to a fixed temperature and pressure. This test allows to determine the intrinsic characteristic of the tested material.

3. Results:

DSC and DMA (Fig.1 and Fig.3) tests show a correlation between thermal and mechanical degradation of the friction material. Both techniques show significant energy dissipation above 250°C.

![Fig.1 Heat capacity versus temperature](image1)

![Fig.2 Wear volume versus temperature](image2)

![Fig.3 Young modulus and phase angle versus temperature](image3)

This high energy dissipation is responsible for the deterioration of the mechanical properties (as illustrated by the young’s modulus quick decrease above 250°C), an increase of the volumetric wear (Fig.2) and an important mass loss.

4. Conclusion:

Those mechanisms will be explained by observing and analysing the worn surfaces and the third-body of the friction material. As a result, the thermal degradation of the material could be connected to the deterioration of its mechanical properties; both effects directly influence the material’s wear and friction behaviour.

5. References

Development of Titanium Nitride – Molybdenum Disulfide Composite Thin Films with Low Friction Property and High Hardness

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1. Introduction

Titanium nitride (TiN) films show high hardness, high adhesive strength and good thermal stability. However, the friction coefficients of TiN films are higher than those of diamond-like carbon (DLC) films.

In order to reduce the friction coefficient of TiN films, the authors tried to deposit TiN-MoS 2 composite films by addition of molybdenum disulfide (MoS 2), which showed low frictional characteristics. In addition, the authors investigated the influence of Ti interlayer on the mechanical properties of the TiN-MoS 2 composite films.

2. Experimental

TiN-MoS 2 composite films were deposited on the silicon substrate by using the DC magnetron sputtering method. TiN-MoS 2 composite targets, in which MoS 2 content was changed from 10 to 30 wt.%, were used to obtain the composite films with varying MoS 2 concentration. In addition, Ti interlayer (0.02–0.08µm thick) was formed between the silicon substrate and TiN-MoS 2 composite films.

The chemical bonding state and MoS 2 concentration of TiN-MoS 2 composite films were characterized by X-ray photoelectron spectroscopy (XPS). Nano-indentation and microscratch tests were performed to evaluate the microhardness and the adhesive strength of composite films. The friction coefficients were measured using a ball-on-disk tribometer in dry conditions at room temperature.

3. Results (varying MoS 2 content)

MoS 2 content of the TiN-MoS 2 composite films was varied from 6 to 22% by increasing MoS 2 content of targets from 10 to 30 wt.%. In addition, XPS Mo3d spectra of the composite films correspond to the standard spectral line of MoS 2 (229.0eV) 1).

The maximum hardness of TiN-MoS 2 composite films was about 22GPa, which was almost equal to that of the TiN film. However, the hardness of composite films decreased with the increase of MoS 2 content and was about 15GPa.

On the other hand, the adhesive strength of composite films did not change with increase of MoS 2 content.

Figure 1 shows the friction coefficients of TiN-MoS 2 composite films, TiN film and DLC film. The friction coefficient of the composite film was lower than that of the TiN film. Especially, the composite film with high MoS 2 content (22%) showed low friction coefficient of around 0.1, which was almost equal to that of the DLC film.

4. Results (forming Ti interlayer)

The hardness of TiN-MoS 2 composite films (MoS 2 content : 22%) did not change by forming Ti interlayer (0.02–0.08µm thick).

On the other hand, the adhesive strength of composite films increased with the increase of thickness of Ti interlayer.

The friction coefficients of TiN-MoS 2 composite films with Ti interlayer were around 0.1 in dry condition at room temperature. In addition, the wear depth of composite films was decreased by forming Ti interlayer, and the specific wear rate of TiN-MoS 2 composite film with Ti interlayer (0.08µm thick) was around 3.0×10⁻¹⁰ mm²/N, which was lower than those of composite films without Ti interlayer.

5. Conclusions

1. With the increase of MoS 2 content in the composite films, the hardness decreased while the adhesive strength did not change.

2. The composite films with high MoS 2 content (22%) showed low friction coefficient of around 0.1, which was almost equal to that of the DLC film.

3. Wear property of the composite films was improved by forming Ti interlayer. It was caused by the increased adhesive strength of the composite films with Ti interlayer.

6. Reference

Tribological Properties of Micro Sized Texture Surface Fabricated with Lower Flow Rate Shot Peening

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1. Introduction

It is well recognized that controlling surface texture is an effective means to reduce and stabilize friction resistance because the dimple and the groove act as oil reservoir and entrapment of wear debris. Many methodologies for fabrication of the surface textures have been applied depending on the machining scale and materials. Shot peening is one of the possible techniques as the dimples are formed on the surface. Recent development of the peening apparatus resulted in the surface texture consisting of micro dimples using ceramic particles as the impact media on the hard materials such as cemented carbide. The present study describes the applicability of the micro shot peening treatment as a fabrication method of the surface texture for the modification of the friction properties.

2. Surface texture by micro shot peening

A peening apparatus allowing for a flow of micro sized impact media at low rate was developed and applied to produce a surface texture. Figure 1 shows a schematic diagram of the developed peening apparatus. The impact media were enclosed with rubber balls for crushing cohesive fine particles in the storage tank, then were introduced to the inner part of the double walled nozzle with pressurized air. The impact media mixed with air were further accelerated with the higher pressurized air from the outer part at the tip of the nozzle. With this arrangement, it was possible to control the flow rate and the impact velocity of the media by adjusting the air pressure of outer/inner nozzle individually.

![Fig. 1 Schematic of lower flow rate peening apparatus](image)

After the peening, additional polishing was carried out to eliminate the pile up regions around the dimples. An optical micro image of the peened cemented carbide (ZN01, WC 10wt. % Co, 2000 HV) surface after polishing is shown in Fig. 2. The impact media were alumina beads (50 μm in diameter). Isolated micro dimples of 15 μm in diameter and 0.5 μm in depth, measured with a profile meter were, formed on the surface.

![Fig. 2 Optical image of peened cemented carbide](image)

3. Frictional properties

Friction properties of the textured cemented carbide disc were evaluated using a ring on disc test apparatus at a constant sliding speed of 0.1 m/s and various applied loads from 100 N to 1200 N using a PAO (50cst@40℃) as the lubricant. The mating ring was cast iron with the micro dimples fabricated by the peening and the parallel grooves by machining. The friction coefficient as a function of the applied load is shown in Fig. 3. The results obtained show that the friction coefficient of the textured surface is smaller than that of non-textured (flat) surface at higher load range. Particularly, the friction coefficient of the peened disc mated with parallel grooved ring was stable and the lowest at more than 400 N of the applied load.

![Fig. 3 Friction coefficients of various surface texture combinations as a function of the applied load](image)
A Newly Designed Surface Force Apparatus for Friction Measurements

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1. Introduction

In nanotribology studies there are three main kinds of measurement, namely Surface Force Apparatus (SFA), Atomic Force Microscopy, and Pin on Disk Tribometry. The SFA allows us to study surface forces and deformation versus friction performance using optical interference. We present an extension of SFA that will be closer to practical friction situations beyond the limitations of conventional SFA1, whilst retaining its advantages of allowing optical observation of surface separation, contact area and deformation.

2. Experimental: design concept

We employ ball on disk contact with rotation instead of the crossed-cylindrical contact used in conventional SFA which only allows a short travel length in back-and-forth motion. The new SFA has 3 capacitive sensors that provide displacement measurements at a maximum sampling rate of 45.5 Hz. Two of the sensors measure cantilever deflections resulting from surface force and normal load, and the third sensor measures lateral force. The SFA is designed to have 1 nm resolution in z axis motion by a piezoelectric stack, 1 mN to 10 N of normal load and friction force measured by a capacitive sensor, 0.1 mN to 1 N of surface force measured by optical interference, and approximately 0.3 millidegrees/step of minimum rotation by a stepping motor. To apply optical interference for measuring surface separation and deformation directly between two surfaces, transparent materials are required.

In this work, two thin films of silicon oxide/nitride2 as the explored transparent surfaces are deposited on a spherical lens and a flat glass substrate for ball on disk contact. In order to fabricate sample and probe surfaces, a thin film of silver underneath the silicon oxide/nitride is deposited as a semi-reflective layer for optical interference and germanium as an intermediate layer is used to reduce surface roughness of silicon oxide/nitride3, and to promote adhesion between glass/lens and silver and silver and silicon oxide/nitride.

3. Result: the apparatus setup

Figure 1 shows the SFA setup based on the design concept. The SFA probe is shown in Figure 2. The flat glass substrate has the same configuration of thin films, but with different thickness. The thicknesses of the thin films are important in terms of the performance of optical interference and the measurement resolution. 40 nm of silver and 3 μm of silicon oxide/nitride are applied. Evaluations and revisions of the apparatus are currently underway. Frictional characterizations between rigid surfaces of silicon oxide/nitride with and without lubricants will be investigated to demonstrate the capabilities of the SFA.

4. Conclusion

A new SFA has been developed to allow fundamental studies of tribology on the nano-scale. It can measure surface, adhesion, normal and lateral forces and deformation that allows characterization of the contact mechanics between two surfaces.

5. References

An Energetic Approach to Wear Analysis of Polymeric Composites

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1. Introduction

Polymers and polymer-based composites are increasingly used as dry sliding pairs in industry applications as lower weight alternatives to metals, where the self-lubrication of polymers is of special advantage. However, the frictional heat generation from dry rubbing contact can generally cause thermal softening and even the degradation of polymeric materials, which significantly affects their wear behaviour1. In the present work, the tribological performance of various polymer composites against a polished steel counterpart was investigated using different tribometers. An attempt was made to correlate friction, wear loss and the frictional heat from a general system point of view.

2. Experiments

The wear tests were conducted under dry sliding conditions at room temperature on three tribo-meters, i.e. two different pin-on-disk and one block-on-ring apparatus. The polymeric materials investigated were epoxy, PA 66, PEEK and PEI, reinforced by various tribo-fillers such as short carbon fibers (SCF), graphite flakes and TiO2 nano-particles. During the tests, the friction coefficient and the temperature of the steel counterparts were measured. After the tests, the specific wear rate was calculated based on the mass loss of the specimens.

3. Results

In the steady state wear stage, the frictional energy dissipated can be characterized by the friction power intensity, qf, determined by qf = μpν, where μ is the mean value of the friction coefficient in the steady state range, p is the normal pressure applied to the specimen, and ν is the sliding velocity.

As shown in Fig. 1, the temperature rise of the steel counterparts is almost linearly proportional to the friction power intensity, i.e. ∆T = Tc - RT = k · qf, where Tc is the temperature of the steel counterpart measured within the steady state range, and k is the factor of the thermal diffusion for the tribo-system. It was found that the slope, k, is mostly dependent on the configuration of the tribometer, but is independent of the polymeric specimens or the sliding conditions.

The relationship between the time-related depth wear rate and friction power intensity can be generally described by, wT = α(γ · qf)β, where α is the wear factor depending on the contact mode, γ is the lubricating factor depending on the properties of the transfer film developed on the steel counterpart, and β represents the thermal effect due to the frictional heat. When the contact temperature is relatively low, β = 1 (cf. Fig. 2).

Fig.2 The time-related depth wear rate of two PA 66 based composites as a function of friction power intensity. The inserted SEM pictures show the worn surfaces of graphite+SCF/PA66 tested under 2MPa, 1m/s (left side) and 2MPa, 3m/s (right side).

4. Conclusions

The dry sliding wear behaviour of the polymer composites was investigated using several tribometers. It was found that when the contact temperature was relatively low, the specific wear rate of the specimen was stable and mostly governed by the reinforcements used. However, the wear rate of the material increased exponentially when the contact temperature exceeded a critical value. The upper-boundary contact temperatures are related to the critical temperature parameters of polymer matrices such as softening point or melting point.

Acknowledgement

Dr. Li Chang greatly appreciates the financial support of the Australian Research Council.

5. References

Development of Electrodes with Micro Ploughing Patterns for MEMS Applications

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1. Introduction

Nanoelectrode lithography is a recently developed method for the direct lithography on silicon wafers without resist coating and mask preparation in reference to the local anodic oxidation by scanning tunneling microscope. The electrodes used for this method have nano/micro patterns of concavity and convexity that are prepared using the electron beam or focused ion beam. This is thus time-consuming.

This paper reports the manufacture of electrodes with precision protrusion and concave patterns using micro ploughing. The scratching tests were performed to fabricate the electrodes and the local anodic oxidation was subsequently carried out using the developed electrodes. The method is simple and without chip formation. It is thus suitable for ultra-precision processing.

2. Manufacturing of Micro Ploughing Patterns

An ultra-precision lathe (ULG-100 Ser., Toshiba Machine Co. Ltd.) was used. High-speed scratching experiments that simulate the face turning were conducted on a fine oxygen-free copper workpiece surface (Ra = 56 nm) by using a triangular monocrystalline diamond indenter with 100 nm tip radius.

Fig. 1 shows the cross-sectional profile of the manufactured ploughing pattern measured by a surface roughness tester (SV-9624-3D System, Mitutoyo Co.). As shown in Fig. 1, a micro ploughing pattern with a pile-up height of 1 μm can be observed at the right shoulder of the scratched groove.

3. Micro/nano Ploughing Electrode Lithography

Fig. 2 schematically illustrates the micro/nano ploughing electrode lithography developed here. The local anodic oxidation occurred through the reaction between the developed electrode and a Si(100) bare wafer, when a proper voltage potential was applied between them.

Fig. 3 shows the typical AFM (SPA-300HV, Seico Instruments Inc.) image of the anodic oxidation pattern (SiO2), which was generated using the manufactured micro ploughing electrode. In this specimen, a nano protrusion pattern of 2 nm in height (marked by the dotted line) was successfully generated on a Si(100) surface. The SiO2 patterns can then be used as the mask for alkaline etchants.

4. Conclusions

Scratching was performed on a fine oxygen-free copper electrode specimen using a sharp diamond tool to generate micro ploughing patterns. Local anodic oxidation experiments were also performed on a silicon wafer using the manufactured electrode. The AFM examination showed that nano protrusion patterns of features of 2 nm were fabricated based on the anodic oxidation of silicon.

5. References

Influences of Boundary Conditions at Air-water Interfaces on Bubble Coalescence
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1. Introduction

Knowledge of the basic phenomena involved in bubble coalescence is important in understanding various gas/liquid systems that have a great number of industrial, environmental and biological applications. Film drainage and bubble coalescence behaviours are determined by a balance between surface tension, hydrodynamic and surface forces. However, not much systematic work has been done to investigate dynamic effects on bubble coalescence. We demonstrate the effects of speed of approach and salt concentration on bubble coalescence by two independent experimental systems.

2. Experimental

One system is film thinning experiments conducted with aqueous films between two air phases in a thin film pressure balance (Fig 1). The experiments begin with a comparatively large volume of water in a cylindrical capillary tube a few millimeters in diameter, and by withdrawing water from the centre of the tube the two bounding menisci are drawn together at a prescribed rate. This models two air bubbles approaching at a controlled speed. The other experimental system is a sliding bubble apparatus in which mm-sized bubbles in an aqueous medium rise toward an air meniscus at different speeds obtained by varying the inclination of a closed glass cylinder (Fig. 2). The coalescence times of bubbles contacting the meniscus are monitored.

3. Result

Experimental results from both systems showed the significance of bubble approach speed as well as electrolyte concentration in affecting bubble coalescence, in other words, the stability of aqueous films. In pure water, the results show three regimes of behavior depending on the approach speed: at slow speed a stable film is formed due to repulsive double-layer interactions between naturally charged air/water interfaces; at intermediate approach speed, the films are transiently stable due to hydrodynamic drainage effects; and at fast approach speed, the bubbles coalesce instantly without forming films because the hydrodynamic resistance appears to become negligible. A simple model is developed that accounts for the boundaries between different film stability or coalescence regimes. The model is able to give reasonable estimates of the transition speeds between the observed regimes of behavior in pure water. It can also explain the effects of added electrolyte which essentially kills the low-speed (stable film) and high-speed (rapid coalescence) regimes observed in pure water.

4. Conclusion

The present work highlights the significance of bubble approach speeds in determining bubble coalescence, and points to a connection between approach speeds and the transition concentrations at which inorganic electrolytes have been observed to inhibit bubble coalescence. The results imply that the interplay between boundary conditions at the air-water interfaces and surface forces acting between the air-water interfaces determines the bubble stability.

5. References

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Worn Surfaces of Diamond-like Carbon in Water and Air Environments

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1. Introduction

It is known that diamond-like carbon (DLC) films show excellent tribological properties. The films also show the low-friction coefficients and low-specific wear rate in air and water environments1. However, the origin of low friction properties of DLC film has been unclear. Previously, we proposed the friction mechanism of DLC film against a stainless steal ball in a water environment2. It was considered that the lubrication film generated by tribochemical reaction resulted in the low friction properties. In this study, we investigated the tribochemical reaction of the worn surfaces of DLC film.

2. Experimental

The DLC films were deposited on Si substrate by electron-excited plasma CVD method with negative bias voltage3. A tribological test was performed in deionised water and ambient air with AISI 440C as the mating ball. The properties of scar surfaces were evaluated by the contact angle against water droplets. Furthermore, the adhesion forces of the worn surface were investigated based on atomic force microscope (AFM) measurements with Si chips. The tribochemical reaction in worn surfaces of DLC film was confirmed by the time-of-flight secondary ion mass spectroscopy (TOF-SIMS). Positive and negative secondary-ions were detected in a 180 mm square of analysis area9.

3. Results

The worn surfaces of DLC film were changed to hydrophilic surfaces after the tribological tests. As shown in Fig. 1, the contact angle obtained in ambient air decreased to lower angle compared with that in water environment. On the other hand, the specific wear rates of DLC film were 5.9 x 10^-8 and 2.1 x 10^-8 mm^3/Nm in air and water environments, respectively. The roughness of worn surface obtained in air was higher than that in water. Thus, it is considered that the difference in contact angles was caused by the change in chemical properties of the surfaces. The adhesion force of DLC film was almost zero. However, for worn surfaces these were -121 and -82 nN, in air and water, respectively. On the other hand, the FWHM (full width at half maximum) of the distribution of adhesion forces in air was larger than that in water. Therefore, the worn surface in air and water was changed to hydrophilic surfaces by tribochemical reaction. Furthermore, it is considered that the worn surface in air was ununiformity compared with that in water.

TOF-SIMS analysis has indicated the chemical modification of worn surfaces by the tribochemical reaction. The oxygen and hydroxide groups were located in the worn surfaces. Furthermore, the worn surface of DLC film tested in H_2^{18}O showed that the origin of oxygen in hydroxyl group was from water molecules. Therefore, it is considered that the tribological properties of DLC films against a stainless steel ball were affected by water.

![Fig.1 Contact angles of worn surfaces of DLC film.](image)

4. Conclusions

The tribochemical reaction of worn surfaces of DLC film in air and water environment was investigated by the contact angle, adhesion force, and TOF-SIMS. The following results were obtained:

(1) The friction of DLC film against a stainless steel ball in air and water environments was influenced by tribochemical reaction occurring on the worn surfaces.
(2) The worn surfaces of DLC film were changed to hydrophilic.
(3) DLC films reacted readily with water and/or air by the tribochemical reaction.

5. References

Application of FT-IR Method to Oil Film Thickness Measurement on Rough Surface with Low Reflection Intensity

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1. Introduction

Oil film of porous frictional material for a wet clutch is important to improve shift feeling and cooling performance of clutch. FT-IR (Fourier Transform Infrared Spectroscopy) method was applied to measure the oil thickness in this study. As a first stage of this FT-IR application to the porous frictional material, having a rough surface with low reflection intensity, an experiment for improvement of reflection intensity was conducted for the porous frictional material. Simulations and measurements of absorbance were carried out for a simplified slope to investigate the effects of rough frictional surface on the absorbance.

2. Experimental Procedure

Absorbance of infrared spectrum is defined by equation (1). It is known that absorbance is proportional to sample thickness (Lambert-Beer law).

\[ A(\sigma) = \log_{10} \frac{E(\sigma)}{E'(\sigma)} \]  

\( A(\sigma) \): Absorbance  
\( \sigma \): Wave number  
\( E(\sigma) \): Background intensity  
\( E'(\sigma) \): Sample intensity

Platinum thin film was sputtered on the surface of porous frictional material to improve reflection intensity. Background intensity, representing reflection intensity on frictional material surface, was measured as shown in Figure 1(a). Ratios of absorbance increase with the optical path length change and the apparent increase, measured by missing reflection light coming out of detector, are discussed. The effects of reflection on a rough surface of porous frictional material on absorbance were clarified. Simulations were carried out using the slope model with simplified roughness of porous frictional material as shown in Fig. 1(b). Absorbance was measured using an aluminum model test piece under a variety of inclined angle of reflection surface.

3. Results and Discussion

The reflection intensity of infrared light improved by 70% and 100% at wave number of 1400cm\(^{-1}\) and 2900cm\(^{-1}\) respectively by sputtering platinum thin film on a surface of frictional material as shown in Fig. 2. The simulation results, shown in Fig. 3, demonstrated that the ratio of the apparent absorbance increased with increasing inclination angle of the simplified slope. The ratio of absorbance increase with optical path length was negligible. The absorbance increased in the inclination angle of 0°-10° and decreased over that of 15° for the experimental results as shown in Fig. 4. Possible reason for the discrepancy between experimental and simulation results is that there might be a possibility that a part of measured area was out of focus due to the inclination of the reflection surface.

4. Conclusions

The effect of platinum thin film on the surface of porous frictional material on background intensity, and the effect of inclination angle of the reflection surface on absorbance are discussed in this study.

(1) A sputtered platinum thin film on the surface of porous frictional material is effective for an improvement of the reflection intensity.

(2) It is recognized that the reflection on the slope of rough frictional surface increases the absorbance of infrared light spectrum in FT-IR method.

5. Reference

Retention Behavior by Liquid Film of Inverted Drive Mechanism

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1. Introduction

The precise and smooth motion of the precision machine tool slideways or HDI under the mixed or boundary lubricating conditions is disturbed strongly by the adhesion force of thin liquid film. It is considered that this adhesion force is caused by the surface tension of the liquid film and the negative pressure based on atmospheric pressure etc. generated in the liquid.

On the other hand, it is expected that this adhesion force makes a inverted smooth drive mechanism by the retention force of liquid film possible. In this mechanism, it is expected that the direct contact between the slider and the guide plate can be avoided because that the slider is hanged by the adhesion force of the liquid on the guide plate.

Therefore, in this report, the behaviors of glass slider suspended to the upper glass plate by the adhesion force of the liquid were experimented under several liquid film thicknesses of distilled water and mineral oils that are different viscosity without additives, and examined the relationship between retention behavior by Laplace force on the surface tension of the liquid pulling off (falling down) behavior.

2. Experimental apparatus and method

![Experimental apparatus](image)

Fig.1 Experimental apparatus

Figure 1 shows the experimental apparatus. Two experiments were done using this apparatus. One (Exp.1) is to examine the maximum static friction coefficient of the liquid. This experiment was done to lift the right side of the glass plate by the height gage and to measure the slope angle of the beginning to slide of the slider adhered to the underside of the glass plate. And another (Exp.2) is to measure the liquid film thickness when the slider pulls off (falls down) from the glass plate. In this experiment, the glass plate is kept in horizontal. The average liquid film thickness in this experiment was obtained dividing the known liquid volume measured with micro pipet by the slider area in which the liquid is spread over the area. The diameters of the glass slider are 20, 30, 40, 50 mm, and the weight is same of 0.35 N. Table 1 shows the kinds of liquid used in the experiments.

<table>
<thead>
<tr>
<th>Table 1 Kinds of liquid used and viscosity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid</td>
</tr>
<tr>
<td>----------------</td>
</tr>
<tr>
<td>Distilled water</td>
</tr>
<tr>
<td>Paraffin oil</td>
</tr>
<tr>
<td>P8</td>
</tr>
<tr>
<td>P32</td>
</tr>
<tr>
<td>P46</td>
</tr>
<tr>
<td>P68</td>
</tr>
</tbody>
</table>

3. Experimental results and consideration

Figures 2(a), (b) show the relationship between film thickness and friction coefficient to the normal friction state (namely, slider is set upper side of glass plate) and the reverse type experiment (Exp.1) in the case of distilled water. As seen in the figures, the friction coefficient of the reverse type experiment is not disturbed, compared with normal friction state.

![Graph](image)

(a) Normal friction

(b) Reverse type friction

Fig.2 Relationship between film thickness and friction coefficient

It is considered that the result of Fig.2 (b) is obtained by both effects of the retention behavior of the liquid based on the Laplace force and the non solid contact state between the slider and the glass plate based on the gravity.

4. References

Evaluation of Tribological Properties of Palm Oil on Improved Pendulum Type Friction Tester

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1. Introduction

Recently, palm oil has been gaining acceptance as a lubricant as it exhibits a high biodegradability, and contains many components that improve its lubricating performance. In this research, an improved pendulum type friction tester is used to accurately measure oiliness effect of fatty acid present in the palm oil. The objective of this work is to evaluate the tribological properties of palm oil. Special attention is paid to the boundary lubrication characteristics, i.e. friction and wear, of palm oil under different temperatures. As a reference, the tribological properties of mineral oil are also evaluated to compare with those of palm oil.

2. Experimental Details

During the friction test, the dumped vibration of conventional friction pendulum tester (Fig. 1) is measured. Wear volume of four balls A, B, C, D located in test chamber is measured after number of pendulum swings. The friction tester of pendulum type was originally developed to evaluate the oiliness of lubricants operating in a boundary lubrication regime. A pin is attached to the swinging pendulum, and four steel balls A, B, C, D are mounted in the test chamber. The coefficient of friction is calculated from the following expression:

\[
f = C \cdot \frac{A_0 \cdot n - (A_1 + A_2 + ... + A_n)}{1 + 2 + ... + n}
\]

where:
- \(f\): Coefficient of friction,
- \(n\): Number of swing
- \(A_0\): Initial angle,
- \(A_n\): \(n\)-th swing angle, and
- \(C\): Proportion constant (8.16)

The major components of the conventional friction tester are: pendulum’s magnet, heater, sensor and drive circuit with magnet coil and transistor. These components enable the continuous swinging of the pendulum. Transistor is used to apply electrical current to magnet coil, which pulls the magnet mounted on the pendulum. Timing circuit turns on and off magnetic coil keeping the pendulum swinging. With this improved pendulum number of swings can be controlled/specified. Number of swings is counted and a heater can control the temperature of oil.

Table 1

<table>
<thead>
<tr>
<th>Total Load</th>
<th>3.38N</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum pressure</td>
<td>1.13GPa</td>
</tr>
<tr>
<td>Period per swing</td>
<td>2.56s</td>
</tr>
</tbody>
</table>
Influence of Vitamin E on Lubricating Properties of Vegetable Oils with Different Fatty Acid Types in Four-Ball Tests

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1. Introduction
As promoted by Green Tribology from the viewpoint of natural environmental requirements, applications of biodegradable lubricants are growing. Vegetable oils such as rapeseed oil and soybean oil are biodegradable and have good lubricating performance even at high contact pressure conditions in spite of their lower pressure-viscosity coefficient. The lubricating performance of vegetable oils with different fatty acid types changes depending on the severity of operating conditions1). In low temperature range, the adsorbed films play important roles but in high temperature range the friction polymer and/or reacted films dictate the tribological behavior, which can be controlled with antioxidant additive. Tribological properties of several kinds of vegetable oils with and without additive were evaluated in four-ball tests.

2. Experimental methods
The four-ball tests were carried out under the increasing-temperature condition at a constant rate of 5 °C/min until 200 °C under the constant load at 1 rpm (sliding speed = 0.58 mm/s) and 100 rpm (sliding speed = 58 mm/s) to evaluate the lubricating properties at different lubricating film thickness. Friction and load-carrying properties were examined for five kinds of vegetable oils, i.e., olive, high oleic canola oil, high erucic rapeseed, soybean and linseed oils with low viscosity (25 – 45 mm²/s at 40 °C) and structures with different fatty acid types in triglycerides (Table 1). The effectiveness of addition of 1.0 mass % vitamin E (alpha-tocopherol) as an antioxidant additive was examined.

Table 1 Fatty acid types in structure of vegetable oils

<table>
<thead>
<tr>
<th></th>
<th>Palmitic acid *(C16:0)</th>
<th>Stearic acid *(C18:0)</th>
<th>Oleic acid *(C18:1)</th>
<th>Linoleic acid *(C18:2)</th>
<th>Linolenic acid *(C18:3)</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Olive oil</td>
<td>12.8%</td>
<td>3.0%</td>
<td>74.5%</td>
<td>7.3%</td>
<td>0.8%</td>
<td></td>
</tr>
<tr>
<td>High oleic canola oil</td>
<td>4.2%</td>
<td>2.6%</td>
<td>73.7%</td>
<td>12.2%</td>
<td>3.9%</td>
<td></td>
</tr>
<tr>
<td>High erucic rapeseed oil</td>
<td>2.8%</td>
<td>1.1%</td>
<td>14.3%</td>
<td>12.8%</td>
<td>8.5%</td>
<td>Eicinolic acid *(C22:1) 47%</td>
</tr>
<tr>
<td>Soybean oil</td>
<td>10.1%</td>
<td>4.2%</td>
<td>23.7%</td>
<td>53.7%</td>
<td>7.5%</td>
<td></td>
</tr>
<tr>
<td>Linseed oil</td>
<td>5.3%</td>
<td>1.2%</td>
<td>20.2%</td>
<td>15.1%</td>
<td>52.2%</td>
<td></td>
</tr>
</tbody>
</table>

(*Numerals mean numbers of carbon and double bond)

3. Results and discussion
The friction in boundary lubrication at 0.58 mm/s indicates particular properties of fatty acid types. Olive oil shows low friction at low temperature but an increase in friction at high temperature. Soybean oil with a large amount of linoleic acid exhibited lowering of friction at high temperature range. In contrast, similar friction levels ranging from 0.06 to 0.1 for all vegetable oils were observed in mixed lubrication at 58 mm/s. However, scuffing loads at 58 mm/s are obviously different depending on fatty acid types as shown in Fig.1. Furthermore, addition of vitamin E is effective/beneficial for high oleic canola, high erucic rapeseed and soybean oils (Fig.1).

Under high temperature conditions, the appropriate formation of friction polymer enhances the lubricating performance but excessive friction polymer film prevents the supply of oils into the conjunction (Fig.2), which reduces the lubricating performance. Therefore, appropriate selection of an additive such as vitamin E as an antioxidant is required to control the actual lubrication mechanism in vegetable oils with different fatty acid types.

4. References
Study on Rolling Friction of Ball Bearing Lubricated with Minute Amount of Grease under Low Speed Condition

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1. Introduction

A low friction linear guide-way becomes to be a key device for high precision machines. Therefore, reducing friction of the linear guide-way lubricated with grease is strongly required, especially under low speed condition, because churning loss of grease relatively increases due to low temperature rise.

It was found that rolling friction of a ball bearing under low speed condition can be considerably reduced if the ball bearing is lubricated with a minute amount of diurea grease, whereas the rolling friction does not decrease if using Li-soap grease. Furthermore, it was also found that its reducing effect depends on the degree of mixing and kneading in a grease production process. In this study, the mechanism of this decrease of friction is experimentally investigated.

2. Test grease and methodology

Some types of diurea grease, which consist of same base oil and thickener but only the different structure and micelle size of thickener, are prepared as listed in Table 1. The rolling friction under low speed condition is measured when the thrust ball bearing (51206) is lubricated with a minute amount of test greases. Variations of the shapes of channeling grooves can be measured by infrared interferometry and the degree of oil separation can be also evaluated by intensity ratio of CH bond absorption spectrum (2920cm⁻¹) to that of NH bond (3280cm⁻¹) measured by FT-IR.

3. Results and discussion

From the measurements, it is clarified that well-kneading grease(U2) which has the small sized and isotropic structure of the thickener exhibits very low friction as shown in Fig. 1. In the case of U2, the stiff channeling groove is generated and low film thickness in the Hertzian contact region as shown Fig. 2. Furthermore, It is found that moderate oil separation appears in the Hertzian contact region because the intensity ratio of CH bond absorption in that region indicates larger magnitude than that of side band region as shown in Fig. 3.

In consequence of both phenomena, just the proper quantity of the separated base oil to preserve adequate starved lubrication which leads to low friction might be kept in the rolling track.

4. References

Rheology of Nano PFPE Liquid Film at Hard Disk Surfaces under Blow Off Flow

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1. Introduction

Hard disk surfaces are covered with a perfluoropolyether (PFPE) film of 1-2 nm thickness. The lubricant film protects the hard disk from corrosion, contamination and friction damage. Although the magnetic head is flying over the hard disk surface during operation, the friction damage may occur due to the intermittent contact of magnetic head sliders. The lubricant film must be complete, sustainable during the service life time, and self-reparability is desirable in case that minor friction damage occurs during service. On the other hand, the lubricant film is subjected to the shear stress from both air bearings of magnetic heads and centrifugal flow of air. The rheological behavior of lubricant under shear stress may play an important role in the function of the lubricant in spite of the fact that it is as thin as a monolayer. It is clear that the rheological behavior of such a thin film is dramatically different from that of bulk.

In this paper, the blow off experimental result on both the functional and nonfunctional PFPE is presented.

2. Experimentation

A micro-tunnel is made on a measured surface that is partially coated with a liquid film. A gas stream (nitrogen gas) is supplied from a pressurized gas source through the micro-tunnel. The micro tunnel is $L=12.5\,\text{mm}$ in length, $B=4\,\text{mm}$ in width and $H=40\,\mu\text{m}$ in height. The pressure drop $P$ across the length is 0.1 MPa, and therefore the shear stress acting at the liquid film surface $\tau = P \times H / 2L = 160\text{Pa}$.

3. Experimental Results and Discussion

Two different types of perfluoropolyether (PFPE) were used in this study: nonfunctional PFPE Formblin Z25 and functional PFPE Zdol 3000.

The film thickness distributions in the region around the boundary line of the liquid film at different test times are shown in Fig. 1 for Z25 and in Fig. 2 for Zdol3000. It is seen, from Fig. 1 and Fig. 2, that the Z25 and Zdol behaved completely differently. For Z25, the front line moved from A to B, C, and D in the blow direction during the test time periods of 30 minutes, 60 minutes and 90 minutes. The distances were $X_{AB}=0.75\,\text{mm}$, $X_{BC}=0.58\,\text{mm}$, and $X_{CD}=0.92\,\text{mm}$. The average velocity of the front line is 0.42 $\mu\text{m/s}$ from A to B, 0.32 $\mu\text{m/s}$ from B to C, and 0.51 $\mu\text{m/s}$ from C to D, and the mean velocity from A to D was 0.42 $\mu\text{m/s}$. The reason for the variations in the measured velocity of the liquid film is not clear at present, but it may be due to the accumulation of the liquid at the front area, which changed the film thickness in the area. It is clear, from Fig. 1, that the pileup growth occurred for Z25. This is due to the fact that the micro slip occurred at Z25/hard disk interface, and the micro slip velocity decreased steeply with the distance from the front end.

In addition, the front curve in Fig. 1 shows a ‘knee’ followed a sharp drop. This suggests that PFPE Z25 film has a layered structure due to the pileup and the pileup layer has lower viscosity than the original layer.

4. Conclusion

Blow off experiments on PFPE Zdol3000 and Z25 were conducted. It is concluded that

(1) Micro slip occurs at a weakly adsorbed lubricant/hard disk interface, such as PFPE Z25.
(2) There is no micro slip for PFPE Zdol3000.
(3) Mobile layer may lead to pileup growth.
(4) The pileup layer has a lower viscosity than the original layer of PFPE Z25.
Studies on Three-Dimensional Digital Characterization of Coating Defects and the Effects of Coating Defects on Its Tribological Properties

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1. Introduction

High-speed and high-precision Computer Numerical Control (CNC) machining is the new development in the modern material processing, which makes high demands on the tools. As the tool’s coating has many surface defects understanding of their formation mechanisms is of importance. However, classification and characterization of the tool’s coating defects must be conducted first.

In this paper a 3-D digital model of the tool’s coating defects is established and preliminary studies on their influence of the tribological properties are conducted.

2. Experiment Instrumentation

The scanning electron microscope JSM-5610 was used to examine a large region within the plane of coating defects, while the 3-D characterization of the defects and grain size was measured using XE-70 AFM. Larger coating defects were characterized using white light confocal microscope (WLCM). A series of experiments with different coatings was carried out on Wazau pin-on-disc abrasion machine.

3. Classification of Coating Defects

SEM, AFM and WLCM were used to measure the carbide cutting tool coating defects. A comparison between tool’s defects is shown in Fig. 1. Areas of exposed substrate material are shown in Fig. 1b while macroparticles are shown in Fig. 1c.

The coating defects are classified as: macroparticles (Fig. 2a), shallow protrusions (Fig. 2b), areas of exposed substrate material (Fig. 2c), pin-hole defects (Fig. 2d). The sizes of these defects are shown in Table 1.

4. 3-D Digital Model of the Coating Defects

Based on the AFM data of 3-D morphology of coating defects, such as macroparticles, a 3-D digital model (Fig. 3a) was built. The model calculates the volume of macroparticles using SPIP and Tablecurve-3D software, Figs. 3b and 3c respectively.

Table 1 Size and the classification of TiN coating defects

<table>
<thead>
<tr>
<th>Defects</th>
<th>Width</th>
<th>Height</th>
</tr>
</thead>
<tbody>
<tr>
<td>Macroparticles</td>
<td>D &lt; 30µm</td>
<td>2 µm</td>
</tr>
<tr>
<td>Shallow protrusions</td>
<td>D &lt; 10µm</td>
<td>500 µm</td>
</tr>
<tr>
<td>Areas of exposed</td>
<td>D &lt; 400µm</td>
<td>14 µm</td>
</tr>
<tr>
<td>substrate material</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pin-hole defects</td>
<td>D &lt; 5 µm</td>
<td>500 nm</td>
</tr>
</tbody>
</table>

5. Effect Studies of Coating Defects on the Tribological Properties During Machining

The coatings with small defects and the coatings with large spalls were tested. The results show that the wear and friction coefficient of the second sample with large spalls are higher (Fig. 4).

6. Conclusions

In this paper an attempt was made to classify and characterize the coating defects. It is important to study the formation of tool’s coating defects and to quantify the effects of the defects on the tribological properties and the cutting performance of tools.

Based on 3-D numerical characterization, it has been shown that it is possible to establish a mathematical model of the defects.

7. Acknowledgement

The authors are grateful for the financial support from the major projects for science and technology development of China under Contract No.2010ZX04014-071.
Dielectric Relaxation Behavior of Polyolester Oil Based Greases with Lithium 12-Hydroxy Stearate Thickener

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1. Introduction

When an alternating electric field acts on the material that has the permanent dipole moment, dielectric dispersion and absorption are observed, and the mobility of dipole moment can be evaluated from the dielectric relaxation frequency. Therefore dielectric spectroscopy is a useful tool to probe molecular motions in materials. If this method is applied to lubricating grease, it is expected that the information about the mobility of the grease could be obtained. It is well known that the complicated flow properties of lubricating grease are brought by deformation, breakdown and reconstruction of the thickener network and interaction between the base oil and the thickener. Dielectric relaxation measurements of greases might give the findings about the change in thickener network and interaction between base oil and thickener. In this study, we made the dielectric relaxation measurements of polyolester-based greases with lithium 12-hydroxy stearate.

2. Experimental

A multilayer parallel plate capacitor with approximately 75 pF air capacitance at 25 °C was used for the measurements. The gaps between electrodes were filled with sample grease and equivalent parallel capacitance, \( C_p \), and equivalent parallel conductance \( G \) of grease-filled capacitor were measured in the frequency range from 20 Hz to 2 MHz with LCR meter. Then relative dielectric constant, \( \varepsilon' (=C_p/C_0) \), and relative dielectric loss factor, \( \varepsilon'' (=G/\omega C_0) \), were calculated; where \( C_0 \) is capacitance of air-filled capacitor and \( \omega(=2\pi f) \) is angular frequency. Dielectric measurements were conducted at temperature from –80 to 70 °C.

Neopentyl glycol ester of C9 fatty acid and lithium 12-hydroxy stearate was used for the base oil and the thickener of sample greases, respectively.

3. Results

Figure 1 shows the dielectric relaxation curves of the sample grease at three different temperatures. Closed circles and open circles indicate the dielectric permittivity and the dielectric loss, respectively. Only one dielectric loss peak was observed in this frequency window at –70.7 °C. This loss peak shifted to high frequency region and another loss peak appeared in low frequency region at –53.6 °C. At the higher temperature of –1.9 °C, loss peak of higher frequency side disappeared completely.

Figure 2 shows the Arrhenius plots of the dielectric relaxation time \( \tau \) of sample grease. In this figure results of PAO based grease with lithium 12-hydroxy stearate thickener and polyolester base oil were plotted together. By comparing with the results of PAO based grease, it became clear that the relaxation peak appearing in the higher frequency region was caused by the orientation of hydroxyl group in the thickener. Relaxation time of this mode was longer than that observed for the PAO based grease. On the other hand, the relaxation peak in the lower frequency region was caused by the permanent dipole moment of the base oil. Relaxation time was almost the same as that of the polyolester base oil.

4. Conclusions

Two clear dielectric relaxation peaks were observed for the polyolester oil based grease with lithium 12-hydroxy stearate thickener. These two relaxations showed different temperature dependency.
Comparison of Frictional Behaviors between Remelted and Annealed Highly Crosslinked UHMPWEs against Orthopaedic Grade Co-Cr alloy in Total Joint Replacements

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1. Introduction

UHMWPE is the most widely-used liner material in total joint replacement. Nowadays, conventional gamma-irradiated UHMWPE is often replaced with highly crosslinked UHMWPE (XLPE). The typical XLPEs are the remelted highly crosslinked UHMWPE (RXLPE) and the sequentially irradiated and annealed highly crosslinked UHMWPE (AXLPE)¹². To analyze friction behavior, it is necessary that test is reflected various conditions (contact pressure and lubrication mechanism). The nominal contact pressures at articulating surfaces of different joints are 5MPa in THR (hip), 10MPa in TKR (knee), and 20MPa in TDR (disc). In addition, friction tests should simulate different lubrication conditions such as fully-immersed lubrication with the presence of joint synovium, and rarely lubrication with body fluid without joint synovium. The purpose of this study is to compare frictional behavior between the commercialized RXLPE and AXLPE sliding against Co-Cr alloy including the effects of changing contact pressure and lubrication condition.

2. Experimental and Methods

Two kinds of XLPE pins were machined from hip liners (RXLPE, Zimmer) and knee liners (AXLPE, Striker) to cylindrical shapes (diameter 10mm, length 10mm). Flat disks were machined from orthopaedic grade Co-Cr alloy rod and were polished to the surface roughness (Ra) of 0.005 μm. Friction tests were conducted by using a pin-on-disk type tribotester in a fully-immersed and a rarely lubrication conditions with bovine serum under 5, 10, and 20MPa. For rarely lubrication tests, 0.05ml of lubricant was dropped onto a Co-Cr alloy disk surface every minute. All tests were performed in a repeat pass rotational sliding motion with 60 rpm for a total of 5 minutes. Each friction test was repeated six times (n=6). The coefficient of friction was determined by dividing the measured frictional force by the applied normal force. The average coefficient of friction was calculated at about 200–300 seconds after the steady state has been reached. To analyze the effect of UHMWPE type, student t-test was conducted and p-values less than 0.05 were used to determine the statistically significant difference.

3. Results and Discussion

The average coefficients of friction of RXLPE under 5, 10, and 20MPa in a fully-immersed lubrication condition were 0.0310, 0.0247 and 0.0183, and those in a rarely lubrication condition were 0.0314, 0.0261, and 0.0211, respectively. The average coefficients of friction of AXLPE under 5, 10, and 20MPa in a fully-immersed lubrication condition were 0.0378, 0.0343 and 0.0237, and those in a rarely lubrication condition were 0.0480, 0.0423, and 0.0270, respectively. The average coefficients of friction of RXLPE and AXLPE decreased as the contact pressure increased. The average coefficients of friction of RXLPE were significantly lower than those of AXLPE under each contact pressure in both lubrication conditions (p<0.05). Previous studies reported the frictional behavior of non-irradiated UHMWPE (NGI) and gamma-irradiated UHMWPE (GI) against Co-Cr alloy. The average coefficients of friction of NGI and GI under 5, 10, and 20MPa in a fully-immersed lubrication condition were in the range of 0.0240-0.0409, which were much higher than those of RXLPE (p<0.05)³⁴.

![Fig.1. Average coefficients of friction of both XLPEs for each contact pressure and lubrication condition.](image)

4. Conclusions

In conclusion, average coefficients of friction of RXLPE were significantly lower than those of AXLPE. RXLPE showed much lower coefficients of friction than conventional UHMWPE (GI).

5. References


6. Acknowledgement

This study was supported by the National Research Foundation of Korea grant funded by MEST (2010-0000486).
Estimation of the Tire-Road Friction Based on the Inner Surface Deformation of a Tire

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1. INTRODUCTION
The intelligent tire employing various sensors was developed to enhance vehicle safety and comfort. The role of the tire is to generate traction, braking, and side force. As the origin of these forces is friction, the tire-road coefficient of friction plays an important role in driving cars. Therefore, a study to derive the friction coefficient during driving is essential [1]. The purpose of this study was to estimate the tire-road coefficient of friction based on the inner surface deformation of a tire.

2. EXPERIMENTAL
A parallel stereo method was used to measure the deformation of the inner surface of a tire. The basic idea of the proposed method is shown in Fig.1. With this method 3D positions of the measurement points, based on the principle of the triangulation between two images from two cameras installed in parallel, can be obtained.

A cut tire was rubbed against various mating surfaces in sliding or turning and friction force was measured with a strain gauge glued to parallel leaf springs. The inner surface of a tire is photographed with two cameras installed in the tire wheel rim.

In this experiment, the deformation of the tire inner surface and the friction force were detected as the mating surface was rubbed against a fixed tire or as the cut tire was turned in relation to a stationary mating surface.

The tire specimen was cut from a commercial radial tire (175/70 13R). The measurement points were selected in the tire inside at regular intervals. An aluminum plate, PTFE sheets, and abrasive paper were used as the mating surfaces. The sliding speed was 2.7mm/s when a mating surface moved and 18.2mm/s when the cut tire was turned. The deflection along the z axis ($\Delta z$) and the circumference deformation of a cut tire along the y axis ($\Delta y$) were calculated from the measurement point at the center of the tire.

3. RESULT AND DISCUSSION
Figure 2 shows the relationship between a normal load and the z value of the tire specimen. Regardless of the mating surface tested, z of the tire was proportional to the normal load, but the differences between the loading and unloading were also observed.

The relationship between the $\Delta y/\Delta z$ of the cut tire and the friction coefficient when the tire specimen rubbed against the mating surface is shown in Figure 3. The coefficient of friction tended to increase when $\Delta y/\Delta z$ increased. The friction coefficient deviated from linearity at a tire turning. As a result, the tire-load friction can be estimated during driving since the coefficient of friction is almost proportional to the $\Delta y/\Delta z$.

4. CONCLUSION
A parallel stereo method was adopted to derive the relationship between tire deformation and the coefficient of friction with the following results:
1. The inner surface deformation of a tire can be measured.
2. The coefficient of friction is almost proportional to the $\Delta y/\Delta z$ of cut tire deformation.

5. REFERENCE
Frictional and Rebound Characteristics of Tennis String of Flat-type Cross Section during Simulated Impact Tests

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1. Introduction

Tennis strings in wide variety of materials and structures are commercially available. In this study, rotation speed and displacement of a ball were monitored during impact between rotating ball and single tennis string to obtain information related to the effects of the string cross section shape on frictional characteristics.

2. Experiments

Impact test is conducted as follows. A tennis ball is set in the ball unit where it can rotate freely. The ball unit moves vertically to impact against a tennis string. The string is tensioned at a given load in the direction parallel to the ball rotation. After the initial rotation is given to the ball, the ball unit is then released at the given height away from the string. The rotation speed of ball is monitored with the encoder attached at the end of the rotation shaft, and the distance between the ball unit and the string is measured with the optical sensor. The angular acceleration and the normal force are calculated after tests.

Commercially available nylon string, of two different cross sections, i.e. round and flat, was selected for the tests. The test conditions are listed in Table 1. Impact tests were conducted at least 3 times for each condition.

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Impact test conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial rotation speed, rpm</td>
<td>800, 1500</td>
</tr>
<tr>
<td>Impact speed, m/s</td>
<td>1.4, 2.8</td>
</tr>
<tr>
<td>Initial tension, N</td>
<td>196</td>
</tr>
</tbody>
</table>

3. Experimental results

The results for the case of initial rotation speed of 1500rpm and impact speed of 2.8m/s are shown here as an example. The normal force and the angular acceleration are calculated from displacement and rotation speed, respectively.

The displacement and the normal force during impact tests are shown in Fig. 1. The maximum displacement is almost the same for round and flat type of nylon string, but the maximum normal force of flat type is ~10% higher. The rotation speed and the angular acceleration during impact tests are shown in Fig.2. The change in rotation speed is almost same. As for the angular acceleration with flat cross section string, the peak value is ~10% lower possibly due to the lower initial rotation speed and the duration of peak region is longer.

4. Conclusion

Comparison between round and flat cross section string indicates that flat string provides higher normal force and longer period of higher friction which might relate to the better spin feeling by players. More discussions will be presented.

Acknowledgements

Author would like to express thanks to GOSEN Co., LTD for providing the samples for the tests.
Experimental and Molecular Simulation Approaches on Boundary Friction of Nano-Carbon Materials as EP Additive

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1. Introduction

ZnDTP and MoDTC have been widely used as an EP additive for conventional automotive engine oils. They usually form a protective solid film inside metal contact under boundary friction, thus reduce friction and severe wear. However, in order to address recent environmental requirements, it is necessary to reduce the amount of sulfur and phosphorus in the present additives because these contents poison exhaust catalysts, and development of the alternative additives is thus required. On the other hand, it has been recognized that use of “nano-carbon”, i.e. fullerene, nano-tube and nano-onion, as EP additive is one of the ways to overcome the described issue. In the present study, the boundary friction of C\textsubscript{60} fullerene as the simplest nano-carbon was investigated by using a conventional friction test and a computational chemistry technique.

2. Methods

Pin-on-flat friction test was firstly performed for C\textsubscript{60} coated SUJ2 steel under ultra-high vacuum (UHV) to obtain the tribochemical insights of the system. \textit{In-situ} X-ray photoelectron spectroscopy (XPS) and Raman spectroscopy analyses were performed for the rubbing surface. Furthermore, to observe the dynamic behavior of C\textsubscript{60} inside metal contact, a molecular dynamics (MD) simulation was carried out using “NEW-RYUDO\textsuperscript{1)*}” code. In this simulation, the model shown in Fig. 1(a) was used. To simulate boundary friction, constant pressure and sliding velocity were applied to the upper Fe substrate. Brenner’s bond order potential\textsuperscript{2)} was employed to express Fe-Fe, Fe-C and C-C interactions. For Fe-C, the required parameters were determined so as to reproduce adsorption energy of C\textsubscript{60} on Fe(001).

3. Results and discussion

From the UHV friction test, the friction coefficient of 0.3 was obtained. \textit{In-situ} XPS analysis was performed for the rubbing surface; un-reacted C\textsubscript{60} still remained on surface but Fe\textsubscript{2}C\textsubscript{60} and sp\textsuperscript{2} carbon contents were also detected. Therefore, this result indicates that C\textsubscript{60} reacts with nascent steel surface to form carbide and another carbon species. To observe the dynamics of C\textsubscript{60} inside contact, the MD simulation results were analyzed. Fig. 1(b) shows the dynamic behavior of one C\textsubscript{60} molecule. It was observed that the molecule was initially deformed along the sliding direction, and subsequently, it was completely crushed. Finally, the C\textsubscript{60} layer was totally changed into an amorphous state (Fig. 1(c)). The formed amorphous carbon (a:C) was strongly adhered to the Fe surface via Fe-C covalent bonds. This would be the origin of high friction state observed in the friction test. The vibrational spectrum was further analyzed by Fourier transforming for the auto-correlation function of velocity of atoms, and total-symmetric vibration mode (490 cm\textsuperscript{-1}) was vanished when a:C was formed (Fig. 2(a)). This indicates disappearance of C\textsubscript{60} spherical structures. The similar change in molecular vibration was also observed for the actual rubbing surface. Hence, both the experiment and simulation results suggest that C\textsubscript{60} forms the protective a:C film by being crushed. More details and the relating material design for further reducing friction will be discussed in the presentation.

4. Acknowledgement

This work was fully supported by Grant-in-Aid for Scientific Research from Japan Society for the Promotion of Science Fellows (Project No. 20-6348).

5. References

1. Introduction

The bulk rheological properties of polymer liquid can be easily measured using a conventional rheometer. In the micro/nano-scale machines, the liquid films (lubricant or adsorbed water molecules) are placed in a narrow gap and sheared at a high rate, where the physical and mechanical properties of the confined liquid films are fundamentally altered from those of the bulk liquid. Particularly, the liquid meniscus bridges formed between micro-scale solid bodies play an important role on the dynamic behavior of the liquid.

Recently, we have developed a nanorheometer [1] to investigate the nanorheological properties of the confined liquid films. We found that the liquid meniscus bridge exhibits excess damping in the low oscillation frequency and high surface separation regimes. We think that the excess damping originates from the movement of the contact line on the solid surface, which is related to the surface energy of solid surface.

The purpose of this study is to investigate the effect of surface energy of solid surface on the dynamic properties of liquid meniscus bridge. First, we report the preliminary results.

2. Experiments

Samples

Two 20-μm-diameter borosilicate glass spheres were used to confine liquid film. Diamond-like carbon (DLC) film was deposited on the glass sphere using ion beam sputter to change the surface energy of glass sphere. The perfluoropolyether (PFPE) lubricant is used as a liquid to confine liquid film. The film was deposited on the glass sphere using ion beam used to confine liquid film. Diamond-like carbon (DLC) film was deposited on the glass sphere using ion beam sputter to change the surface energy of glass sphere. The film was deposited on the glass sphere using ion beam sputter to change the surface energy of glass sphere.

The chemical formula of PFPE lubricant is as follows:

\[ \text{HOCH}_2(\text{CF}_2\text{CF}_2\text{O})_p-(\text{CF}_2\text{O})_q-\text{CH}_2\text{OH}, \quad p/q=2/3 \]

Modeling

The oscillatory motion in the direction of the meniscus bridge axis is applied to the glass sphere mounted on the piezo actuator, and thus the other glass sphere mounted on the cantilever oscillates due to the coupling of the two spheres by the meniscus and the viscous forces. The induced oscillation of the cantilever-sphere assembly is detected via a two-phase lock-in amplifier with respect to the modulation frequency and the surface separation. The amplitude and the phase shift of the cantilever-sphere assembly relative to the motion of the sphere, mounted on the piezo actuator, represent the characteristics of the complex modulus of the PFPE film. The amplitude response is interpreted as elasticity, and the phase-shift response is interpreted as viscosity.

By solving the equation for motion (Eq.1) of the sphere mounted on the cantilever, the storage modulus \( G' \) and the loss modulus \( G'' \) of the PFPE film are obtained as shown in equations (2).

\[
m\ddot{x} + k_x F_x(t) + F_M(D) = 0
\]

where \( m \) is the mass of the cantilever-sphere assembly and \( k_x \) the spring constant of the cantilever. \( F_M(t) \) is the hydrodynamic force of the viscoelastic liquid confined between two spheres. \( F_M(D) \) is the meniscus force as a function of the surface separation \( D \).

\[
G' = \frac{2\pi R^2}{3\pi R^2 \omega^2} \left[ \frac{(k_x - m\omega^2)(\cos \phi - 1)}{(A \sin \phi)^2 + (A \cos \phi - 1)^2} + k_M \right]
\]

\[
G'' = \frac{2\pi R^2}{3\pi R^2 \omega^2} \left[ \frac{(k_x - m\omega^2)A \sin \phi}{(A \sin \phi)^2 + (A \cos \phi - 1)^2} \right]
\]

where \( A \) is the inverse of the amplitude ratio, \( \phi \) is the phase shift between two spheres, \( \omega \) is the radian frequency, and \( k_M \) is the meniscus force constant.

3. Results and discussion

Table 1 shows the water contact angles for each surface. The water contact angle of glass sphere increases due to the DLC coating, i.e., the surface energy of glass sphere decreases due to the DLC coating.

<table>
<thead>
<tr>
<th>Solid surface</th>
<th>Water contact angle [deg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>glass</td>
<td>&lt;5</td>
</tr>
<tr>
<td>DLC film</td>
<td>71</td>
</tr>
</tbody>
</table>

Table 2 shows meniscus force and meniscus force constant of the PFPE meniscus bridge system. The PFPE meniscus force agrees well with the theoretical value of \(-1.34\mu\text{N}\), which is calculated from the \(2\pi\gamma\), where \( \gamma \) is the surface energy of PFPE. The meniscus force decreases due to the reduction of surface energy of the solid surface. On the other hand, the meniscus force constant increases due to the DLC coating.

The solid surface energy is related to the meniscus force and the meniscus force constant, namely, the difference of surface energy of solid surface affects directly the viscoelastic properties of liquid meniscus bridge. The dynamic experiments are in progress.

Reference

On the Movement of Entrapped Oils under Pure Rolling Conditions

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1. Introduction

This paper presents a study on the movement of entrapped oils within an elastohydrodynamic lubricated (EHL) contact under pure rolling conditions. Lundberg1 studied the squeeze and sliding motions of an EHL contact and factors for lubrication failure were deduced. Kaneta et al.2 observed the impact dimple movement within the contact region and tried to extract the non-Newtonian responses of lubricants. They reported that under disc sliding, the speed of the inlet edge of the dimple varies with different lubricants. Ehret et al.3 considered the effect of interfacial slippage by using the release time of the entrapped oil. Their numerical results were in good agreement with Kaneta’s. Recently, Guo et al.4 proposed that the speed of the entrapment within an EHL contact can provide more insights on the boundary slippage than the dimple release time. Thus, the current study was carried out to look into the entrapped oil movement with the EHL contact.

2. Experimental and Numerical

Experiments were carried out with a conventional steel ball/glass disc optical EHL test rig. Initially, a thin layer of oil on the glass disc is squeezed by the steel ball and a dimple is formed. Subsequently, the EHL contact moves with pure rolling. The experiment is illustrated schematically in Fig. 1.

Fig. 2 gives the displacement of the dimple and a reference point of the moving disc. It is clear that there exists a critical displacement value $d_c$ for the dimple. For less than $d_c$, the dimple moves with the entraining speed. However, the dimple slows down with a speed of about 1/3 of the entrainment for its displacement beyond $d_c$.

Numerical analyses were carried out to simulate the impact i.e. the formation of the dimple, and the subsequent lateral moving action. The governing equations include the Reynolds equation, elastic equation, and the pressure-viscosity-density relations. Typical numerical solutions of the movement of dimple are exhibited in Fig. 3. Comparing with the experimental results shown in Fig. 2, the agreement between the two curves is very satisfactory.

3. Conclusion

There exists a critical displacement value when the entrapped lubricant moves. Below this value, the dimple moves with the entrainment speed and the dimple immediately slows down as soon as it beyond this value.

4. Acknowledgement

The work described in this paper was fully supported by the Research Grants Council of the Hong Kong S.A.R., China [Project No. CityU 122408].

5. References

Deposit Formation of Engine Oil at Lower Temperature around 160°C

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1. Introduction
In turbocharged engine, blowby gas is returned to suction air pass of turbocharger due to positive crankcase ventilation system. Blowby gas includes engine oil mist, as engine oil could invade the air pass of compressor. Blowby gas sometimes causes oil coking deposition on the surface of air pass at lower temperature in engine bench test. Suction air temperature after compressor could become 160 ~ 200°C. Although, engine oil cooking at low temperature has not been reported3).

In this paper, the mechanism of engine oil coking deposit formation at low temperature was considered and new experimental method for low temperature engine oil coking deposit formation was developed.

2. Low temperature oil coking deposit
2.1 Analyses for oil coking deposits
Low temperature oil coking deposits was collected from turbocharger used in engine bench test, and the deposit was analyzed. Oil coking deposit made by panel coking test2) at 280°C was also analyzed. The panel coking test deposit is regarded as high temperature oil coking deposit

Fig.1 shows the element content of coking deposits obtained from organic elements analysis (CHNO) and X-ray fluorescence spectrometry (XRF). High temperature oil coking deposit includes Ca, Zn, P and S. These come from engine oil additives.

Fig.2 shows content of oil, sludge, carbon and residual obtained from thermogravimetric analysis (TG). Low temperature deposits contain oil and sludge mainly.

2.2 Low temperature oil coking deposit formation mechanism
In compressed air, partial pressure of oxygen could be high, which accelerates oxidation of oil. In addition, as very thin oil film is formed on the surface of air pass, it would be oxidized easily. Therefore, low temperature oil coking deposit formation is thought to be caused by oxidizing and carbonizing quickly in active oxidation condition.

3. Experimental method for low temperature oil coking deposit
Experimental method for low temperature oil coking deposit formation was developed by applying rotary pressure vessel oxidation test3). In this method, a sample oil of 1g is enclosed in the vessel with oxygen gas. Initial pressure is set at 0.6MPa. The thin film of oil is formed due to the rotation of vessel.

Using this experimental method, oil coking deposit formation was observed at low temperature. It was confirmed that the oil coking deposit made by this experimental method was almost the same as the low temperature oil coking deposit picked from turbocharger by analyses. The experimental deposit was composed of oil and sludge mainly like the picked deposit (Fig.2). This developed method is useful in order to optimize the formulation of oil by increasing their resistance to low temperature deposit formation.

Fig. 1 Element content obtained from CHNO and XRF

Fig. 2 Content of oil, sludge, carbon and residual

4. Conclusion
The low temperature oil coking deposit formation mechanism is different from high temperature oil coking deposit formation. The former deposit is thought to be caused by oxidizing and carbonizing quickly in active oxidation condition. The experimental method for evaluating low temperature oil coking deposit formation was developed in order to optimize the formulation of oil and their resistance to low temperature deposit formation.

5. Reference
2. Fed-TMS -791B
3. ASTM-D2272
Analysis of the Wear and Tribofilm Formation On Silicon Nitride for Total Hip Joint Replacement

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1. Introduction

Total hip joint replacement is one of the most common and successful orthopaedic surgeries. With an older and more active patient population, the number of failures of the total hip implants is increasing [1]. Many of the failures are related to tribology, i.e. wear of the cup, head and liner. Accumulation of wear particles at the implants is linked to osteolysis which leads to bone loss and eventually aseptic implant loosening [2]. Therefore, it is highly desirable to reduce the generation of wear particles from the implant surfaces. The purpose of this work is to examine the friction and wear properties of silicon nitride (Si3N4) as a material in hip joint implants. Si3N4 is chosen due to its ability to slowly dissolve in vivo, which will minimize the risk of osteolysis caused by wear particles.

2. Experimental

Ball-on-disc experiments were performed with balls of silicon nitride sliding against discs of either Si3N4 or cobalt chrome (CoCr). The tests were carried out in SBF-solution (Simulated Body Fluid). The normal force was 1 N, the sliding speed was 0.05 m/s, which correspond to the maximum contact pressure in the hip joint and a normal walking speed. The tests were carried out for about 900 000 revolutions. Before the tests, the discs were polished to a surface roughness (Ra) of about 2.5 nm. The nature of the tribofilm formed on the surfaces and the particle generation was studied using SEM (Scanning Electron Microscopy) and the volume loss was studied using Wyko (optical profiler). The chemical composition of the tribofilm was studied using XPS (X-ray Photoelectron Spectroscopy). The superficial surface hardness of the wear tracks on the discs was measured with an ultra nano hardness tester (UNHT).

3. Results

The SEM studies of the worn Si3N4 disc did not show any clear wear marks, whereas the wear marks on the CoCr disc appeared distinctly (Fig. 1).

The width of the wear track was approximately 100 µm on the Si3N4 and 500 µm on CoCr. The XPS analysis showed that the Si3N4 surfaces on both the ball and discs were covered with a thin Si3O4(OH)2-H2O deposit. Here, x, y and z varied with probe position and a higher z value was measured close to the surface.

The wear and the coefficient of friction were much lower for Si3N4 against itself compared to CoCr (Table 1). Both of the friction tests started with a coefficient of approximately 0.4 which then decreased to a stable level in the course of 600 revolutions for Si3N4 and 15 000 revolutions for CoCr, respectively.

Table 1  Mean value of friction and wear area.

<table>
<thead>
<tr>
<th>Disc material</th>
<th>µ (mean value)</th>
<th>Cross sectional wear area (µm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Si3N4</td>
<td>0.005</td>
<td>3.0</td>
</tr>
<tr>
<td>CoCr</td>
<td>0.33</td>
<td>9800</td>
</tr>
</tbody>
</table>

The original surface hardness for CoCr (9.1±0.5 GPa) was lower than the hardness in the wear track (10.4± 0.2 GPa) but the opposite trend was observed for Si3N4 with a hardness of 21.3±0.6 GPa on the original surface. However, no statistically significant difference was found.

4. Discussion

The combination of Si3N4 against Si3N4 showed both low wear rate and low coefficient of friction, in contrast to CoCr against Si3N4. It is reasonable to assume that it is the Si3O4(OH)2-H2O tribofilm that gives the Si3N4 material its low coefficient of friction. The surface modification and tribofilm build up occur during the running-in period, and then the coefficient of friction and wear rate decrease.

Although this simple test is far from mimicking the actual implant conditions, concerning friction and wear properties, Si3N4 against Si3N4 was shown to be a potential material combination for hip joints.

5. Acknowledgment

The authors are grateful for the financial support from Swedish Foundation for Strategic Research (SSF), Strategic Research Centre on Materials Science for Nanoscale Surface Engineering (MS²E).

6. References

Tribological Behaviour for Low-Friction Coatings at Conformal Contact Pressure in Traditional and New Alternative Fuels

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1. Introduction

The increasing demand for lower emissions together with the impoverishment of the fossil fuels have lead to an intensive research on environmentally friendly fuels and materials. In the strive for improvements, there is a strong driving force towards reduced friction and wear in order to increase the efficiency of the engines and as a pleasant byproduct also lower the emission rates\(^1,2\).

With the large number of vehicles used around the world only a few percent improvement would save enormous amount of energy and considerably decrease the emission of greenhouse gases.

One increasingly popular solution for the tribologically problematic components in the automotive industry is to use a hard amorphous diamond like carbon coating (DLC)\(^3\). These coatings are very resistant to wear and surface fatigue, and they may also decrease the friction compared to uncoated contacts\(^4\).

Previous work by the authors has shown that the coefficient of friction and wear vary substantially, not only between different coatings, but also for each coating between the various fuels. It was found that for some fuels, two of the experimental coatings show a much lower coefficient of friction than conventional DLC coatings. However, these tests were conducted in a non-conformal contact and hence they had a contact pressure which was some order of magnitude higher than the pressure used in most conformal contacts. This yields an unfair situation for some of the softer experimental coatings. The promising results from the non-conformal sliding motivated the present investigation, using a lower contact pressure in reciprocating sliding of low-friction coatings against steel in five different traditional and alternative fuels.

2. Experimental

To achieve a low contact pressure, a new type of specimen holder (see Fig. 1) was designed to fit an already existing test rig for ball-on-flat reciprocal sliding. The ball specimen is replaced by a segment from a 200 mm circular specimen with a width of 2 mm, which also constitutes the line contact width. For a normal load of 5 N, the initial Hertzian contact pressure is 20 MPa, which represents many conformal contact situations. The circle segment is made of ball bearing steel and has a polished surface equivalent to the finish of a ball bearing steel ball.

Five different commercial fuels were used as a test medium: E85, Petrol 95 octane, Diesel EC1, White spirit and Rapeseed Methyl Ester (RME). They were applied continuously to the contact during the test by a syringe pump. The flow rate was adjusted to match the evaporation rate of each specific fuel.

Fig. 1 Cross section image of self aligning circle segment holder. Segment 1 and coated specimen 2. Arrow marks the sliding direction.

Five different coatings were evaluated. Three different leading commercial DLC coatings and two experimental PVD coatings. All of the coatings were deposited on polished (Ra \(\approx\)10nm) steel substrates. As a reference, a non-coated substrate was used.

The coefficient of friction was measured during the test and wear volume and wear scars were evaluated. The test speed was adapted so that the contact would stay in the boundary lubricated regime for the fuel with the highest viscosity, RME.

3. Results

The results show variances in both friction and wear for the different coatings in different fuels. It is clear that some coatings work better than others in specific fuels.

4. Conclusions

- Compared to the steel reference, several candidate coatings perform substantially better both in terms of wear and friction properties.
- For both friction and wear properties, not always coupled, there are candidates that show large potential. However the most promising candidate differs between the various fuels.
- Extremely good wear resistance is common for all the DLC coatings. In many cases the wear on the coating is undetectable.

5. References

Additive Effect on Friction and Wear Performance of DLC Coatings

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1. Introduction

Diamond-like coatings (DLC) are becoming quite widely used as coatings for machine components in lubricated systems such as crankcase engines. One issue of practical concern is the way that these coatings interact with the various additives present in formulated lubricants and especially the impact of additive-DLC interactions of friction and wear. A number of previous studies have investigated the impact of lubricant additives on DLC lubrication. Most of these have focused on steel/DLC contacts but a few have looked also at DLC rubbing against DLC.

This paper describes friction properties of a range of boundary lubricating additives in DLC-DLC sliding-rolling and also compares them to their behavior in a steel on steel contact. Different types of DLC have been considered, ranging from hydrogenated diamond-like to a non-hydrogenated graphitic type.

A systematic study, using the MTM (Mini Traction Machine) friction test instrument, and AFM (Atomic Force Microscopy) as well as ToF-SIMS surface analysis was performed in order to study the performance and further the mechanisms by which conventional lubricant additives act on DLC surfaces.

2. Experimental

A rolling-sliding Mini Traction Machine (MTM), (PCS Instruments, London), was used to form additive reaction films and monitor their friction properties. Prolonged rolling-sliding was carried at a low entrainment speed of 0.1 m/s, a slide-roll ratio of 50%, and a load of 30 N. Periodically, motion was halted and Strubeck friction curves obtained in which friction was measured over a range of entrainment speed. Before resuming low speed rubbing, in situ optical interference technique was used to monitor the tribo-film thickness formed on steel surface of the ball.

At the end of the test, MTM disc wear tracks were further examined using Atomic Force Microscopy (AFM) in order to determine morphology, roughness and lateral force characteristics of additive films formed.

3. Results

Figure 1 shows a series of Strubeck curves measured during prolonged rubbing of a ZDDP/OBCaSu solution on steel compared to Strubeck curves obtained with the same additive solution on a DLC surface. It can be seen, that rubbing on steel caused a dramatic rise of both, boundary as well as mixed lubrication friction, while that on DLC showed little effect on friction in the 2h rubbing on either.

In Figure 2, AFM topography images of 20x20 um² areas obtained in the centre of wear tracks are presented. Both, the topography as well as corresponding line profiles taken across the film showed that while the AW additive system formed a thick and rough film on the steel surface, significant smoother and thinner boundary film was formed on DLC by the same additive solution.

4. Conclusions

In this study, a series of lubricant additives has been studied on different DLCs. Further, friction, film forming and wear properties of the same additives solutions have been compared to their behavior on steel.

It could clearly been shown that the properties and response to additive systems is rather different for each type of the DLC surface and also differs significantly from their behavior on steel.

5. References

Effect of Magnetic Field on Triboplasma Generation

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1. Introduction

Triboplasma is generated by discharging of ambient air in the rear gap of sliding contact. The triboplasma causes beneficial or harmful effects on the surrounding gas, lubricating oil, solid lubricants and the mating sliding partners. For developing a new triboplasma technology, it is necessary to develop technologies to control triboplasma generation. It has been shown that the triboplasma can be controlled by changing the electric resistivity of the mating sliding partners. One of other possible factors to control the triboplasma is application of magnetic field to the triboplasma, since the Lorenz force from the magnetic field should act on the moving charged particles in the plasma (Fig. 1).

In the present paper, effect of the magnetic field on the triboplasma generation has been studied to have knowledge whether the magnetic field can change the intensity and the distribution of the triboplasma.

![Fig. 1 Triboplasma generation under magnetic field](image)

2. Experimental procedure

Figure 2 shows the principle of the experimental apparatus. Two-dimensional image (triboplasma image) of photons emitted from the sliding contact and its vicinity was measured, while a diamond pin with a tip radius of 2 mm, sliding on a sapphire (single crystal Al₂O₃) disk with a thickness of 1 mm at a room temperature of 25 degree C in an ambient air with the relative humidity of 44% under the normal force, F_N = 940 mN and sliding velocity, V = 420 mm/s under magnetic field application. Details of the measurement method are described in the previous report.

![Fig. 2 Principle of the experimental apparatus](image)

3. Results and discussion

Figure 3 shows the two-dimensional image of the total photons emitted from the sliding contact and its vicinity, in the dark under application of the magnetic field of approximately 2500 G. It is seen that by applying the magnetic field, the half of the circular triboplasma disappeared. This disappearance was caused by the action of the Lorenz force due to the magnetic field to the moving electrons and ions in the plasma.

![Fig. 3 Triboplasma image under application of magnetic field](image)

4. Conclusion

The results showed that by applying a magnetic field to the sliding contact, the distribution and intensity of the triboplasma can be changed.

Acknowledgement
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References
Evaluation of Surface Properties of DLC Films by a Micro Slurry Jet Erosion (MSE) Test and Nano-scale Observations of Eroded Surfaces

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1. Introduction

There are several evaluation techniques for the surface strength of diamond-like carbon (DLC) films, but nevertheless they are not yet sufficiently established.

In the present study, in order to evaluate the surface properties of the DLC films, micro slurry jet erosion (MSE) tests were conducted. The erosion mechanisms are discussed using nano-scale observations of eroded surfaces.

2. Experiment

The DLC films were deposited on the substrate of cemented carbide (square of 12 x 12 mm² with 5 mm in height) by unbalanced magnetron sputtering using bias voltages of 0, -50, -100 and -150 V (designated c-0, c-50, c-100, c-150). The thickness of the DLC films and metal interlayer was about 2 µm and 1 µm, respectively. Nano-scale hardness and Young’s modulus of the DLC films, which were measured by a nano-indenteter, increased with the bias voltage and reached the same value for c-100 and c-150, see Table 1.

The details of the MSE tester are described elsewhere1,2. A stream of water containing solid particles, sucked from the tank, is mixed with compressed air in the nozzle, and eventually the slurry-jet is ejected at high velocity into the atmosphere. The cross-section of the nozzle is a square of 3 x 3 mm², which generates a square wear scar on a flat, coated specimen. For the pressure of 0.5 MPa used in this experiment, the maximum velocity was estimated to be around 100 m/s at the exit of the nozzle2. The impingement angle of the slurry-jet was set to 90°. The test piece was mounted at 10 mm distance from the end of the nozzle. The test liquid was pure water containing angular alumina particles with the average diameter of 1.2 µm. The concentration of the erodent was 3 wt %. The geometry of the worn surface was measured with a stylus profilometer along the centerline of the square wear scar. At each test interval the distance between the original and the worn surface was measured at the deepest position and designated as the wear depth.

3. Results and discussion

The wear rates of DLC films are summarized in Table 1. The wear rates are dependent on differences in dimension and frequency of surface ruptures. Sequential observations with an atomic force microscope (AFM) show that the erosion proceeds by crack propagation for soft DLC film (0 V), see Fig.1, and by small impact fracture for hard DLC film.

The transmission electron microscope (TEM) images of the cross-section of the worn surfaces were examined. The mechanical fracture in the size of about µm is found for 0 V, see Fig.2, but only smooth surface is observed for -50 V. The surface analysis also reveals that the material removal during the MSE test occurs on a nano-scale of less than several 10 nm for DLC films.

4. Conclusions

When increasing the bias voltage, the wear rate of the DLC films decreased rapidly from 0 V to -50 V and afterwards decreased gradually. The wear proceeds by crack propagation for a soft DLC film, and by small impact fracture for a hard DLC film.

5. References


Table 1 Surface roughness, nano indentation test results and wear rates for DLC films

<table>
<thead>
<tr>
<th>Surface roughness</th>
<th>Hardness H, GPa</th>
<th>Reduced modulus E, GPa</th>
<th>Wear rate of DLC film, µm/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sa, nm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>c-0</td>
<td>21</td>
<td>4.2±0.3</td>
<td>86±4</td>
</tr>
<tr>
<td>c-50</td>
<td>27</td>
<td>12.6±1.6</td>
<td>250±30</td>
</tr>
<tr>
<td>c-100</td>
<td>40</td>
<td>13.8±0.7</td>
<td>275±10</td>
</tr>
<tr>
<td>c-150</td>
<td>32</td>
<td>14.0±1.2</td>
<td>280±20</td>
</tr>
</tbody>
</table>

Fig.1 AFM images of sequential observation of DLC films (0V)
Fig. 2 TEM image of the cross-section of the worn surface of DLC film (0V)
Characterization and tribological properties of low-friction tribofilms formed on WS₂ coatings in dry sliding-nanoparticle based coatings compared to sputtered coatings

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1. Introduction

Thin coatings of MoS₂ and WS₂ are widely known for their low-friction behavior in un lubricated sliding in non-humid environments, while in humid air their surface properties rapidly break down. It has been reported that during sliding the outermost atomic planes reorient and align along the sliding direction to produce a tribofilm with extremely low shear resistance. Typically, some of these planes become transferred to the countersurface where a similar tribofilm is formed. Inorganic fullerene-like nanoparticles (IF) based on MoS₂ and WS₂ have been incorporated into metallic coatings and have been reported to show excellent performance under certain conditions. In these cases the contact has been covered with a similar tribofilm, formed by exfoliation of the closed structured nanoparticles. It has further been seen for these materials that increasing the normal load and hence the contact pressure decreases the coefficient of friction in sliding contact.

The present study investigates the tribological properties of WS₂ based coatings of both the planar and the IF-structures. The focus is on the tribofilms and their properties such as structure, morphology, hardness and chemical content.

2. Experimental

A coating consisting of inorganic fullerene-like WS₂ nanoparticles in an electrodeposited Ni matrix and a sputtered WS₂ coating were investigated in un lubricated unidirectional sliding contact at a sliding speed of 0.1 m/s in dry nitrogen atmosphere. The experiments were performed in a pin-on-disc setup against 6 mm ball bearing steel balls under normal loads varying from 5 to 50 N.

The shape of the wear tracks was measured using white light interferometry to calculate the wear volumes after different number of passages.

TEM-cross sections were cut out from the wear tracks and the ball surface to study the microstructure and thickness of the tribofilms formed. The chemical composition of the surfaces was investigated by Auger electron spectroscopy and the hardness of the tribofilm was measured using nanoindentation in the wear tracks.

3. Results

Figure 1 shows the friction results at various loads for both the sputtered coating and the nanoparticle coating. Both coatings showed very low steady state friction values, and for both types of coatings the coefficient of friction decreased with increasing normal load. The running-in behavior for the nanoparticle coating was dramatically improved at the higher normal loads.

It took several thousand passages to reach a friction coefficient below 0.02 at 5 N load, but only a few hundred at 50 N load.

TEM cross sections show that the tribofilm is clearly thicker for the higher load case and it appears that the atomic planes are affected to larger depths. AES analysis results indicate that there is a lower oxide content in the higher load case. Nanoindentation results in the wear tracks indicate that the tribofilm becomes much harder at higher loads.

![Fig.1 The coefficient of friction as a function of the applied normal load for both coatings tested in dry nitrogen.](image)

4. Conclusions

Interestingly, both types of coatings give generally similar friction behavior and tribofilm, despite their extremely different microstructures. Both show very low friction after running-in and decreasing µ with increasing load. Friction and wear mechanisms seems to change, the chemical content of the tribofilm seems differently distributed, the hardness and structure of the outermost atomic layers change. Further, the affected depth appears to increase when increasing the load.

5. References

Friction and Wear Dependence of nc-TiNiC: a-C by Environment and Test Conditions

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1. Introduction

PVD nanocomposite TiNiC in a matrix of amorphous carbon has shown promising results as a low friction material and also as a material that could be used in electrical contacts. Earlier testing of these coatings, as well as other carbon based low friction coatings, have shown that test conditions like air humidity, lubrication, counter surface etc. are of great importance. In this investigation the coatings have been subjected to such systematic testing followed by close investigations of the worn surfaces.

2. Experimental

The coatings where tested in a ball-on-disc setup. Flat samples were coated with the nanocomposite coating and ball bearing steel balls or alumina balls, 6 mm in diameter, where used as counter surface. The experimental setup was covered by a hood into which gases where flooded creating the desired environment during the test. By this method, tests were conducted in dry air (less than 5% RH), lab conditions (~30 % RH) and nitrogen. The tests were run until the coefficient of friction reached 0.7 or until 70000 revolutions. Tests were also performed in transformer oil to simulate one example of conditions encountered in electrical contacts. After the test the surfaces where studied using scanning electron microscopy.

3. Results

Some of the test results show that humidity has a big effect on the coefficient of friction. For the dry condition the coefficient of friction stabilizes at 0.35 after running in. This friction level is maintained until approximately 3000 revolutions where the coefficient of friction rises and becomes more unstable. The unstable friction coefficient stays below 0.7 until approximately 15000 revolutions. In nitrogen, the coefficient of friction reaches a steady state level of about 0.25 after the running in. Also in this test, the friction is steady until about 3000 revolutions when it starts to increase. In nitrogen the coefficient of friction stays below 0.7 for about 4500 revolutions. The big difference in behavior is at 30% relative humidity where the coefficient of friction reaches about 0.2 after the running in and then stays on that level for all 70000 revolutions. The results for all conditions are shown in Fig.1.

Scanning electron microscopy of the worn surfaces shows worn through coatings for all tests except for the tests in humid air. In this case we find surfaces that have been worn smooth as by polishing, in the sliding direction of the wear track, which can be seen in Fig. 2.

4. Concluding Remarks

The results show that the environment in which the coatings will be used is very important. Under humid conditions the tested nanocomposite coatings function very well and give low friction and wear. In dry conditions one solution might be to deposit these coatings in a hydrogen rich atmosphere, a method used for some DLC coatings, to create hydrogen containing coatings.

5. Acknowledgement

The authors are grateful for the financial support from Swedish Foundation for Strategic Research (SSF), Strategic Research Center on Materials Science for Nanoscale Surface Engineering (MS²E).

6. References

Effect of Wavy Shape on the Performance of Wavy Mechanical Face Seal Used for Boiler Feedwater Pump

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1. Introduction

On the sealing region of wavy mechanical face seal, hydrodynamic pressure generated by wavy surface profile is used for maintaining thin fluid film. Because the wavy shape placed on rotor surface affects the performance of wavy mechanical face seal, it is necessary to understand the effect of wavy shape parameters (peak-to-peak amplitude of wave, number of 1 cycle wave placed on rotor surface and area of sealing dam region)1). In this study, numerical analysis using finite volume method was conducted to investigate the performance of wavy mechanical face seal as wavy surface profile was varied.

2. Analysis

Figure 1 shows the schematic view of analysis region. For considering the variation of wavy profile placed in analysis region, \( r_{\text{dam}} \), \( A_{\text{wave}} \) and \( n_{\text{wave}} \) were varied appropriately. And Table 1 presents analysis conditions.

To get the pressure distribution and perturbation pressure distribution generated in the analysis region, the Reynolds equation was solved using finite volume method. Then, from the pressure distribution and perturbation pressure distribution, static characteristics (opening force, fluid leakage and friction torque) and dynamic characteristics (stiffness coefficient and damping coefficient of the fluid film) were calculated.

3. Results

Figure 2 shows opening force and fluid leakage when \( n_{\text{wave}} \) is 18. For the same \( n_{\text{wave}} \) and \( h_{\text{min}} \), opening force, fluid leakage and stiffness coefficient of the fluid film increase with the increase in \( A_{\text{wave}} \). And they also increase with the decrease in \( r_{\text{dam}} \). On the other hand, friction torque and damping coefficient of the fluid film decrease with the increase in \( A_{\text{wave}} \). And they also decrease with the decrease in \( r_{\text{dam}} \).

For the same \( r_{\text{dam}} \), \( A_{\text{wave}} \) and \( h_{\text{min}} \), opening force, fluid leakage and stiffness coefficient of the fluid film increase with the increase in \( n_{\text{wave}} \). However damping coefficient of the fluid film decreases with the increase in \( n_{\text{wave}} \). And friction torque is invariable regardless of \( n_{\text{wave}} \).

4. Conclusions

The static and dynamic characteristics of wavy mechanical face seal were affected by the variation of wavy profile placed on rotor surface, and showed consistent trends with variation of \( r_{\text{dam}} \), \( A_{\text{wave}} \) and \( n_{\text{wave}} \) for the same \( h_{\text{min}} \).

5. Acknowledgement

This work was supported by the Brain Korea 21 Project in 2010.

6. Reference


Table 1 Analysis conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tr>
<td>Outer radius of analysis region ( r_{\text{out}} )</td>
<td>89.54 mm</td>
</tr>
<tr>
<td>Inner radius of analysis region ( r_{\text{in}} )</td>
<td>84.01 mm</td>
</tr>
<tr>
<td>Working fluid pressure at ( r_{\text{out}} ) ( p_{\text{out}} )</td>
<td>2.063 MPa</td>
</tr>
<tr>
<td>Working fluid pressure at ( r_{\text{in}} ) ( p_{\text{in}} )</td>
<td>0.101 MPa</td>
</tr>
<tr>
<td>Rotation speed of rotor ( \omega )</td>
<td>609.5 rad/s</td>
</tr>
<tr>
<td>Viscosity of working fluid ( \eta )</td>
<td>0.0006 Pa·s</td>
</tr>
<tr>
<td>Minimum thickness of the fluid film ( h_{\text{min}} )</td>
<td>1–2 ( \mu )m</td>
</tr>
</tbody>
</table>
Experimental Taylor Vortex Flow

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1. Introduction

Experimental Taylor vortex flows were reported by many investigators, e.g. Coles, Benjamin and Mullin, and Anson. However, they either recorded only a frontal view of the Taylor type flow patterns or not clearly presented the cross-section views. Present work uses polymer micro-spheres with an average dimension of 91 μm for flow visualization of the formation, steady-state structure, and collapse of Taylor vortices; they are presented in the longitudinal cross-section of the clearance between the rotating shaft (inner cylinder) and the outer stationary cylinder.

2. Results and Discussion

When the speed is lower than the critical Taylor speed, the flow exhibits a Couette regime type structure, Fig 1a). Note that due to the difference between the centrifugal forces of the oil and that of the particles, only some of the particles are accumulated near the inner cylinder (left side). When the speed approaches the critical speed, one can distinguish the formation of the kernel of two Taylor cells initiated near the inner cylinder, Fig 1b). With the further increase of speed, the cells' boundaries become more and more discernible, while their areas are extending toward the outer cylinder, Fig 1c). Eventually, the cells occupy fully the space between the inner and outer cylinders, Fig 1d), their boundaries becoming clearly defined, signaling that the cells are fully formed, Fig 1e) and near, or at steady-state. At critical speed a very small disturbance would destroy the Taylor cells. Figure 2) presents the process of the collapse of the Taylor vortices with the increase of time. At the beginning of this process the Taylor vortices are well formed and clearly defined, Fig 2a). However, the shear force between the vortices is large enough to cause deformation and destabilize them with the increase of time, Fig 2b) and Fig 2c). The cells' disintegration advances (with time) as the particles migrate towards the vortices central area, Fig 2d). Finally the vortices of Fig. 2d) disappear completely as they are replaced with Pre-wavy flow, Fig 2e).

The collapse process duration depends on the kinematic viscosity of the fluids. The collapse times are in decreasing order 40, 7 seconds, respectively, as the oils kinematic viscosity increases from ν = 7.7 × 10⁻⁶, to 4.0 × 10⁻⁵. This finding is consistent with Snyder’s experimental work showing that the relaxation (collapsing) time is proportional to L²/ν, where L is the length of the vortex column.

3. Conclusions

Couette, Taylor, Pre-wavy and wavy regime flows are visualized and discussed. To the authors’ best knowledge, the Pre-wavy regime was identified visually for the first time, and reported consequence of this investigation.

4. References

Measurements of Adsorption Behavior of Lubricant Additive by Quartz Crystal Microbalance

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1. Introduction

Formation of adsorption layer on the sliding surface plays important role in friction and wear of materials under the boundary lubrication with additives. Ability to observe adsorption behavior of molecules is the key to a range of fundamental and applied issues in tribology and nanotechnology. There are many methods to measure formed surface films, e.g. Atom Force Microscope (AFM) and ellipsometer. However, there is no established method available for monitoring the adsorption behavior of lubricant additives.

In this study, the Quartz Crystal Microbalance (QCM) for the in-situ measurements was used. The QCM is a sensitive mass measuring method based on an inverse piezoelectric effect. Oleic acid is added as an additive at a concentration of 0.1wt% to hexane and hexadecane. Mass change which is derived from adsorption of additive is measured as frequency shift by QCM. In this experiment, two types of sensor with different material surfaces, i.e. gold (Au) and carbon (C), were compared.

In addition, friction coefficient was measured by using a pendulum friction tester. The relationship between the friction properties and the adsorption behavior including the effects of the lubricant temperature was investigated.

2. Experiments and Results

Oleic acid adsorption in hexane was measured by QCM using Au sensor and C sensor. Specifications of QCM sensors are shown in Table 1. Sensor was washed in ultrasonic cleaner. After the cleaning, sensor was mounted in the holder and immersed in hexane. When the frequency signal becomes stable, oleic acid is added at a concentration of 1.0wt% to hexane. After that, frequency shift is monitored throughout the process, and the run ended when the frequency reached a new stable level.

The temperature of lubricant is raised with QCM using oleic acid, the frequency signal begins to change to a new level. Frequency shifts derived from the adsorption of the oleic acid using different material sensor are shown in Fig.1. In this figure, Au sensor’s frequency shift is stable at about 270Hz. This is corresponds to a mass change of 270ng. Forming oleic acid monolayer causes 300ng mass change according to height and density of monolayer. From these results, the measurement using QCM is considered to be reasonable in monitoring the adsorption behavior. C sensor’s frequency shift declines slowly compared to Au sensor. This indicates lower adsorption activity of carbon than that of gold.

Frequency shifts due to the lubricant’s temperature are shown in Fig.2. Only in case of using oleic acid, the frequency shift begins oscillating at about 60°C. This temperature is near the desorption temperature of oleic acid. This oscillation is attributed to desorption of oleic acid.

During the friction test, the change also occurs at about 60°C.

3. Results and Discussion

Frequency shifts due to the adsorption of the oleic acid using different material sensor are shown in Fig.1. This figure, Au sensor’s frequency shift is stable at about 270Hz. This is corresponds to a mass change of 270ng. Forming oleic acid monolayer causes 300ng mass change according to height and density of monolayer. From these results, the measurement using QCM is considered to be reasonable in monitoring the adsorption behavior. C sensor’s frequency shift declines slowly compared to Au sensor. This indicates lower adsorption activity of carbon than that of gold.

Frequency shifts due to the lubricant’s temperature are shown in Fig.2. Only in case of using oleic acid, the frequency shift begins oscillating at about 60°C. This temperature is near the desorption temperature of oleic acid. This oscillation is attributed to desorption of oleic acid. During the friction test, the change also occurs at about 60°C.

4. Conclusions

The QCM analysis showed the differences between C and Au sensor in the adsorption behaviors. Obvious changes were observed both in the adsorption and the friction behavior at a temperature of 60°C, which was a desorption temperature of oleic acid.

5. References

Effects of Heat Treated and Sliding Condition on Structure Changes of DLC Film

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1. \textbf{Introduction}

Diamond-like carbon (DLC) films have unique tribological properties, such as smooth surface, high mechanical hardness, high wear resistance and low friction coefficient. DLC films are used in various industrial applications, for example, in cutting tools, steel molds and in general machinery elements. On the other hand, some practical problems associated with the use of DLC films still remain. Recently, heat resistance, relief of residual stress and improvement of adherence are focused on. Especially, heat resistance must be improved to utilize DLC film for a metal molds which surface temperature can be up to 400°C. Generally, the structure of DLC films changes at high temperatures. Graphitization of the DLC structure causes the hardness deterioration and peeling off of the DLC. In this study, we focus on the effects of heat treatment conditions and sliding on structural changes of DLC films.

2. \textbf{Experimental}

Hydrogenated DLC films were deposited on high-speed tool steel (JIS-SKH51) and high carbon high chromium alloy tool steel (JIS-SKD11) by the plasma based ion implantation (PBII) using hydrocarbon gases. Ball on disk sliding tester were employed to evaluate the friction properties of the each DLC films heat-treated at the temperature of 300°C, 350°C, 400°C. After the sliding test, structural changes of the DLC films were evaluated by the micro laser Raman spectroscopy.

3. \textbf{Results}

The friction behavior of the DLC films on the SKD11 substrate is shown in Fig. 1. Friction coefficients fluctuated between 0.20 and 0.25 after 1000 revs of sliding in each condition. Friction behavior of the DLC films on the SKH51 substrate is shown in Fig. 2. Only the non-heat treated DLC film on the SKH11 showed higher friction coefficient. There are, however, few differences in the friction behavior of the heat treated DLC films both on the SKD11 and SKH51.

Analytical results of the DLC films by the micro laser Raman spectroscopy are shown in Fig. 3. DLC film is generally characterized by the D (Disorder) peak at around 1350cm\(^{-1}\) and the G (Graphite) peak at around 1550cm\(^{-1}\). G peak band gives useful information on the structure of DLC film. Decrease of the G band width means the occurrence of graphitization at high temperature under heat treated conditions. From these results, the graphitization caused by the heat treatment does not significantly affect the friction behavior. The influence of the structure change caused by sliding, however, should be examined in the future.

4. \textbf{Conclusions}

1. Heat treated DLC films showed the friction coefficient between 0.20 and 0.25 after 1000 revs sliding under each condition.
2. There are few differences in the friction behavior of the heat treated DLC films deposited on both the SKD11 and SKH51.

The micro laser Raman analytical results showed the occurrence of the graphitization in the heat treated DLC films. Such a graphitization caused by heat treatment does not significantly affect the friction behavior.
Effect of Laser Surface Texturing on Friction Behavior of Ceramics
Under Lubrication with Water

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1. Introduction
Surface texturing is viewed as one of the surface modification technique to improve the hydrodynamic lubrication. A large volume of work, consisting not only of experiments but also providing theoretical basis for the texture geometry optimization, has been reported in the literature [1-3]. However, the purpose of surface texturing in this work is not to increase the hydrodynamic lift. Surface geometry of ceramics is changed during sliding. A low friction is attributed to smoothing of the surface with the tribo-chemical reaction products [4,5]. The surface texturing is expected to play an important role in the formation of smooth sliding surface. In this paper, an improving effect of the laser surface texturing on the water lubrication of silicon nitride is discussed based on the experimental and the surface analytical results.

2. Experiment
Friction behavior was evaluated using a thrust-cylinder type tribo-tester. The schematic diagram of the disk and the ring specimen, both of which were made of the sintered Si₃N₄, is shown in Fig. 1. Several types of dimple array patterns were textured on the sliding surface of the disk specimens using a Q-switch YAG-Laser. Before friction measurements, a pair of specimens was rubbed for ten minutes under the running-in condition at a load of 11N and a rotational speed of 200 rpm. In order to generate the Stribeck curves, sliding condition was changed continuously every 2 minutes using the same sliding pair as shown in Table 1.

![Fig.1 Schematic of sliding specimens: (a) Disk (b) Ring](image)

Table 1 Sliding conditions
<table>
<thead>
<tr>
<th>Thrust-cylinder type friction tester</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotational speed [rpm]</td>
</tr>
<tr>
<td>(Sliding velocity [m/s])</td>
</tr>
<tr>
<td>Load [N]</td>
</tr>
<tr>
<td>(Contact pressure [MPa])</td>
</tr>
<tr>
<td>Lubricant</td>
</tr>
<tr>
<td>Temperature [°C]</td>
</tr>
</tbody>
</table>

3. Results & Discussion
The shift of the Stribeck curve to the low bearing characteristic number with the laser surface texturing is shown in Fig. 2. The hydrodynamic lubrication region of the textured surface (P100-D31) was observed to shift to the low speed and the high load condition side compared with the non-textured, original, surface. The friction coefficients under the boundary lubrication were also reduced on the laser textured surfaces (P60-D30 and P100-D31). It is considered that tribo-chemical reaction products formed on the sliding surface act as a kind of lubricant and protect the sliding surface from severe wear. Moreover, the formation of silicon hydrate increases the viscosity of water solution near the sliding surface and extends the hydrodynamic lubrication region.

![Fig.2 Change of friction behavior in Stribeck curve](image)

4. Conclusions
The laser texturing showed the friction reducing effect for the water lubricated Si₃N₄ under sliding condition of the low bearing characteristic number. It is considered that the textured dimple pattern affects the formations and facilitates the smooth sliding surface with the tribo-chemical reaction products. It is necessary to pay more attention to promotion and practical use of the tribo-chemical reaction when designing the optimal surface textures.

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Tribological studies on composites with Polyetherimide and γ-irradiated carbon Fabric
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1. Introduction

Carbon fibers, most favored as reinforcement, are also known for poor wettablility with matrix and need pre-treatment. Amongst various surface treatment methods, γ-ray radiation is an effective approach since it adds surface roughening and polar functionality which lead to enhancement in fiber-matrix adhesion. Though this is a well investigated technique,[1,2] effect of various dozes of irradiation on quality of composites and tribological behavior is not reported in depth. In this work, γ-ray treated carbon fabric (CF) was used with Polyetherimide (PEI) matrix to observe the effect of treatment on adhesive wear of composites.

2. Experimental

The CF treatment was done by exposing it to Cobalt 60 γ-ray source for three doses (100, 200 and 300 kGy) at the rate of 4.54 kGy/hr. Composites were developed by impregnation of CF in PEI solution followed by compression molding[3]. Composites were designated as CG0, CG100 and CG200 and CG300 where suffices 0-300 indicate γ-ray dozing for CF. Composites were characterized for physical and mechanical properties as per standard methods and data are shown in Table 1. For adhesive wear studies on a pin-on-disc machine a pin was slid against a mild steel disc at a speed of 1 m/s for sliding distance of 7.536 km under variable loads (200-600N). SEM was done on CF to observe the surface roughening on CF.

3. Results and Discussion

Data in Table 1 show that the fiber–matrix adhesion improved in increase in dozing and as seen in Fig 1, increase in surface roughening due to treatment was a key factor responsible for the same.

![Figure 1: SEM (X10000) of CFG0 and CFG300.](image)

Friction and wear performance of composites is shown in Figs 2 With increase in load, K0 increased for all composites and μ decreased which is as per observed trends[3]. Composites showed K0 and μ in the range of 15-28.5x10^-16 m^3/Nm and 0.1-0.22 which are very low indicating excellent tribo-potential of composites.

Table 1-Properties of developed composites

<table>
<thead>
<tr>
<th>Property</th>
<th>CG0</th>
<th>CG100</th>
<th>CG200</th>
<th>CG300</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fiber vol (%)</td>
<td>55.5</td>
<td>55.2</td>
<td>55.8</td>
<td>56.5</td>
</tr>
<tr>
<td>Void vol (%)</td>
<td>0.48</td>
<td>0.51</td>
<td>0.51</td>
<td>0.51</td>
</tr>
<tr>
<td>Density g/cc</td>
<td>1.54</td>
<td>1.55</td>
<td>1.56</td>
<td>1.56</td>
</tr>
<tr>
<td>ILSS (MPa)</td>
<td>54±1.4</td>
<td>51±1.42</td>
<td>51±1.42</td>
<td>51±1.42</td>
</tr>
</tbody>
</table>

Approximately 20 % decrease in K0 and μ was observed due to treatment. Higher the doze, higher was the improvement. Thus approx. 60% improvement in ILSS due to treatment led to enhancement in friction and wear performance of composites.

4. Conclusions

γ-ray irradiation of CF proved a effective tool to enhance fiber-matrix adhesion as evident from increased ILSS. It also enhanced friction and wear performance of composites. Higher the doze, better was the performance. SEM analysis showed increase in roughness of fiber with increase in irradiation dose which was responsible for more adhesion.

5. References

Boundary and micro - Elasto-Hydrodynamic-Lubrication of Textured Surfaces

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1. Introduction

In engineering, as well as in physics, multi-scale and multi-physics experimental modeling solves complex problems exhibiting important features of multiple spatial and/or temporal scales. The major difficulty involves scales linking. In case of fundamental tribological approach to the interfacial contact problems there is a lack of explicit and emphatic knowledge associated with the transitional phenomena of Boundary and Elasto-Hydrodynamic Lubrication (BL, EHL) regimes. Even a casual survey of the general fluid rheology literature leads to the conclusion that the assumed temperature, pressure and shear rate dependence of lubricant in the studies of Boundary and EHL cannot be supported by the empirical evaluations if the morphological multi-scale approach of surfaces is neglected. Critical analysis of the existing theoretical and experimental knowledge of transition from Boundary and micro-EHL (µEHL) regimes, taking into account multi-dimensional scales, is presented in this work. Few unexpected results obtained during the experimental modeling of µEHL of textured surfaces, taking into consideration morphological multi-scale aspects, are presented and discussed.

2. Experimental

Major problems still exist in our understanding of the transitional interfacial mechanisms from BL to EHL and vice versa. One way to analyze this is using a multiple beam interferometric visualization [1] of the interface described in details in [2]. Investigations of BL regime require the use of a singular tribometer offering different results. This tribometer is not described here due to the page limit, however, it will be discussed during the oral presentation. One of the features of the EHL tribometer is the simultaneous recording of high speed imaging allowing for a real-time evaluation of film thickness between steel ball of 12.7mm radius and flat glass disc under dynamic conditions. Surfaces are textured by femto-second L.A.S.E.R process offering different micro-cavity patterns. Surface morphologies are characterized in the SEM (Scanning Electron Microscope), WLI (White Light Interferometer), AFM (Atomic Force Apparatus), confocal microscopy, OM (Optical Microscope) and surface profilometry in order to fully understand surface texture effects on tribological characteristics [3].

3. Results

As it can be depicted from Fig. 1, the inner micro-cavity morphology affects micro fluidics behaviour at the exit of the contact which is characteristic of µEHL. Investigations of different micro-cavity patterns and their internal morphologies reveal their impact on tribological characteristics of the interface.

![Interferometry image of the EHL contact](image1.jpg)

Fig.1 Interferometry image of the EHL contact between the steel ball (AISI 52100) and flat disc (glass) with individual entrapped micro-cavity at the Hertzian mean contact pressure of 300 MPa, sliding velocity 25 mm/s, mineral oil of dynamic viscosity 0.2 Pas (top), lubricant film thickness (middle), section profile of the cavity (bottom)

4. Conclusions

The influence of inner nano-meter morphology of the textured surfaces has a dominating influence in the EHL regime. The entrapment time of the lubricant at the sheared interface conforms to the well known transitional behaviour.

5. Acknowledgements

The authors wish to express their gratitude to French National Research Agency (ANR) financial support and industrial partners: PSA Peugeot Citroën, Mahle, Lubrizol, Impulsion, ALTIMET for their contributions.

6. References

Prediction of surface parameters and friction stress during sheet metal forming process with the metallic thin film on die material

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1. Introduction

The real area of contact and the surface parameter features actually dominate the friction and the distribution of lubricant at the interfaces between the workpiece and tool. The contact phenomena in metal forming operations are mostly affected by high contact pressures and severe plastic deformations of the workpiece. Due to high contact pressures the asperities are significantly flattened and high values of the real contact area fraction typically occur in metal forming processes. Wanheim et al. [1] and Bay [2] assumed adhesive friction mechanism and a rough workpiece–smooth tool (RW-ST) interaction mode. Wilson and Sheu [3] proposed an upper-bound plane-strain model of surface asperity flattening in metal forming processes.

2. Finite element asperity flattening in sliding contact for simulation

Figure 1 shows the schematic diagram of the surface asperity flattening in sliding contact system with the tool (the metallic thin film on die) and workpiece. The width of each triangular asperity is equal to \( S_m \), which represents the asperity spacing, and the contact area ratio \( A \) is defined as

\[
A = \frac{l}{S_m} \tag{1}
\]

where \( l \) is the asperity contact length of the central peak.

Figures 2 and 3 shows the effects of parameters such as the nondimensional sliding distance \( S \), normal pressure \( P \), strain rate \( e \) and Young’s modulus of film \( E \) on the contact area ratio and surface roughness (Ra is the initial surface roughness and Ra’ is the surface roughness after asperity is deformed.) of the asperity flattening in sliding contact process. The contact area ratio increase and the surface roughness ratio decrease as the nondimensional sliding distance increases. In addition, large value of normal pressure and strain rate results in high value of the contact area ratio and low value of the surface roughness ratio at the same nondimensional sliding distance, but the Young’s modulus is opposite.

3. Surface parameters abductive network prediction model and application

To validate the accuracy of the prediction model, another data sets of the suitable range are tested for the surface parameters (A, Ra’/Ra) of surface asperity flattening process. The surface parameters between the abductive network prediction and FEM simulation under various combinations of process and material parameters are compared. Prediction results for the surface parameters are consistent with FEM simulations quite well. Therefore, the developed networks have a reasonable accuracy for modeling of the surface parameters of surface asperity flattening in sliding contact process. The surface parameters prediction models are then combined with the finite element stretch forming analysis to predict the workpiece surface parameters, friction stress and strain distribution during the sheet metal forming process.

4. Conclusions

This study established a prediction model for predicting surface parameters of asperity flattening in sliding contact for metallic thin film on die material using the FEM combined with an abductive network. Predicted results for surface parameters from the abductive network prediction model are in good agreement with FEM simulation results. The surface parameters prediction models are then combined with the finite element sheet metal forming analysis to predict the workpiece surface parameters, friction stress and strain distribution during the sheet metal forming process.

5. References

Study on Evaluation of Cavitation Erosion Characteristics of Hard Coating Using Ultrasonic Vibration

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1. Introduction

Cavitation erosion is one of the major problems yet to be solved in energy plant system. To improve cavitation erosion resistance, application of a surface modification technique is proposed such as a nitriding treatment and hard coating. Diamond-like carbon (DLC) films exhibit excellent tribological properties such as smooth surface, high hardness, high wear resistance and low friction coefficient. Experimental research on cavitation erosion resistance of metallic materials and hard thin metallic alloy films have been performed. However, cavitation erosion study of DLC films has not been yet carried out. In this work cavitation erosion properties of hard coatings are evaluated using ultrasonic vibration and the results reported.

2. Experimental Details

2.1. Experimental Condition and Apparatus

The vibratory cavitation erosion test apparatus is used in this study. The apparatus generates bubbles by oscillating horn which is connected to the vibrator. The surface of a specimen, which is set just below the horn, is eroded by the collapsing bubbles. At the end of the horn, a stainless steel tip (JIS SUS304) of 16 millimeters in diameter is mounted. The horn frequency is 19kHz and double amplitude (peak to peak) is 50μm.  The edge of the tip connected to the horn is positioned 15mm below the water surface. A distance between the tip and the surface of specimen is 1 mm. Deionized water is used as a test liquid and the temperature is maintained at 25°C ± 2°C.

2.2. Test Specimens

Material of the test specimen is bearing steel (JIS SUJ2) coated with titanium nitride thin film (TiN of thickness 2.4μm), chemical vapor deposition DLC thin film (CVD-DLC of thickness 1.1μm) and physical vapor deposition DLC thin film which contains little hydrogen (H-free DLC of thickness 0.9μm).

2.3 Evaluation of Erosion Wear

To evaluate the degree of cavitation erosion quantitatively, the erosion depth of the specimen surface is measured. The specimen’s surface is observed by laser microscope and the erosion depth is defined as a distance between the level of eroded area and the initial surface (no eroded area). The condition of the surface is examined in the scanning electron microscope (SEM).

3. Result and Discussion

The relationships between the erosion depth and the hardness of each thin film are shown in Fig. 1. Comparing the erosion depth between different DLC thin films, CVD-DLC thin film which has lower hardness is eroded deeper than the H-free DLC thin film, i.e., CVD-DLC film exhibits more damage on its surface. The H-free DLC film contains less hydrogen and more sp3 structure than the CVD-DLC film. It makes H-free DLC film harder than the CVD-DLC films. Since cavitation erosion occurs through various processes, it is difficult to represent the damage by a particular property value. However, it is reported that the higher hardness, strength and fatigue strength a material has, the less damage it tends to exhibit. In this work, the same tendency is observed. But, although CVD-DLC film is harder than the TiN film, the CVD-DLC film exhibits deeper erosion. This seems to be because the top layer of CVD-DLC surface changes into graphite. The hardness of CVD-DLC film before and after the test is shown in Fig.2. It shows that the hardness decreases by 13GPa after the test, compared to the initial hardness of 19GPa.

4. Conclusions

(1) Comparison of the erosion depth between different DLC thin films shows that H-free DLC thin film exhibits better cavitation erosion resistance than CVD-DLC film.

(2) The cavitation erosion test decreases the hardness of the DLC thin film top surface compared to the original hardness.

Fig.1 Effect of specimen hardness on erosion depth

Fig.2 Specimen hardness before and after cavitation erosion test
Grease Leakage from Shielded Rolling Bearings under Some Particular Conditions

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1. Introduction

Usually, grease leakage from shielded rolling bearings is caused by the grease being pushed out by balls or cage motion, centrifugal force, or grease flow due to high temperature. However, under certain specific conditions, we found other mechanisms causing significant leakage emerging. Such conditions, investigated in this study, were (1) using PFPE-grease with a residue of mineral oil-grease, (2) precession motion of shaft and (3) shaft inclination.

2. Bearing conditions investigated

(1) PFPE grease with a mineral-oil-grease residue
This condition could occur, for example, when packing the bearing with PFPE-grease without sufficient removal of previously packed mineral-oil-grease.
(2) Shaft inclination
When imposing a static overhung load on a shaft.
(3) Precession motion of shaft
When imposing a rotating overhung load on a shaft.

3. Grease immersion tests

Simplified grease leakage tests to estimate the leakage mechanisms from shielded bearings without pushing-out by balls or cages were conducted. Figure 1 shows a schematic of the tester. The disk diameter and gap were 50 and 0.1 mm, respectively. In the tester, the circumferential motion of actual bearings was transformed to a lateral motion for easy visualization of the grease movement. The grease leakage flow into the gap between the shaft and shield of the actual bearing was equivalent to the grease immersion into the gap between the disk and glass plate.

To evaluate case (1) above, mineral-oil-grease was lightly pre-coated onto surfaces of the disk and glass plate, and a lump of PFPE-grease deposited on the same (Fig. 1 (a)). For cases (2) and (3), the disk or glass plate was set tilted, respectively (Fig. 1 (b)). The disk rotation speed was 900 rpm for case (1) and 320 rpm for cases (2) and (3). Although these simplifications included some discrepancy in the motion of the actual bearings, the basic driving mechanism for grease leakage was actualized.

4. Experimental results and discussion

Figure 2 indicates the time evolution of the grease immersion depth into the gap for case (1). Only mineral-oil-grease coated tests indicated the grease immersion into the gap. The mechanism causing the immersion of grease into the gap was the rotation of the lump of grease due to torque generated by adhesive force exerted on both the disk and glass plate.

Figure 3 shows the results for cases (2) and (3).

5. Conclusion

Grease leakage could occur without a conventional mechanism such as pushing-out by moving balls or cages, or by centrifugal force etc. under certain particular conditions.

6. References

Clarification of superlow friction mechanism of CNx coating in N₂ with AFM

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1. Introduction

Carbon nitride (CNx) coating is one of the carbonaceous materials, that was reported to show superlow friction coefficient (<0.009) in Nitrogen environment\(^1\). The superlow friction that appeared after a running-in process of several hundreds of friction cycles and its mechanism have not yet been fully explained. Umehara et al. explained that the soft thin layer generated on a hard CNx coating after the running-in process contributes to the superlow friction\(^2\). However, the mechanical properties of the layer have not been experimentally assessed. The reason is that the layer is too thin to be measured by conventional indentation methods. Therefore, we carried out in-depth measurements of mechanical properties by nano-scratching tests using an atomic force microscope (AFM). Also, we used two kinds of CNx specimens for comparison. First specimen was as-deposited CNx coating while the other one was after-running-in CNx coating that shows superlow friction after the running-in process. The after-running-in CNx showed higher wear and was estimated to have lower hardness and shear strength at the top layer. Therefore, the existence of soft thin layer was experimentally investigated in this study.

2. Experimental

The nano-scratching tests were carried out using AFM as shown in Fig. 1. Diamond tip was used to minimize the tip wear. The in-depth progresses of wear were measured by repeating scratching under high load and scanning under low load including friction tests.

3. Results

Figure 2 shows the AFM images after 0, 10, and 30 nano-scratching cycles. After-running-in CNx showed more wear than as-deposited CNx. The estimated hardness of after-running-in CNx was lower than as deposited CNx (Fig.3).

4. Conclusions

The mechanical properties of superficial layer of after-running-in CNx coating that shows superlow friction were experimentally measured. The existences of the thin soft layer on the top of after-running-in CNx film have been confirmed.

5. References

The Effect of UV Irradiation on Tribological Properties of Hydrogenated DLC

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1. Introduction

Surfaces with high wear resistance and low friction coefficient are in demand for several industrial fields, e.g. automobile. The low friction surface was demonstrated using several surface coatings. Diamond-Like Carbon (DLC) coating is one of the materials which show low friction and high wear resistance. Therefore it is applied to magnetic storage device, tools, dies, cams and tappets systems of automobile, pistons, etc. DLC coating has different tribological, mechanical and photo electronic properties depending on whether the coating includes hydrogen or not. The amorphous carbon (a-C) coating showed lower friction coefficient and higher wear resistance compared to the hydrogenated amorphous carbon (a-C:H) when it slid under lubricant1). However, a-C:H coating is more easily synthesized on the three dimensional substrates than a-C coating. Thus, a-C:H coating with low friction coefficient and higher wear resistance compared to the hydrogenated amorphous carbon (a-C:H) when it slid under lubricant1). However, a-C:H coating is more easily synthesized on the three dimensional substrates than a-C coating. Thus, a-C:H coating with low friction coefficient under Poly alpha olefin (PAO) lubricant is desirable. For the low friction mechanism of a-C coating under PAO it was assumed that the topmost layer changed to graphite-like layer during the running-in process. Since the process is complex we proposed that ultraviolet (UV) light could be used instead2, 3.

In this study, 3 different wavelengths UV (i.e. 254, 312, and 365 nm) were used and they had 469, 382, and 327 kJ/mol respectively. Friction tests were carried out using a-C:H subjected to the UV light irradiation, and frictional properties were compared.

2. Experiments

DLC coatings were deposited by PECVD to obtain a 1.8 μm thickness coating on Si(100) substrate. The coating had 16% hydrogen. The specimens were UV irradiated for 30, 60, 120 and 240 minutes, and friction tests were carried out using a ball-on-disk type frictional tester. The load was 1 N and the sliding speed was ~0.031 m/s. The ball made of SUJ2 had a 8 mm diameter. PAO lubricant was used.

3. Results and Discussion

The friction coefficient versus number of sliding cycles for an un-irradiated a-C coating is shown in Fig. 1. The friction coefficient had an initial value of about 0.1. The value decreased to 0.06 and then remained at about 0.059 on average from 25000 to 30000 cycles. The friction coefficients of a-C:H coating are shown in Fig. 2. For the un-irradiated a-C:H coating the friction coefficient was ~0.1. The 254 and 312 nm UV irradiated a-C:H coatings had friction coefficients of ~0.056 and ~0.047, respectively. Compared to 365 nm UV the coefficients were lower. Results obtained indicated that UV irradiation could change the topmost layer of a-C:H coating in such way that its frictional properties are comparable to those of graphite-like layer.

4. Conclusions

The un-irradiated a-C:H coating had a higher friction coefficient than the a-C coating. If UV irradiation is used the a-C:H coatings had lower friction coefficients than the a-C and un-irradiated a-C:H coatings.

5. References

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Observation of Water Behavior in the Contact Area of Porous Rubber

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1. Introduction

Recently, porous rubber is often used as a matrix rubber for winter tire. It has been generally assumed that the pores raise the coefficient of friction due to the absorption and removal of water in the contact area. However, there are a few studies of porous rubber about the water behavior in the vicinity of the contact area during sliding. The purpose of this study is to determine the effect of pores on the absorption of water when the porous rubber is slid under water lubrication.

2. Experimental

Model tire samples having holes from 0.5 mm to 5 mm in diameter are rubbed against a glass prism. The water behavior in the contact region around the pore is observed by optical methods. The inside of the pore is also investigated by X-rays tomography at the synchrotron radiation facility, SPring-8.

A rubber wheel specimen was rubbed against a mating glass prism. For the friction experiment, a rubber wheel with 60 mm in diameter and 12.5 mm in width was slid at the sliding speed from 1 to 20 mm/s and at an applied load of 14.7 N, respectively. The X-ray tomography was conducted using a rubber specimen with 90 mm in diameter and 5 mm in width at the sliding speed of 1 mm/s and at an applied load of 0.98 N, respectively.

3. Results and Discussion

Variations of coefficient of friction with the pore diameter under water lubrication are shown in Fig. 1. It seems that the coefficient of friction of the rubber with holes tended to be higher compared with that of without holes. Figure 2 represents the contact region around the leading and trailing area. The shade corresponds to the contact area between the prism and the rubber surface or the water, consequently the light area represents the prism surface contacting with nothing except the air.

As the rubber rotated, bubbles come out from pores (Fig. 3 (a)). This is due to the pressure rise inside the pore because of the rubber compressed by the normal load. The pore seems to suck water at the trailing area since the light area in the pore varied shaded area. The reduction of the stress led the decompression on the inner pressure of the pore. To affirm the water absorption, X-ray tomography was applied. The results are shown in Fig. 3 and Fig. 4. The density difference of the shaded area represents the distinction of material. Judging from unchanged shade of gray in the pore, it is found that water was not sucked inside the pore with respect to the leading area of the rotating rubber wheel. Slight density dissimilarity is equivalent to the boundary between an air bubble and water. After the pore crossing the trailing edge, the air bubble inflated and the one end of the bubble border kept contacting with the prism surface. The pressured air inside the pore dilated because the stress by the normal load was released when the pore moved out from the contact area. As a result, it was found that the pores did not absorb water during sliding though the aperture inside the contact area looked like absorbed water at the trailing edge.
A Screening Method for Assessment of Micropitting Resistance of Industrial Gear Oils


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1. Introduction

Micropitting in gears has been receiving increased attention as a serious mode of failure in many large industrial drives in cement/steel plants as well in gear boxes of the wind turbine industry with increased turbine outputs. It is a significant damage mode in such applications that can impact gear tooth accuracy, leading to increased noise and vibration and reduced gear life. The onset of micropitting is dependent upon the gear metallurgy, design as well as the lubricant in use.

There are few methods available to assess these factors using simulated or actual gear contacts. This paper discusses a short duration laboratory screening method using simulated gear contacts that is able to assess the micropitting resistance properties of gear lubricants.

2. Experimental

This study was conducted using a three disc machine. Three ‘counterface’ rings of equal diameter are separated by a smaller diameter roller located in the middle and in contact with all the rings. This arrangement allows the test roller to be subjected to a large number of rolling contact cycles in a short period of time, e.g. at a test entrainment speed of 3.05 m/s used, the central test roller experiences 1x10⁵ cycles per hour.

Fig 1: Test machine configuration

Speeds of the rings and the roller are controlled separately, therefore allowing any combination of slide-roll ratio and entrainment speeds. The data of the specimens, materials and test conditions is given in Table 1

Table 1 Specimens, Materials & Test Conditions

<table>
<thead>
<tr>
<th>Hertz pressure</th>
<th>1.71 GPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Slide roll ratio</td>
<td>0.052 (roller slower)</td>
</tr>
<tr>
<td>Lubricant supply temperature</td>
<td>70 °C</td>
</tr>
<tr>
<td>Test specimen material</td>
<td>Carburized 16MnCr5</td>
</tr>
<tr>
<td>Case depth</td>
<td>1.0 mm</td>
</tr>
<tr>
<td>Roller hardness (Vickers)</td>
<td>650-710 kgf mm⁻²</td>
</tr>
<tr>
<td>Counter disc hardness (Vickers)</td>
<td>750 kgf mm⁻² min⁻¹</td>
</tr>
<tr>
<td>Roller roughness (CLA)</td>
<td>0.1-0.15 μm</td>
</tr>
<tr>
<td>Counter disc roughness</td>
<td>0.53 ± 0.02 μm</td>
</tr>
</tbody>
</table>

A dip lubrication system using about 150 ml of oil is used to supply lubricant into the contacts. This study was conducted on three Industrial Gear Oils of identical ISO VG 220 viscosity simulating field test conditions.

3. Results

The times for onset of Micropitting are given in Table 2. The test was run in stages of 4 hours after which the central roller was inspected for micropitting under an optical microscope.

Table 2: Times for Onset of Micropitting

<table>
<thead>
<tr>
<th>Sr No</th>
<th>Oil</th>
<th>Onset of Micropitting, hrs</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>A</td>
<td>4.0</td>
</tr>
<tr>
<td>2</td>
<td>B</td>
<td>16.0</td>
</tr>
<tr>
<td>3</td>
<td>C</td>
<td>Not observed</td>
</tr>
</tbody>
</table>

Fig 2(A): Micropitting Oil A(4 hrs) / 2(B) Oil B(20 Hrs)
Fig 3(A): Micropitting Oil B(60 hrs) / 3(B) Pitting oil C (80 hours).

Oils A & B have poor micropitting resistance and micropitting occurs on the central roller, as shown in Figs 2(A), 2(B) and 3(A). Oil C prevents micropitting up to 80 hours of testing. After 80 hours pitting occurs as shown in Fig 3B.

4. Conclusions

The micropitting test can be used to successfully screen the oils for their micropitting resistance in a short period of time. Oils can be classified on the basis of the micropitting resistance in the service application.

5. References

Friction and Wear of PTFE and PTFE Composites in High Pressure Hydrogen Gas

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1. Introduction

Fuel cell vehicles (FCV) are using gaseous hydrogen as their energy source and are expected as a future low emission vehicle for reducing greenhouse gas emissions. To extend the cruising distance of FCVs, hydrogen gas should be compressed to extremely high pressure at hydrogen fueling stations and squeezed into fuel tanks of FCVs. Consequently, some tribological machine elements in FCVs and hydrogen stations must work with high pressure hydrogen gas. However, there is a paucity of available data on the friction and wear behavior of materials in pressurized gaseous hydrogen.

In this study, a tribometer sealed in a high pressure vessel was developed. Then friction and wear behavior of polytetrafluoroethylene (PTFE) and PTFE based composites were characterized in hydrogen under the surrounding gas pressure of 40 MPa, since they are representative polymeric materials used as seals in high pressure gas compressors and regulator valves [1].

2. Materials and methods

The tribometer developed in this study has pin-on-disk configuration and is designed to evaluate friction and wear of materials in high pressure gas environments. A cylindrical polymer pin specimen with a diameter of 6 mm and a length of 15 mm is subjected to an axial load by dead weights, while a metal disk specimen with a diameter of 58 mm is rotated at a constant speed. Both specimens and the friction measurement system were sealed in a high pressure vessel which can hold gases at a pressure up to 40 MPa and maintain constant gas temperatures between 223 K and 373 K.

By using a developed tribometer, tribological characteristics of unfilled PTFE (VALFLON Rod, Nippon Valqua Industries, Ltd., Japan) and 15% graphite filled PTFE (Filled VALFLON Rod, Nippon Valqua Industries, Ltd., Japan) were examined in hydrogen gas pressurized to 40 MPa. 316L austenitic stainless steel disk (SUS316L, Sanyo Special Steel CO., LTD., Japan) was used as a sliding counterface. All disk surfaces were polished to have a Ra value of 0.05 μm. The sliding test was also conducted in pressurized helium under the same test conditions.

3. Results and discussion

Results of friction measurements for unfilled PTFE and graphite filled PTFE are shown in Figs.1 and 2, respectively. The frictional behavior of unfilled PTFE in high pressure hydrogen was similar to that in high pressure helium. On the other hand, the friction coefficient between graphite filled PTFE and stainless steel in 40 MPa hydrogen gas became lower compared with the friction in 40 MPa helium gas. The chemical composition of the polymer transfer film formed on the disk surface was analyzed by using X-ray photoelectron spectrometer (XPS) after sliding tests. Results indicated that a part of surface oxide layer of stainless steel was reduced and the amount of transferred graphite increased in high pressure hydrogen gas. These characteristic effects of the high pressure hydrogen environment might be related to the difference in the tribological behavior.

This study was conducted as a part of Fundamental Research Project on Advanced Hydrogen Science administrated by New Energy and Industrial Technology Development Organization (NEDO).

4. References

Effects of Contact Pressure and Lubricant Composition on Wear Rate and Wear Particle Morphology of UHMWPE

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1. Introduction

Ultra-high molecular weight polyethylene (UHMWPE) components of joint prostheses release very fine wear particles during articulation. These fine wear particles can cause serious adverse reactions to surrounding living tissues and finally induce aseptic loosening of joint components [1]. Therefore, the wear behavior of implanted UHMWPE and characteristics of released wear particles are now recognized as decisive factors for the durability and long term survivorship of artificial joints. However, the in vivo wear mechanism of UHMWPE and the formation process of fine polyethylene wear particles have not yet been fully understood.

In this study, the effects of the contact pressure between UHMWPE and a metal counterface on the volumetric wear rate of UHMWPE and morphological characteristics of released polyethylene wear particles were examined experimentally to gain an understanding of in vivo wear mechanism of UHMWPE. The multidirectional sliding pin-on-plate wear tester was employed to reproduce the “cross-shear” sliding condition of the hip joint articulation [2] in a laboratory wear test. Effects of protein and lipid molecules supplemented to the test lubricant were also evaluated.

2. Materials and Methods

The wear behavior of UHMWPE was evaluated in the custom-build multidirectional sliding pin-on-plate wear tester [3,4]. Cylindrical polyethylene pin specimens with a diameter of 6 mm were machined from GUR415 bar stock and medical grade cast Co-Cr-Mo alloy (ASTM-F75) was used as plate specimens. Phosphate buffered saline (PBS) solutions containing protein and lipid molecules were prepared and used as test lubricants in the wear test. Bovine serum γ-globulin was chosen as a protein constituent of test lubricant to represent the protein molecules contained in the joint fluid. On the other hand, synthetic phospholipid (L-α-dipalmitoyl phosphatidylcholine (DPPC)) was used as an alternative of natural lipid constituents.

All tests were conducted at room temperature. The sliding speed was 20 mm/s or 50 mm/s, and the mean contact pressure was varied from 1.0 MPa to 5.8 MPa. Each wear test was run for the sliding distance of 10 km and the specific wear rate of the UHMWPE pin specimen was determined from the weight changes measured every 5 km sliding. After the last 5 km sliding, test lubricants were collected into a centrifugal tube and processed according to ISO17853 to isolate polyethylene wear particles. Wear particles collected onto polycarbonate filters were observed by SEM. Captured SEM images were subjected to image processing using image analysis software (Scion Image, Scion Corporation) to evaluate morphological characteristics, such as a representative size and a shape index.

3. Results and Discussions

Measured specific wear rates of UHMWPE in three test lubricants with 2 wt% γ-globulin and different lipid concentrations were plotted against the contact pressure as shown in Fig.1. The specific wear rate of UHMWPE clearly depended on the contact pressure. It decreased from 1.8 x 10^{-6} mm^3/Nm, which is equivalent to the clinically reported value [5], to 5.2 x 10^{-7} mm^3/Nm in the lubricant containing 0.02 wt% DPPC with increasing the contact pressure from 1.0 MPa to 3.4 MPa. At the same time, the size of polyethylene wear particles increased significantly. The polyethylene wear rate and morphological characteristics of wear particles were also very sensitive to the sliding speed and the composition of test lubricants.

4. References

Challenges in Developing User Friendly Neat Cutting Oils


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Introduction

Development of user friendly lubricants has become essential need for today’s business environment. Till recent times, neat cutting oils were formulated using conventional additive chemistries such as chlorinated paraffin, sulphur, fatty oils, synthetic ester, phosphors additives etc. in API Group (Gp) I base oils. These conventional additives have provided excellent Extreme Pressure (EP) properties for cutting oil applications. However, ill effects of chlorinated paraffin and active sulphur on humans & environment forced chemists to search for new safe additive chemistries. In addition to the above, gradual shift in base oil production pattern from API Group I to API Group II base oils posed further challenge of storage stability and thermo-oxidative stability for finished products. Therefore, developing neat cutting oils with alternate synergistic additives chemistries in Gp II base oils and matching tribological performance replacing conventional additives has become a challenging task.

This paper deals with the study on some of the challenges faced by the authors in developing user friendly neat cutting oils.

Experimental

Evaluation of different additive chemistries was carried out in API Gp I and Gp II base oils. Blend stability with different additives and their combinations in both types of base oils was determined using cyclic thermal stability test. In this test, oil was kept at 50°C for 8 hours and at 0°C for 16 hours in each cycle. Physical appearance/additive separation in the blends was monitored after completion of each cycle. Oxidation characteristics of some of the blends with Gp I and Gp II base oils were determined by IP 48 method. Neat cutting oil blends avoiding chlorine additive and utilizing additives such as thiophosphate ester, dithiocarbamate, synthetic ester and sulphurised olefin etc. were studied in Gp II base oils for extreme pressure (EP) properties using a 4 ball weld load test after evaluating the storage stability characteristics.

Results

Cyclic thermal stability tests clearly differentiated the solubility of polar additives in API Gp I and Gp II base oils. Some of the additives like sulphurised fat and their combinations with other additives have shown poor solubility characteristics in Gp II base oils. The poor solubility of additives in Gp II oils is attributed to removal of natural polar constituents during refining and presence of higher paraffin content in the base oil. Four ball weld load results were comparable for both Gp I and Gp II type base oils of even different viscosities. This confirms the fact that EP properties are independent of base oil type and its viscomatrix.

Need for antioxidants in Gp II oil containing blends was observed in IP 48 oxidation test. IP 48 test indicated better volatility characteristics of Gp II oils vs Gp I oils, leading to lower VOCs emissions during actual use.

The neat cutting oil blends prepared with four alternate additive chemistries (EF-1 to EF-4) have shown good EP characteristics Fig. 1. The EP characteristics for light duty cutting applications can be easily achieved without chlorinated paraffin or active sulphur additives. However, to achieve EP properties with more than 250 Kg weld load, the available options are limited and only one of the selected chemistry was able to give higher weld loads of 700 Kg. The evaluation studies indicate that EP characteristics of additive blended oils do not change by increase the additive concentration beyond 5% wt. However, possibility of getting higher EP characteristics by synergistic combinations and other additive chemistries needs to be explored.

Conclusions

Shifting from API Gp I to API Gp II base oils for more user and environmental friendly neat cutting oils and quest for searching viable alternate additives to replace chlorinated paraffin and active sulphur has posed many challenges. The studies conducted by the authors have clearly indicated that issues like thermo oxidative properties of group II base oils need to be addressed adequately by optimizing suitable dosage of antioxidants. Studies with non-chlorine EP additives such as thiophosphate ester, dithiocarbamate, synthetic ester and sulphurised olefin were completed and the results indicated value upto 700 Kg (in four ball weld load test), which can be further enhanced by suitable top-up treatments of additives.

References

Experimental Study of Rubber Traction with Concrete Model Surfaces
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1. Introduction

Rubber materials are widely used in tribological applications such as automotive tires and paper feeding units in printers and ticket dispensers where higher rolling traction has to be maintained. One of the factors that affect rubber traction is microgeometry of the surfaces in contact with rubber.

The authors made experimental studies of traction in rolling/sliding contact of rubber [1-3]. The experiments with model surfaces with ceramic balls [1] showed that there was no clear effect of ball radius and intervals on traction coefficient. The coefficient decreased significantly when water was present at the contact, which was not the case with the actual pavement [2]. More recently, the authors used concrete as a material for model surfaces, and found that model surfaces made with parallel cylinders gave higher traction with smaller cylinder radius and with larger intervals between cylinders [3]. In this work, traction at various slip ratios was measured with model surfaces consisting of arrays of semi-spheres made of concrete.

2. Experimental

The test rig used was a compact tester called Roughness and Traction Measuring System [1]. This can give rotation and linear motion of a cylindrical specimen against a ground surface independently with two motors. The ground surface can be either actual pavement, or any model surfaces with artificial roughness. Load is applied with weights. While the specimen is making rolling/sliding contact with the surface, the tangential force is determined with a strain gauge attached to a vertical arm which holds the specimen. Test specimens had a diameter of 62 mm and a width of 12.7 mm, and were made of SBR rubber filled with carbon black.

Concrete model surfaces had semi-spheres aligned at the same height at regular intervals as shown in Fig. 1. The radius of the asperities ranged from 2 mm to 4 mm. The components of concrete were high-early-strength Portland cement 1, Toyoura silica sand 2 and water 0.55 in weight ratio.

Experiments were conducted in both dry and wet conditions where a layer of distilled water, 1 mm thick, placed above the surface.

3. Results and discussion

Figure 2 shows the traction coefficient for the asperity radius of 6 mm and the lateral interval \( v \) of 2.5 mm. It can be seen from the figure that the traction coefficient is greater with larger asperity interval in the direction of motion for entire slip ratio regions. It was also found that smaller asperity radius gave higher traction coefficient. These results agree well with those obtained using the cylinder arrays [3], and suggest that the traction coefficient depends more on contact pressure than on contact area. This implies that the traction is governed more by hysteresis term than by the adhesive term under the present contact conditions.

Another important finding is that there is substantially less difference between the traction coefficients in dry conditions and that in wet conditions with the concrete model surfaces. This agrees with the case of the actual pavement surfaces, but is different from the results for the ceramic balls. This may be because the surface roughness on the surfaces of the concrete semi-spheres prevented, in the wet conditions, full elastohydrodynamic film to be formed. This implies both the advantage and the disadvantage of using concrete as a model material in investigating roughness effects in the traction of actual pavement.

4. References

3. C. Bell, \textit{et al.}, to be submitted.
A Local Investigation of the Thermophysical Behaviour of TEHL Contacts

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1. Introduction

This work aims to study the local behaviour of ThermoElastoHydrodynamic Lubricated (TEHL) contacts. A recently developed Full-System Finite Element TEHL solver\textsuperscript{1} is used to examine the effect of material properties, operating conditions, pressure and temperature dependence of lubricant’s rheological and thermophysical properties (density, thermal conductivity, heat capacity, etc.) on the operation of EHD contacts. Both the global (film thickness, friction, etc.) and local (shear stress and temperature distributions, heat partition, etc.) scales are investigated. For this purpose a typical lubricant (mineral base oil) is considered and its thermophysical and rheological properties’ dependence on pressure, temperature and shear stress are evaluated. The resulting models are included in the TEHL solver which is used to run an intensive series of numerical tests.

2. Results

In a recent work\textsuperscript{2}, the authors showed the importance of accounting for the variations with pressure and temperature of the thermophysical properties of the lubricant (thermal conductivity $K$, heat capacity $C$, etc.) for an accurate estimation of the global behavior of EHL contacts (film thickness, traction coefficients, etc.). Fig. 1 shows traction curves as a function of the slide-to-roll ratio (SRR) obtained for a steel-steel point contact. Both experimental and numerical results are reported. The numerical results include isothermal and thermal cases. For the thermal results two cases are considered. In the first case, thermal properties are taken to be constant at their ambient pressure and temperature values, whereas in the second case, their variations with pressure and temperature are taken into account.

Results clearly show that even when thermal effects are considered, neglecting the dependence of thermophysical properties on pressure and temperature leads to underestimated friction coefficients. This is because heat capacity and thermal conductivity normally increase with pressure and temperature leading to an enhanced evacuation of heat from the contact area. This leads to lower temperatures in the contact area, and subsequently to higher viscosity values. This explains the higher friction coefficients that are observed when thermophysical properties are allowed to vary.

3. Discussion

It is suggested that the observed decrease in film thickness when thermophysical properties are allowed to vary stems from a relocalization of heat inside the contact. Although temperature decreases in the central area of the contact, it is believed that part of the evacuated energy heads towards the inlet of the contact. This leads to a temperature increase in this area, leading to the observed film thickness decrease.

4. References

2. Habchi et al., Tribology International, Article in Press.
The Role of Water in Aqueous Based Lubrication with Glycoproteins.

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1. Introduction

While investigating the adsorption and tribological behaviour of synovial fluid proteins, we found that the glycoproteins showed interesting properties in terms of the reduction of friction\textsuperscript{1}. The two glycoproteins studied, alpha-1 acid glycoprotein (AGP) and alpha-1 antitrypsin (A1AT) contain complex-type glycan chains that are relatively well characterised. Whereas each AGP contains five mainly tetraantennary and triantennary glycan residues, the three glycans in A1AT are mainly biantennary. These glycoproteins give much lower friction coefficients when adsorbed onto hydrophobic surfaces than albumin, the most abundant protein in synovial fluid. It is unlikely that the glycoproteins play an important role in the lubrication of artificial joints as the concentrations are very low, however, the low friction achieved makes these interesting systems to study as alternatives to the more common brush-like polymers presently used for aqueous based lubrication\textsuperscript{2,3}.

2. Mass of adsorbed glycoprotein

The mass of the adsorbed glycoprotein was determined using Optical waveguide lighmode spectroscopy (OWLS). These measurements allow the area occupied by each molecule to be estimated. When A1AT was adsorbed from phosphate buffered saline (PBS) onto a hydrophobic surface, this value was 59 nm\textsuperscript{2} per molecule and on the hydrophilic surface it was 105 nm\textsuperscript{2} per molecule. This indicates that the A1AT packs more closely onto the hydrophobic surface. This was attributed to the strong hydrophobic interactions between the peptide and the surface. As the area occupied by the native confirmation of the biantennary glycans on A1AT was estimated to be similar to that measured by OWLS, it was believed that the glycans determined how closely the glycoproteins were packed together. In order to determine whether this was the case, the A1AT was adsorbed onto the surface from a solution of 1.5M NaCl in PBS. At this salt concentration, the interactions between the glycan chains and the peptides become sufficiently weak to allow the chains to extend into the aqueous solution. If the glycan residues had been limiting the packing of the glycoprotein then there would have been an increase in the adsorbed mass. However, no significant difference could be detected, indicating that the size of the peptide limits how closely the chains can be packed.

The area on the hydrophobic surface occupied by each AGP was estimated to be 58 nm\textsuperscript{2}. AGP has a larger amount of glycan than A1AT but a smaller peptide. As with the A1AT, the native structure of the glycan residues on AGP was estimated to occupy all of the space available. Adsorption from a high salt concentration led to a small increase in the adsorbed mass. Partial deglycosylation was achieved using enzymes that remove the glycans. This leads to molecules with, on average, a higher proportion of peptide than in the original glycoprotein. Adsorption of the partially deglycosylated AGP led to an increase in the adsorbed mass of peptide, however, the area available for each glycan was unchanged. These results indicate that in the case of AGP, it is the size of the glycan residue that limits the adsorption onto the hydrophobic surface.

3. Nanoscale friction measurements

Figure 1 shows the results of AFM friction measurements for PE sliding against alumina in solutions of albumin, AGP and A1AT. Only addition of albumin to PBS resulted in an increase in friction (not shown), AGP and A1AT both caused a decrease in friction. This decrease was attributed to the ability of the glycan structures to trap water at the surface.

Fig.1. AFM friction measurements on PBS and solutions of (a) AGP, and (b) A1AT.

4. References

Effects of Oxygen Gas and Water Vapor on the Friction and Wear Properties of CNx Coatings in Inert Gas Environments

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1. Introduction

The super-low friction of the carbon nitride (CNx) coatings in inert gas environments (i.e., N₂, Ar and He) is adversely affected by the oxygen gas and water vapor. However, the low friction of CNx coatings under inert gas environments with high oxygen concentration and/or high relative humidity is necessary in order to extend the application of CNx coatings to the real environment (e.g., ambient air). In this study, a new pre-sliding technique was introduced to investigate the possibility of achieving low friction of CNx coatings in inert gas environments with high relative humidity and/or high oxygen concentration.

2. Experimental

The CNx coatings were prepared on Si₃N₄ balls and/or Si₃N₄ disks using ion beam assisted deposition system with thickness of ~400 nm. The sliding friction tests were performed in a ball-on-disk tribometer equipped with an environmental controlled chamber. A 2-step friction test, where the worn Si₃N₄ ball in step-1 (i.e., pre-sliding) was replaced by a new CNx coated Si₃N₄ ball in step-2, was introduced to achieve the contact material combination of CNx/CNx.

3. Results

Representative change of friction coefficients of Si₃N₄ ball and CNx coated Si₃N₄ ball sliding against CNx coated Si₃N₄ disk (Si₃N₄/CNx and CNx/CNx) in Ar gas with increasing oxygen concentration is shown in Fig.1. The effects of oxygen concentration and relative humidity in inert gas environments on the friction coefficients of Si₃N₄/CNx and CNx/CNx are shown in Fig. 2 (a) and (b), respectively. In inert gas environments, both Si₃N₄/CNx and CNx/CNx exhibited low friction (i.e., less than 0.05), and sometimes the CNx/CNx showed a lower friction than Si₃N₄/CNx. However, the change of contact material combination had great influence on the friction and wear behaviors of CNx coatings in inert gas environments with increasing oxygen concentration and relative humidity. The friction of Si₃N₄/CNx increased rapidly with the increasing oxygen concentration and relative humidity, and reached high values of 0.10 and 0.18 at oxygen concentration of 30 vol. % and relative humidity of 40 %RH, respectively. However, the friction of CNx/CNx increased very slightly with the increasing oxygen concentration and relative humidity. When CNx coating remained on the ball surface after friction test, the CNx/CNx maintained low friction less than 0.05 even in high oxygen concentration (26 vol. %) or high relative humidity (37 %RH).

4. Conclusions

By introducing a new pre-sliding technique, low friction coefficient (i.e., less than 0.05) was observed in CNx/CNx in inert gas environments with high oxygen concentration (26 vol. %) or high relative humidity (37 %RH).

5. References

Rheology And Surface Effect In Friction In EHL Regime

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1. Introduction

The subject of elastohydrodynamic lubrication arose from studies of the physical mechanism of gear lubrication before extension to ball and roller bearings and cams and followers. It is now commonly accepted that film thickness in such a lubricated contact depends mainly upon the conditions at the entrance of the contact where pressure is low and hydrodynamic conditions are preponderant\textsuperscript{1}. On the other hand, friction induced by the introduction of sliding through the contact results from the rheological behaviour of the interface in the Hertz’ contact zone where pressure and shear rates are high. Although there exist in the literature a number of regression from numerical simulations for calculating the film thickness\textsuperscript{2}, there appear to be no equivalent equations to predict either friction or the rheological properties of the confined interface. In addition, the lubricant constituents play a major role in friction despite the only small number of studies. The objective of this study is to address the relative influence of the additives and of the base oil on the frictional behaviour of EHL films.

2. Experimental

Experiments have been carried out on the IRIS tribometre whose specificity is the simultaneous measurement of friction and the visualization of the contact in controlled kinematics conditions. The friction behaviour of various lubricants has been investigated for an EHL contact and a sliding over rolling ratio (SRR) ranging from 0 to 80\% at a constant mean entrainment speed of 0.05, 0.2 and 0.5 m/s under a mean contact pressure of 300 MPa at room temperature.

The lubricants tested in this study are composed of three base oils with the same combination of additives. The first base oil consists of PolyAlphaOlefin and hydrocracked base oil; the second one is composed of a mix of ester and alkylated naphthalene and the third one is a mineral base oil. The same additives have been studied in these three base oils.

The main characteristics of the lubricants are listed in Table 1. The viscosity has been determined using a capillary viscometer at room temperature for the various lubricants.

<table>
<thead>
<tr>
<th>Base oil (Reference of the formulated lub)</th>
<th>Viscosity (mPa.s)</th>
<th>Pressure coefficient of viscosity (GPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PAO/HC (BF)</td>
<td>48</td>
<td>15</td>
</tr>
<tr>
<td>Ester (BF)</td>
<td>82</td>
<td>15</td>
</tr>
<tr>
<td>Mineral (BF)</td>
<td>93</td>
<td>20</td>
</tr>
</tbody>
</table>

3. Results and discussion

First, the film formation capability is determined: although the base oil and the polymer, via their viscosity, govern the film thickness in EHL regime, the additives control the existence of a boundary film at low entrainment speed. The relative role of the additives and of the base oil on the frictional behaviour is studied for various experimental conditions: this clearly demonstrates that the piezo-viscous response of the lubricant controls the Newtonian/non-Newtonian transition and the friction level. The role of thermal effects is discussed. A strong influence of the entrainment speed, over the frictional response is unexpectedly observed, even in full film regime, as shown in Fig.1. This demonstrates that surface phenomena contribute to the friction response even if there is no solid contact. The contact of thickness, h, is depicted as a multi-layer interface with adsorbed surface layers of thickness h\textsubscript{s} and a bulk layer. A rheological model based on Ree-Eyring theory supposing a heterogeneous flow due to adsorbed surface layers is proposed\textsuperscript{1} to discuss the organization of the molecules within the contact and its effect on the frictional response of the contact.

![Figure 1](image)

\textsuperscript{a) Influence of the contact kinematics on the frictional behaviour of BF. The modified Ree-Eyring model is also plotted in continuous line.

4. References

Surface Texturing for Super-low Friction of Silicon-based Ceramics in Water

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1. Introduction

Water lubrication realizes oil-free lubrication, which decreases the environmental pollution. Mating silicon-based ceramics in water gives lower friction (μ<0.01) due to tribo-chemical reaction. Smooth surface and boundary layer of hydrated silane are formed during running-in period. And its superior property is enhanced by introducing surface texture (1).

The object of this paper is to propose a guideline for design of the surface texture from viewpoint of running-in, which realizes super low friction in water.

2. Experimental procedure

Friction tests were performed with SiC ring/SiC disk sliding pair. An active running-in process was conducted before friction test by step loading, and “running-in load Wr” was defined as maximum step load. Micrometer-sized spiral texture and nanometer-sized groove texture were introduced by femto-second laser on SiC disk surface as shown in Fig. 1.

3. Experimental results and discussion

Both the spiral and groove textures supply water to the contact surface and also make SiC surface more hydrophilic as shown in Fig. 2. It is expected that these effects accelerate tribo-chemical reaction during running-in and form conformable surfaces.

Figure 3 shows friction properties of SiC/SiC and SiC/Textured SiC after well-running-in. It is clear that surface texture as shown in Fig. 1 gives much lower friction coefficient of 0.0005 and higher critical load of 1100N. Textures on SiC disk surface gradually worn out and make smooth surface during running-in. As a result, friction properties are improved by increasing running-in load even after removal of initial surface texture (Fig. 4).

It is clearly show from the results that the surface texture works from not only geometrical point but also running-in (tribo-chemical) viewpoint. The surface texture for controlling running-in is one of important guidelines for design of the surface texture that realizes super low friction in water.

4. Conclusion

Stable friction coefficient of SiC/SiC in water was reduced 80% and critical load was increased four times by introduction of micrometer-sized spiral texture and nanometer-sized groove texture.

References

1. Introduction

The friction and wear mechanisms of thermoplastic elastomers (TPEs), such as TPOs (olefinic thermoplastic elastomers: the mixture of in-situ cross linking of EPDM rubber and polypropylene) and TPEEs (thermoplastic ester elastomers: block copolymers of alternating hard and soft segments connected by ester and ether linkages) were examined. The TPEs are recyclable thermoplastic elastomers, having mechanical properties comparable to rubber vulcanizates. In this study, the friction and wear properties of TPEs were investigated when rubbed against an abrasive cloth and metal gauze.

2. Experimental

A pin-on-cylinder type tribometer was used. A TPE pin (6 mm x 6 mm) was rubbed against #AA-240 abrasive cloth or #250 metal gauze, wrapped around a 25 mm diameter shaft. The experiments were carried out under a contact pressure of 0.136 MPa, temperatures 23 ℃ to 120 ℃, and sliding speed of 0.12 m/s. The TPE specimens are shown in Table 1.

<table>
<thead>
<tr>
<th>Specimen No.</th>
<th>TPE</th>
<th>Shore hardness</th>
<th>Tensile strength at break, MPa</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>TPO</td>
<td>73A</td>
<td>8.6</td>
<td>101-73</td>
</tr>
<tr>
<td>A2</td>
<td>TPO</td>
<td>80A</td>
<td>11.3</td>
<td>101-80</td>
</tr>
<tr>
<td>A3</td>
<td>TPO</td>
<td>73A</td>
<td>8.6</td>
<td>201-73</td>
</tr>
<tr>
<td>A4</td>
<td>TPO</td>
<td>80A</td>
<td>11.3</td>
<td>201-80</td>
</tr>
<tr>
<td>A5</td>
<td>TPO</td>
<td>73A</td>
<td>7.0</td>
<td>111-73</td>
</tr>
<tr>
<td>A6</td>
<td>TPO</td>
<td>80A</td>
<td>8.6</td>
<td>111-80</td>
</tr>
<tr>
<td>A7</td>
<td>TPO</td>
<td>73A</td>
<td>7.0</td>
<td>211-73</td>
</tr>
<tr>
<td>A8</td>
<td>TPO</td>
<td>80A</td>
<td>8.6</td>
<td>211-80</td>
</tr>
<tr>
<td>A9</td>
<td>TPO</td>
<td>70A</td>
<td>7.6</td>
<td>8201-70</td>
</tr>
<tr>
<td>A10</td>
<td>TPO</td>
<td>80A</td>
<td>9.8</td>
<td>8201-80</td>
</tr>
<tr>
<td>B1</td>
<td>TPEE</td>
<td>45D</td>
<td>21</td>
<td>EM460</td>
</tr>
<tr>
<td>B2</td>
<td>TPEE</td>
<td>55D</td>
<td>32</td>
<td>EM550</td>
</tr>
</tbody>
</table>

3. Results

The coefficient of friction and specific wear rate were obtained under steady state rubbing. The results of abrasive and fatigue wear are shown in Figs.1 and 2, respectively. The coefficients of friction for the TPEs decreased with increasing temperature in both the abrasive and fatigue wear experiments. The abrasive wear rates for the TPEs were relatively high at 23 ℃ and abrasion patterns were observed on the worn surfaces. After the minimum wear rates at 40 ℃, the rates increased in the range from 40 ℃ to 120 ℃ (Fig.1). Then, the adhesive wear was partially observed. As shown in Fig.2, fatigue wear rates of TPEs decreased with increasing temperature. The lowest wear rates were observed above 80 ℃ for the TPEs (Fig.2(b)).

4. Conclusions

From the wear experiments of thermoplastic elastomers, the rubber-like wear behaviors are observed at lower temperature. At higher temperature, the wear of TPEs is similar to that of plastics.
Non-Edible Natural Esters – An Environment Friendly & Economically Viable Alternative for Mineral Base Stocks

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1. Introduction

Natural esters are environment friendly alternative to mineral oils. The esters are finding way into more applications as eco-friendly lubricant bases with superior characteristics such as very good lubricity and excellent biodegradability (1) even though they have inferior stability and low temperature fluidity. (2) Since the edible varieties of oils are scarcely available for industrial applications in India, there is a need to look for other viable non-edible and economical alternatives (3).

Three non-edible vegetable oils (VO), i.e. Jatropha, Dilo, and Castor were studied for tribological and physicochemical properties and compared against mineral oils, synthetic esters and some other edible oils. These studies showed promising results in terms of tribological and physico-chemical characteristics for lubricant bases and revealed that the incorporation of additives improves stability of these oils.

2. Experimental

Physico-chemical properties and chemical characteristics of samples of the three oils were determined for applying them as lubricant base. The tribological testing of the oil samples was carried out using a reciprocating ball-on-disc tribometer. Thermooxidative stability was determined by Rotatory Pressure Vessel Oxidation Test and hydrolytic stability by Beverage Bottle Test. Different sets of antioxidants were blended in the oils (Blend 1-4) and the improvement of thermooxidative stability was studied using modified and unmodified Rotatory Pressure Vessel Oxidation Test.

3. Results

Compared with mineral oils, the oils studied possessed desirable characteristics to behave as lubricant base oils in terms of viscosity, viscosity index, flash point and pour point. In comparison to mineral oil, pour point of Jatropha oil and Dilo oil was found to be slightly higher. The highly desirable low pour point of castor oil is comparable with the synthetic esters. Iodine values and fatty acid distribution results were indicative of better oxidation stability compared to other VO.

In RPVOT studies (Fig.1) it was found that the three candidate non-edible vegetable oils showed better thermooxidative stability in comparison to the synthetic esters and soybean oil. Castor oil showed the maximum induction time followed by Jatropha and Dilo oils. Soybean oil performed poorly in comparison to all the oils. The induction time values for all vegetable oils were found to be much lower than mineral oils due to the presence of single and poly-unsaturation in VO.

4. Conclusion

Three oil samples studies were found to have fairly good scope for application as lubricants base oils based on their inherent biodegradability and response towards antioxidant doping. Castor oil has shown exceptional inherent thermo oxidative stability. The incorporation of right antioxidant in these three readily available non-edible vegetable oils, which are economical in Indian scenario, improved the thermooxidative stability to a significant level.

5. References

Evaluation of the ASTM D97 Method for the Determination of the Pour Point of Vegetable Oil Based Lubricants

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1. Introduction

Vegetable oils are increasingly being used for lubricant applications as they are environmentally more preferable. ASTM D97 method is conventionally used to determine the pour point of lubricants. On prolonged storage vegetable oils solidify at higher temperatures than that is obtained by using ASTM D97 method^1. Differential scanning calorimetry (DSC) tracks enthalpy changes for a system undergoing physical and chemical changes during heating or cooling.

At high cooling rate, large proportion of the crystals formed at low temperature is the unstable (α) polymeric form. Formation of this unstable material is almost absent at slow cooling rates (0.1 °C min⁻¹), when only the stable polymeric form (β) was observed^2. In heating experiments (DSC), the initially frozen sample, which is in α polymeric form recrystalizes into more stable β or β' form before melting^2.

2. Experimental

Pour point values of six locally available vegetable oils (Table 1) were tested as per ASTM D97 method. The samples are held in the same bath for 15 minutes after each test and the congelation temperatures were noted down.

DSC experiments on cooling and heating were conducted on coconut oil (a representative saturated oil with very high congelation temperature) at different rates for comparative study of the thermal behavior in cooling and heating. DSC heating experiments were conducted on all the vegetable oils (Table 1) and the exothermic peak values were determined.

3. Results and Discussion

Pour point (PP) values as determined from ASTM D97 method and the congelation temperatures (PP*) of the different vegetable oils tested are shown in Table 1. The rate dependence of peak occurrence temperatures of coconut oil is shown in Fig 1 and Fig 2. Cooling experiments (Fig 1) show that at the cooling rate of 0.1 °C/min the peak value is near to the congelation temperature (PP*). Kinker^3 evaluated the cooling rate in ASTM D97 method as approximately 0.6 °C/minute. Hence ASTM D97 method does not give realistic pourpoint values of vegetable oils. Fig. 2 shows that peak occurrence temperatures in heating are rate independent. The peak occurrence temperatures in DSC heating (PV) are listed in Table 1.

4. Conclusion

The pour point values of Vegetable oils determined by ASTM D97 method are lower than the actual congelation temperature. DSC (heating) is an effective tool to evaluate the cold flow properties of vegetable oils.

Table 1 Pour point (PP), congelation temperature (PP*) and DSC peak values (PV) of vegetable oils

<table>
<thead>
<tr>
<th>Sl. No</th>
<th>Oil</th>
<th>PP °C</th>
<th>PP* °C</th>
<th>PV °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Castor oil</td>
<td>-33</td>
<td>-27</td>
<td>-26.2±2.2</td>
</tr>
<tr>
<td>2</td>
<td>Coconut oil</td>
<td>21</td>
<td>24</td>
<td>25.3±0.7</td>
</tr>
<tr>
<td>3</td>
<td>Groundnut oil</td>
<td>3</td>
<td>5</td>
<td>*</td>
</tr>
<tr>
<td>4</td>
<td>Mustard oil</td>
<td>-18</td>
<td>-15</td>
<td>-14.5±0.9</td>
</tr>
<tr>
<td>5</td>
<td>Olive oil</td>
<td>-9</td>
<td>-5</td>
<td>-3.8±0.6</td>
</tr>
<tr>
<td>6</td>
<td>Sunflower oil</td>
<td>-18</td>
<td>-16</td>
<td>17.2±1.3</td>
</tr>
</tbody>
</table>

* Multiple peaks

Fig. 1 DSC thermograms in cooling

Fig.2 DSC thermograms in heating

5. References

Impact of Combustion Products on the Wear of Piston Rings and Cylinder Liners

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1. Introduction

Model tribometer tests provide an easy and relatively cost-effective way to investigate the wear properties of different materials or lubricants. For this study tests were run to investigate the impact of different ethanol-combustion products on the piston ring and cylinder liner tribosystem. To that end, the lubricant used in tribometer tests was aged with ethanol combustion products as contaminants. Piston ring wear was measured continuously using radioactive isotopes. After the wear testing, X-ray Photoelectron Spectroscopy (XPS) was used to analyse the worn surfaces of the cylinder liner counter body to ascertain the influence of the cylinder liner tribolayer on the wear behaviour of the piston ring.

2. Experimental

Tests were run on an oscillating model tribometer (SRV) with a piston ring – cylinder liner setup. The lubrication was provided by engine oil, which was aged artificially. The oil was contaminated with various ethanol combustion products during a controlled aging process. In this paper “aging 1” refers to the addition of acetaldehyde during aging and “aging 2” to the addition of acetic acid. Aging with air and without additional contaminants is referred to as “aging (reference)”. Piston ring wear was measured using radioactive isotopes. A thin surface layer located in the wear zone contains these radioactive isotopes, which are consequently included in the wear particles. These are transported to a radiation detector by the oil circuit and measured continuously during the experiments. The activity measured during the experiment is used to calculate the wear depth of the piston ring.

In addition to the wear testing described above, the tribolayer on the cylinder liners used in the model tribometer tests was analysed using XPS.

3. Results

Wear testing revealed different wear behaviour for the two types of fresh engine oils and differences between fresh and aged lubricants. All aged oils resulted in higher running-in wear, but lower constant wear rates than in tests run with fresh engine oil. Fig. 1 shows typical wear curves for these tests.

XPS investigations of the cylinder liner surfaces used during the tests show that the composition of the tribolayer is affected by the oil condition. The tribolayer has a higher concentration of carbon for the samples lubricated with fresh engine oil. The tribolayer composition also differs for samples lubricated with oil aged with different ethanol combustion products. The aged lubricant produced a higher concentration of elements from the additives than the base oil. There are also significant differences in the tribolayers of the two aging processes. The acetaldehyde-aged lubricant resulted in more oxidized compounds in the tribolayer.

4. Conclusions

Some of the changes engine oil undergoes during the aging process have a beneficial effect on the long term wear behaviour of the piston ring during model tribometer tests. The addition of ethanol combustion products to the oil circuit in a real engine test results in a similar phenomenon: the constant wear rate is lower. A possible interpretation of the wear behaviour is the formation of a protective tribolayer during the running-in period, which can be seen in the wear curves of the aged oils (Fig. 1). In experiments using fresh engine oil as lubricant, the running-in is less pronounced and the constant wear rate higher.

The XPS investigation found indications that the fresh lubricant shows a higher influence of the base oil, resulting in a tribolayer with a high carbon concentration. Samples lubricated with aged oil result in more additive-related substances in the tribolayer, leading to the conclusion that the additives in the engine oil are more effective after a certain amount of aging time.

5. References

Tribological Behaviour of Zinc and Iron Bulk Polyphosphate Glasses

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\textsuperscript{2)}Dipartimento di Chimica Inorganica e Analitica, Università degli Studi di Cagliari, S.S. 554 Bivio per Sestu, Monserrato, 09042, Italy
\textsuperscript{*}Corresponding author: spencer@mat.ethz.ch

1. Introduction

Zinc dialkyldithiophosphates (ZDDPs) are used as lubricating oil additives. ZDDPs react at the rubbing surfaces forming reaction layers of glassy zinc polyphosphates, which are able to reduce wear. Also the antiwear performance of many other phosphorus-containing additives closely correlates with the formation of phosphates (iron polyphosphates in absence of zinc) in the wear tracks. A systematic study of bulk polyphosphate glasses and their tribochemical properties might provide a further insight into the wear mechanism, which is still poorly understood.

2. Experimental

Zinc, iron and mixed zinc/iron phosphates with different chain lengths were synthesized starting from stoichiometric mixtures of zinc and iron oxide in the presence of $\text{NH}_4\text{H}_2\text{PO}_4$. Tribological tests were performed by rubbing 100Cr6 (52100) steel balls against the glassy polyphosphates in a poly-$\alpha$-olefin (PAO) bath. The surfaces of the tribopairs were investigated after the tribological tests by small-area and imaging x-ray photoelectron spectroscopy (XPS).

3. Results

Compared to a steel-vs-iron polyphosphate tribopair, the coefficients of friction and wear were lower in general for zinc phosphates. A trend towards lower wear and friction was observed when changing from long- to short-chain-length glasses (see Fig.1).

A depolymerization of the glass occurred as a consequence of the reaction of zinc polyphosphates with iron oxide in the presence of high pressure and shear stress. The depolymerization is more pronounced for long-chain polyphosphates, while only small changes are detected in the case of zinc pyrophosphate. All zinc polyphosphates were able to form a glassy transfer film on the contact area of the ball. No differences could be observed between the transfer films formed by zinc polyphosphate of different chain lengths. Moreover, the chemistry of the ball seems not to influence the formation of the transfer film since it is also generated on inert quartz balls. In the case of iron phosphates the glass transferred from the disc during the test was found to accumulate towards the border rather than adhering to the contact area. This difference could explain the higher wear coefficient of the ball for the iron polyphosphates.

4. Conclusion

These results suggest that the formation of an adhesive glassy transfer film on the ball surface is crucial for having effective antiwear action. More work is needed in order to further elucidate the mechanism.
Effects of high pressure hydrogen gas on surface properties of steels

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1. Introduction

In the future hydrogen energy systems, hydrogen may be carried and stored in the form of high pressure gas, and bearings and seals have to operate in high pressure hydrogen. The authors have conducted extensive tests in hydrogen at ambient pressure, and found that friction and wear are often more affected by small amount of oxygen and water than hydrogen itself1,2). At high pressure, however, the situation is different; high pressure enhances permeation of gas into solid and also reduction of oxides on solid surfaces3). This work is aimed at understanding changes in steel surfaces in high pressure hydrogen. Several methods are used to analyze steel surfaces.

2. Experimental

A high pressure chamber was used to expose specimens to hydrogen at pressure and temperature up to 40 MPa and 373 K. The exposure was conducted for hours after repeating purging with dry nitrogen and evacuating with vacuum pump three times. The exposure conditions are shown in Table 1.

Hardened bearing steel AISI52100, martensitic stainless steel 440C, austenitic stainless steel 316, 316L were tested. Machined 14 mm diameter and 4 mm height disk were used as specimens. All specimens were polished with diamond slurries just before the exposure test.

After the tests, hydrogen contents in the steels were measured by TDS, thermal desorption mass spectroscopy. In order to see the changes in surface properties of the steels, chemical analyses with XPS, X-ray photoelectron spectroscopy, AES, Auger electron spectroscopy and micro hardness measurements were made.

3. Results

It was found that the amount of hydrogen that permeated into the steels by the exposure to high pressure hydrogen increased with the test temperature. There was significant increase in hydrogen in stainless steels after being exposed to 40 MPa hydrogen at 373 K but negligible increase after exposure at 303 K.

Depth profiling for steel surfaces by XPS suggested that the exposure to 40 MPa hydrogen at 373 K reduced the thickness of the oxide layer, as shown in Fig.1. In addition, the steel surfaces exposed at 373 K exhibited higher concentration of carbon than that at 303 K. Results by AES mapping also suggested that the exposed martensitic steel surfaces were dotted with carbon layers. In contrast, the exposure to high pressure hydrogen at 303 K promoted oxidation of steel surfaces. These results also suggested that the relationship between the changes in surface layers and the pressure hydrogen, however, the situation is different; high pressure enhances permeation of gas into solid and also reduction of oxides on solid surfaces3).

4. Conclusions

High pressure hydrogen caused the changes in surface properties of the steels that were not observed under normal pressure. These changes are pronounced at higher temperature. It was also suggested that the changes in surface properties affected the content of hydrogen in the steels.

5. Acknowledgements

This study was conducted as a part of “Fundamental Research Project on Advanced Hydrogen Science” through administration by NEDO.

6. References

1. Tanaka, H. et al., Tribology Online, 4, 4, 2009, p. 82.
1. Introduction
Wear is surface damage that involves progressive material loss due to relative motion between the surface and the contacting substance or substances. Based on the interaction between surface and erodent, material can be removed through cutting and/or deformation. For any particle-surface combination, energy required to remove unit mass of material is called unit energy factor for material removal.

In this paper, results of single particle impact tests are presented and compared with standard erosion test results for a better understanding of strain rate dependent material response.

2. Experimental
Single particle impact tests were conducted using a high pressure gas gun at the University of Newcastle. Using specially designed projectile, small particles (50-200 μm) are propelled up to speed of 200 m/s and impact on polished sample of mild steel. Figure 1 shows the schematic of the Gas Gun.

![Gas Gun Schematic](image)

**Fig. 1:** High pressure Gas Gun used for the single particle impact test.

Individual impact craters were analysed using laser scanning confocal microscope (LSCM) for crater depth and volume for different impact velocities. Crater volumes were used to determine energy factors for deformation and cutting according to the following equation:

\[ W = \frac{1}{2} MV^2 \cos^2 \alpha + \frac{1}{2} M(V \sin \alpha - K)^2 \]  
\[
\text{for } \alpha > \alpha_0
\]

where \( W \) is the erosion value, \( M \) is the mass of particles striking at angle \( \alpha \) with velocity \( V \), \( K \) is the threshold velocity for material removal, \( \alpha_0 \) is the angle for maximum wear, \( \phi \) is the cutting energy factor and \( \epsilon \) is deformation energy factor.

The crater depths were calculated based on the material properties, particle diameter and impact velocity according to the following relationship:

\[ \delta = \left( \frac{D^2}{6} \right)^{\frac{3}{2}} \frac{\rho V^2}{Y} \]

Where \( Y = 3Y \), \( Y \) is the uni-axial flow stress of metal, \( D \) is the particle diameter, \( \rho \) is the density of the surface material and \( V \) is the particle impact velocity.

3. Results and discussions
Cutting and deformation energy factors for different surface and particle combinations for modelling of wear have been determined using a micro sandblaster. In the current study the energy factors are determined using single particle impact tests for comparison. It was found that the energy factors obtained in single particle impact tests are almost a degree of magnitude higher than those obtained in erosion tests.

Figure 2 shows the measured and calculated depth of crater for different particle velocities. Calculated depth of crater consistently overestimated the measured depths and difference between the calculated and measured depths increased with increasing particle velocity. This showed that the particle is being less efficient with increasing velocity due to the energy loss in strain hardening or increased temperature.

![Crater Depth Comparison](image)

**Fig. 2** Comparison between measured and calculated depth of crater with particle impact velocity.

4. Conclusions
The cutting and deformation energy factors as well as depth of craters were calculated from single particle impact tests and compared with the values determined from the standard erosion tests. The difference between the values is explained in the context of elastic-plastic deformation together with the rate-of-strain effects, work hardening and thermal softening of the surface material.

5. References
Investigation of Hydration and Frictional Properties of Polymer Brushes in Aqueous – Based Mixed Solvents

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1. Introduction

Hydrated poly(L-lysine)-based copolymer brushes have been shown to lubricate non-conformal contacts in an aqueous environment. This is challenging with pure water due to its low pressure-viscosity coefficient. When such copolymers are adsorbed on the surface and the average distance between neighboring water-soluble polymer chains is less than the radius of gyration, $R_g$, in free solution, the polymer chains form a brush-like structure which improves the lubrication properties of the surface.

The quality of the solvent surrounding the brushes can alter the structural properties of the polymer brushes and can also affect their lubrication properties. Previous studies on the interactions of polymer brushes in binary solvents have shown preferential solvation of the polymer by one solution component. Here, we have varied the solvent quality by means of aqueous-based mixed solvents for the poly(L-lysine)-g-poly(ethylene glycol) (PLL-g-PEG) and poly(L-lysine)-g-dextran (PLL-g-dex) copolymers, and studied the influence of these mixed solvents on frictional properties.

2. Experimental

The quality of the solvent surrounding the polymer brush was systematically varied using mixed solvents of aqueous buffer-glycerol or aqueous buffer-ethylene glycol (EG) solutions. The buffer used was a solution of 4-(2-hydroxyethyl)piperazine-1-ethanesulfonic acid (HEPES). A quartz crystal microbalance (QCM) was used to measure the hydration and structural properties of the polymer brushes when adsorbed on silica-coated quartz crystals. Changes in the frequency observed for the brush-coated crystal, when the surrounding solvent is exchanged, are correlated with the degree of hydration of the brushes. The frictional properties of the brushes are investigated under these mixed solvents using colloidal-particle lateral force microscopy, in which a 20 μm diameter silica sphere is sheared against polymer brushes adsorbed onto oxidized silicon wafers. These results are also compared with the macro-tribological properties of the brushes, which are measured using a mini traction machine (MTM).

3. Results

Water-soluble poly(ethylene glycol) is insoluble in glycerol and ethylene glycol, whereas dextran is soluble, since it can interact with these solvents via hydrogen bonding. The enthalpies of solution for dextran with these solvents are comparable with that for dextran with water. Upon adsorption of PLL-g-PEG and PLL-g-dex onto a silica-coated quartz crystal from aqueous buffer, an increase in frequency was observed when the solvent was exchanged from buffer to buffer-glycerol or buffer-EG mixtures. The increase in frequency observed with PLL-g-PEG in these aqueous mixtures was much higher than for PLL-g-dex, in accordance with their solubility behavior as observed in bulk. This increase in frequency can be associated with a loss of hydrated mass and a collapse in the polymer film.

The structural transitions observed during hydration studies were correlated with the frictional properties of the polymers. By means of lateral force microscopy we observed that PLL-g-PEG and PLL-g-dex copolymers, in spite of the observed difference in hydration behavior, show very similar friction response when sheared. Further macro-tribological experiments have been conducted to see the effect of polar solvents on the underlying brush structure.

4. Outlook

Dimethyl sulfoxide (DMSO) is a better solvent for dextran than water. We have observed an increased solvation of the brushes (by QCM) when the percentage of DMSO was increased in the aqueous solvent mixture. Understanding the influence of such solvents on frictional properties can lead to a better understanding of the role of solvation in brush-mediated lubrication.

5. References

Performance Evaluation of TiN Duplex Coating Sliding Against an Aluminum Alloy under a Transfer Condition

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1. Introduction
The titanium nitride (TiN) duplex coating has potential to be useful for tools and machine parts. Unlubricated sliding experiments on the duplex treated alloy tool steel blocks against aluminum alloy rings were conducted under the oxidation-limited condition simulating the tool and machine surface where the transfer of aluminum alloy occurs.

2. Test specimens and their surface characteristics
Substrate of test block specimens is made of hot work tool steel JIS SKD61 with a heat-treated hardness of 480 HV. Two kinds of test specimens are prepared i.e., the duplex-coated specimen (DM specimen) and the ion-plated specimen without a nitriding (IP specimen). For duplex-coated specimen the substrate was subjected to the plasma bright nitriding. Then the specimens were tempered in hydrogen at 823 K for 2, 4, 8 and 16 hours respectively. TiN film was coated by HCD ion plating on them. For both DM and IP specimens, TiN film hardness is about 2300 HV with a measurement load 10 mN. The residual compressive stress of the film is about -2.0 GPa for DM, and about -2.3 GPa for IP specimen.

3. Sliding experiment
Sliding experiments were conducted on a ring-on-block arrangement. A block specimen was slid against a ring specimen, made of aluminum alloy JIS A6063-T5, under a load of 250 N at a sliding speed of 0.1 m/s in nitrogen without lubrication.

4. Result and discussion
In the sliding experiments the transfer of aluminum alloy was predominating and little wear of the TiN film took place. However, after sliding for a certain distance, under an Al-alloy transfer layer a shell-like chipping consisting of several fractures appeared on the TiN film surface (Fig.1). The chipping grew downstream and caused a macroscopic spalling of the film.

The film life defined as the sliding distance at which the spallation larger than 50 to 100 μm occurred is markedly prolonged by the tempering of the substrate. Figure 2 shows the film life as a function of the tempering duration in hydrogen. The film life of DM and IP increases with tempering duration. The film life of IP remarkably improves with 4 hours tempering.

Using Smith’s expression, the tensile stress arising on the film surface is calculated to be as large as 1 GPa with a transfer layer thickness of 70 μm, a contact width of 200 μm, which is the same magnitude of a representative size of wear debris, and a friction coefficient of 1.2, which is a maximum value observed. This calculated tensile stress is smaller than the film strength, which is estimated to be about 3 GPa with intrinsic film strength of 1 GPa and its compressive residual stress of 2 GPa. Under the repeated which is as large as 1 GPa stresses, the film fails with a fatigue phenomena.

Tempering the substrate in hydrogen reduces substrate surface oxidized layer and reduces hydrocarbon contamination. It increases the strength of the substrate-film interface and causes a long film life.

5. Conclusions
Through the sliding test against Al-alloy under the condition at which a transfer of Al-alloy occurs, the followings were revealed.
(1) TiN film shows a characteristic failure behavior. That is, after sliding for a certain distance, a shell-like chipping appears on the TiN film surface under an Al-alloy transfer layer. The chipping grows downstream to cause a macroscopic spalling of the film.
(2) The tempering of the substrate in hydrogen before a coating prolongs the film life markedly. This is caused by the improvement of the strength of the interface between the substrate surface and the film.

6. References

Fig.1 Typical microscopic failure of TiN
2 hours tempering, Sliding distance : 4000m
The arrow indicates the sliding direction

Fig.2 Relationship between the film life and the tempering duration
Study on Improvements in Power Transmission Efficiency of the Toroidal CVT

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1. Introduction

Various types of automotive transmissions exist that have achieved remarkable improvements. For example, Fig. 1 shows an infinitely variable transmission (IVT) that consists of a toroidal-type speed-ratio changing mechanism (variator) and a planetary gearset. For a traction-drive continuously variable transmission (CVT), major technical challenges related to variators including higher transmitting efficiency, higher-torque capacity, greater downscaling, higher cost-performance, and others need to be overcome.

The aim of this study is to increase the transmitting efficiency of variators. The first of two items to be discussed is the use of microtexture on the traction surface as a means of increasing the traction coefficient. The second item is an optimized design of power roller bearings. The bearings are one of the major sources of power loss in a toroidal variator. Evaluations of the newly designed power roller bearings are based on experimental and nonlinear FEM analysis results of the power roller.

2. Improved Traction Characteristics

In order to improve traction characteristics by increasing the traction coefficient, the traction surfaces were textured with micropatterns. Figure 2 shows examples of the surface textured. The surface of example (d) has an improved traction coefficient of 3.0 percent without sacrificing rolling fatigue life of the variator according to experimental results. Micro-elastohydrodynamic lubrication (micro-EHL) analysis1 results introduced in this paper also demonstrate that there is no metal-to-metal contact, which cause surface flaking, and show an improved traction coefficient of 2.0 percent.

3. Optimization of the Power Roller Brings

Figure 3 shows a schematic diagram of the power roller bearing used in this study. The defined contact angle of the conventional bearings is 90 degrees. Figure 4 shows load distributions for a conventional power roller bearing and the optimized power roller bearing. Nonlinear FEM analysis results indicate that loads acting on the rolling elements of a conventional bearing depend on the position and location of the rolling elements as shown in Fig. 4 (a), where load distribution is nonuniform and is caused by deformation of the support members (trunnion). Support members cannot avoid deformation because the power roller is subjected to high thrust loads in the toroidal variator. Nonlinear FEM analysis results show that load distribution becomes uniform at a contact angle of 102.5 degrees as shown in Fig. 4 (b). Experimental results, which used power roller bearings at different contact angles, demonstrate that the newly designed power roller achieves higher transmitting efficiency and longer fatigue life than conventional power rollers, where the optimized contact angle depends on the type of support members.

4. Conclusion

The microtextures on the traction surface are able to improve the traction coefficient without sacrificing rolling fatigue life. Experimental and analysis results demonstrate that an optimized design improves fatigue life. These results contribute to improving the transmitting efficiency and torque capacity of the toroidal CVT.

5. References


Fig.1 Infinitely variable transmission
Fig.2 Surface profile
Fig.3 Contact angle of the power roller bearing
Fig.4 Load distributions for power roller bearings
Wear Resistance of Porous High Pressure Crystallized UHMWPEs as Artificial Cartilage Biomaterials

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1. Introduction

Wear and lifetime of ultra high molecular weight polyethylene (UHMWPE) are two major concerns when using polymer-on-metal bearing couples, especially as an artificial cartilage. Crosslinking is a well-known technique to reduce wear. However, it could lead to long-term oxidation and a substantial reduction in mechanical properties. Williams et al.2 suggested that the increase of biological activity due to the decrease in debris size, and the decrease in wear rate of crosslinked polyethylene should decrease the osteolytic potential. Bassette et al.3 reported that higher crystallinity and extended chain morphology induced in UHMWPE in the hexagonal phase at temperature and pressure above the triple point can lead to the improvement of mechanical properties. In this so-called high pressure crystallization (HPC) process, the growth of extended chains in the hexagonal phase was attributed to the ability of chain association to reduce less hindered crystallization kinetics. Oral et al.4 suggested that the use of a plasticizing agent (vitamin E) in UHMWPE during HPC, could reduce the hindrance on the crystallization kinetics. In this work, we adapted the biomimetic concept from Neville et al.5 and Wu et al.6; and applied vitamin E as a plasticizing agent as well as an antioxidant in porous HPC UHMWPE.

2. Experimental

Porous UHMWPEs were prepared by molding a GUR4130 UHMWPE-based powder (molecular weight: 5.6×10^6 g/mol) which was mixed with NaCl particles (particular diameter: 0.12±0.04 mm) through a HPC process. For each HPC process, 0.5 g powder was heated to 180°C and kept at that temperature for 3 h, after which the pressure was raised to 310 MPa. The temperature and pressure were held constant for 7 h. The sample, as shown in Fig.1, was then water-cooled down to room temperature under pressure; and the pressure was released subsequently. Vitamin E (D, L-α-tocopherol, >98%, Fisher Scientific) in 0.3 wt. % was further incorporated into the porous material by stirring UHMWPE-based powder in a xylene solution for 3 h before the HPC process. A solid UHMWPE without blending vitamin E during HPC was also made as a benchmark. Two porous HPC specimens, with or without vitamin E, were aged under 80°C in the air for 5 weeks. Their long-term oxidation effect on wear resistance was then evaluated. Differential scanning calorimetry was used to detect samples’ crystallinity and oxidative stability. Infrared spectra of the original and aged specimens were studied to compare their long-term oxidation behaviors. The surface morphology and the porosity of the freeze-fractured cross section were observed by a scanning electron microscope. The tribological properties, including wear rate and friction coefficient, of the UHMWPEs were investigated by using a pin-on-disk tester (average pressure = 18 MPa, sliding speed = 0.03 m/s, sliding distance = 562.5 m) under the lubrication of a normal saline. The wear debris were collected and their dimensions were statistically analyzed to survey the possibility of biological response.

3. Results

The material properties of the porous UHMWPE containing vitamin E as a plasticizing agent and an antioxidant were discussed. The synergistic tribological performance of the mentioned material properties and the biomimetic porous structure under lubrication was also addressed.

Fig.1 A typical UHMWPE sample (left).

4. Conclusions

Crystallinity and oxidation stability could be enhanced by using vitamin E during HPC process. The synergetic reduction of wear rate resulted from the squeeze-film effect of porous structure during boundary lubrication and the mechanical property was shown. The tribological results and thermal analyses of aged samples also confirmed the possible long term usage of this promising material.

5. References

Experimental and Numerical Investigations for Analysis of Temperature Rise on the Traction Contact Surface of Toroidal CVTs

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1. Introduction

A half-toroidal continuously variable transmission (CVT) with higher efficiency is needed to reduce fuel consumption. In order to raise efficiency even more, the designed traction coefficient must be set higher than before. Temperature rise in traction contact areas is one important factor that influences traction coefficient. For examining the influence of temperature rise on the traction coefficient, it is necessary to first clarify temperature rise in the traction contact area.

In this paper, we show measured temperature distributions on the traction contact surface under conditions that are typically found in a toroidal CVT using a twin-disc test machine and thin-film sensors firstly; secondary we compare calculated temperature rise using an analytical program with the measured temperature rise, and present an analytical program that can evaluate temperature rise on the traction contact surface; finally, we present and discuss calculated temperature rises on the traction contact surface of a toroidal CVT.

2. Comparison between the calculations and measurements

Temperature distributions were measured in an FZG twin-disc test machine. A thin-film platinum temperature sensor was sputtered on the discs. The changes in electrical resistance of the sensor, resulting from changes in temperature and pressure when the sensor passed through the contact surface, were measured using a Wheatstone bridge.

The solid lines in Fig. 1 show measured results of temperature distribution in the elliptical contact areas for varying degrees of slip ratio under no-spin conditions1. REIB992, which is a traction analysis program developed at the FZG, was used to calculate temperature distributions in the contact area. In this program, the elliptical contact zone is divided into grid elements, and local shearing stress at each element is calculated using viscoelastic and elastoplastic models. Local temperature in each element is calculated as the total sum of (a) bulk temperature of the rolling parts, \( \theta_{be} \), (b) temperature rise at the edge of entry, \( \Delta \theta_{be} \), (c) integrated temperature rise from edge of entry to the element, \( \Delta \theta_{int} \), and (d) temperature rise at the element, \( \Delta \theta_{el} \).

Broken lines in Fig. 1 show the calculated temperature distribution in the elliptical contact areas under the same conditions for measurements. Characteristics of the calculated results were qualitatively in good agreement with measured results. Measured temperatures were more or less two times higher than the calculated temperatures under any condition in the experimental range. Thus, it seems reasonable that the traction analysis program, REIB99, can simulate the tendency of temperature rise on the contact surface.

3. Analysis of temperature rise in toroidal variators

Temperature rise on the traction contact surface of the full- and half-toroidal variator was calculated using the above verified traction analysis program, REIB99. The calculated conditions are similar to real-vehicle operating conditions; \( T_{in} = 350 \text{ Nm}; N_{in} = 6 \times 10^3 \text{ rpm}; i_v = 2.236; \theta_{id} = 120 °C \). While the maximum temperature rise of the half-toroidal variator was 42 K, the maximum temperature rise of the full-toroidal variator reached 294 K (i.e. Celsius contact temperature of the rolling parts is 414 °C). Temperature values in the traction contact surface were calculated without considering the measured temperature that was two times higher as shown in section 2. Thus actual temperature rises are expected to be even higher. The influence of variator ratio, input torque, and input rotational speed on temperature rise was also calculated. In case of the full-toroidal CVT under the maximum torque, it was necessary to cool the input disc at the point of its lowest reduction ratio, and to cool the output disc at the point of its highest reduction ratio. As for the half-toroidal CVT, cooling the discs was most effective when the variator ratio was between -1.5 and -0.75.

4. References

2. Graswald, C., Diss. TU München, 2001

Fig. 1 Temperature rise on the traction contact surface of measurements (solid lines) and calculations (broken)

\( P = 980 \text{ MPa}; v_1 = 12 \text{ m/s}; \theta_{id} = 50 °C; \theta_{id} = 50 °C \)
Deep drawn Cup Profile Thickness Studies on 304 L Steel Sheet Using Vegetable Oils as Lubricant

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1. Introduction

The most important factor deciding on the success of deep drawing operation is initiating the metal flow\textsuperscript{3}. The role of lubricant is vital in initiating the metal flow, which dictates the quality of the component\textsuperscript{2}. Formulated non-edible vegetable oils, Pongam (\textit{Pongamia pinnata}) and Jatropha (\textit{Jatropha carcas}) are used for deep drawing of 304 L steel sheets and the cup profile thicknesses are studied.

2. Experiments and results

The deep drawing operations are carried on stainless steel 304 L using a hydraulic press under different loads. The 3D images (Fig.1) of drawn cups are obtained by laser scanning. Further, punch load as a function of stroke and draw-in-length for various lubricants is studied.

![Fig.1 3D image of drawn cup and its sectional view](image)

![Fig.2 Variation of cup wall thickness with stroke](image)

It is observed from the results (Fig.2) that, for the given stoke of 25 mm, cups drawn under Pongam raw oil (PRO), Pongam methyl esters (PMEs), Jatropha methyl esters (JMEs) and Mineral oil (MRO) ruptured during the operation. Under Jatropha Raw (JRO) and Epoxidised Jatropha (JREp) oils lubricated condition, almost uniform material flow is observed in terms of wall thickness of the cups.

The punch load (Fig 3) necessary for drawing full cups is found to be lowest under Jatropha and its epoxidised mode.

![Fig.3 Variation of punch load with stroke](image)

Larger draw-in-lengths are observed for Jatropha raw oil compared to others, owing to better lubrication between the punch and die (Fig.4).

![Fig.4 Draw-in-length for different lubricants](image)

3. Conclusion

Deep drawn cups using Jatropha raw lubricant and its epoxidised version showed uniform wall thickness profiles and larger draw-in-lengths compared to other modes of lubrication.

4. References

Application of DLC Film Deposited by Unbalanced Magnetron Sputtering to Rolling Bearing

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1. Introduction

Diamond-like carbon (DLC) coatings have been used in various applications. DLC films can prevent damage in mechanical elements, including bearings. However, their usage is limited due to the fact that the DLC coatings exhibit low endurance under high contact pressure conditions.

In the previous study, the mechanism of delamination of the intermediate layer of a DLC film under high contact-pressure conditions was investigated experimentally and analytically1. Results showed that the delamination resistance of DLC films depends strongly on the Young’s modulus of the intermediate layer.

In this paper, we study seizure resistance of a DLC film using a two-roller testing machine and a spherical roller bearing under starved lubrication conditions.

2. Experimental

The DLC film consisted of three layers, i.e. a metal layer, an intermediate layer, and a carbon layer, which were applied to a bearing steel substrate using unbalanced magnetron sputtering (UBMS) (Fig.1). The intermediate layer for DLC1 consisted of tungsten and carbon while the layer for DLC2 consisted of silicon and carbon. Young’s moduli of the intermediate layers of DLC1 and DLC2 were 340 GPa and 240 GPa respectively. Previous studies showed that the delamination resistance of DLC2 was good under rolling contact1. Therefore, two-roller and bearing tests were conducted using these two DLC films.

In the two-roller tests, a coated and an uncoated disks were tested under maximum stress conditions of 1.5 GPa and a sliding speed of 1.5 m/s with a slip ratio of 26.7 %. Seizure resistance was evaluated from the time lubricant was no longer fed until seizure.

Smearing tests using spherical roller bearing with the coated disks were conducted under starved lubrication conditions. The test bearing dimensions were as follows: 370 mm outer diameter, 220 mm bore, and 150 mm width. Smearing was determined by examination of surface damage and cross-sectional structure after testing.

3. Results

The results obtained from the two-roller tests are shown in Fig.2. The coated rollers were superior to the uncoated rollers in terms of seizure life. Furthermore, the seizure life of DLC2 was four times longer than that of DLC1. Surface examination revealed DLC film delamination. Therefore, DLC films that are highly resistant to delamination can extend bearing seizure life.

For the uncoated bearing test, visual examination revealed that the bearing surface was damaged by adhesion and that the sectional structure changes are due to surface softening (Fig.3(a)). On the contrary, the DLC films on the rollers remained undamaged (Fig.3(b)), and smearing was not observed on the bearing with coated rollers.

4. Conclusions

DLC films can prevent seizure and smearing under starved lubrication conditions. Therefore, it is important for a DLC film to retain its effectiveness over the long periods of time in addition to being highly resistant to delamination.

Results obtained showed that DLC films have a potential to be useful in applications to rolling bearings.

5. References

Tribotests on AA6061 and AISI 1040 with Vegetable Oils as Lubricants

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1. Introduction

The long and polar fatty acid chains of triglyceride structure provide high strength lubricant films that interact strongly with metallic surfaces, reducing both friction and wear. Two non-edible vegetable oils, Pongam (Pongamia pinnata) and Jatropha (Jatropha carcass) are used and their tribological behaviour with AA6061 and AISI 1040 material is studied.

2. Experiments and results

Experiments are conducted on AA6061 and AISI 1040 pins under the dry, mineral and vegetable oil lubrication using a pin-on-disc tribometer. Friction, wear and pin temperatures are measured for various loads and sliding distances. Also, the Strubeck curves are drawn to compare the regimes of lubrication.

Rapid and moderate drop in wear for AA6061 and AISI 1040 respectively are observed under Pongam oil (PRO) and Jatropha oil (JRO) for the whole range of loads and sliding distances, compared to mineral oil (MRO) (Figs.1 and 2).

AA6061 showed (Fig.3) about 66 % and 55 % drop in co-efficient of friction (COF) under Jatropha and Pongam oil respectively, compared to the petroleum based lubricant. However, for AISI 1040 (Fig.4), almost similar trends are noticed for both vegetable oils and mineral oil.

3. Conclusion

Friction and wear for AA6061 and AISI 1040 under both vegetable oils modes of lubrication are lower compared to mineral oil. Strubeck curves prove their boundary lubrication property.
Reynolds Equation Flow Factor Estimates by means of Homogenization

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1. Introduction

The way of incorporating the effects of roughness through flow factors first presented by Patir & Cheng (P&C) has been widely used ever since. Although this method is still the first choice for the application engineer an averaging technique based on homogenization theory is at present starting to attract the attention of the scientific community. The reason is that homogenization provides a rigorous framework for deriving flow factors for different types of Reynolds problems\(^3\).\(^2\). The homogenization process is easy to follow and it has been shown that the P&C flow factors approximate the corresponding ones obtained by homogenization under certain circumstances.

Lately Almqvist et al.\(^2\) have employed a technique, which has proved to be highly successful in composite engineering. This homogenization technique provides estimates, or bounds, of averaged properties. This paper is an effort towards further introducing the concept of homogenization and particularly to introduce a method to estimate the flow factors that consider the effects roughness in the averaged form of the Reynolds’ equation.

2. Governing equations

In this paper we consider the case when one surface is moving and the other one is stationary. Moreover, the moving surface is assumed to be smooth while the stationary may be rough. The following Newtonian, incompressible Reynolds equation is used

\[ \nabla \cdot (h \nabla p) = 6 \mu U \nabla \cdot (h e) \text{ on } \Omega, \quad (1) \]

where \( \mu \) is the viscosity and \( U \) is the speed of the smooth surface in the \( x_1 \)-direction. The film thickness \( h \) is modeled by

\[ h(x, y) = h_0(x) + h_1(y), \quad (2) \]

where \( h_0 \) describes the global scale, i.e., the geometrical properties of the application and \( h_1 \) describes the roughness contribution.

3. Homogenization

In this case, the homogenized problem becomes

\[ \nabla \cdot (A \nabla p_h) = 6 \mu U \nabla \cdot b \text{ on } \Omega, \quad (3) \]

where

\[ A = \int \left( \begin{array}{ll} h^3 \left( 1 + \frac{\partial u}{\partial y_1} \right) & h^2 \left( \frac{\partial u}{\partial y_1} \right) \\ h^2 \left( \frac{\partial u}{\partial y_2} \right) & h \left( 1 + \frac{\partial u}{\partial y_2} \right) \end{array} \right) dy, \]

\[ b = \int \left( h - h^2 \left( \frac{\partial u}{\partial y_1} \right) - h^2 \left( \frac{\partial u}{\partial y_2} \right) \right) dy, \]

and where \( u_i \) are the solutions to the local problems;

\[ \nabla \cdot (h \nabla u_i) = \nabla \cdot (h e_i) \text{ on } \Omega, \]

\[ \nabla \cdot (h \nabla u_i) = -\nabla \cdot (h^2 e_i) \text{ on } \Omega, \]

\[ \nabla \cdot (h \nabla u_i) = -\nabla \cdot (h^2 e_i) \text{ on } \Omega. \quad (4) \]

4. Flow factor estimates

Solving (3) requires less effort than solving (1). However, the number of degrees of freedom introduced for solving these for a measured surface may still exceed the capabilities of today’s technology. Therefore a concept of deriving estimates for averaged properties is explored here. The outcome of the analysis is

\[ \nabla \cdot (A p^* \nabla) = \nabla \cdot (b^* e) \text{ on } \Omega, \quad (5) \]

where

\[ a^* = \int \left( \int h \nabla y \right) \nabla y^{-1} dy, \quad a^* = \int \left( \int h \nabla y \right) \nabla y^{-1} dy, \]

\[ b^* = \int \left( \int h \nabla y b \right) \nabla y^{-1} dy, \quad b^* = \int \left( \int h \nabla y b \right) \nabla y^{-1} dy \]

In Figure 1, pressure solutions for a particular choice of roughness are depicted. The upper left corner shows the deterministic solution plotted together with the estimates. The upper right corner shows an enlarged portion and in the bottom part the estimates are plotted together with the homogenized solution. It is clear that the estimates very closely bound the homogenized solution.

![Fig.1 Pressure solutions.](image)

5. References

Phosphonium Ionic Liquids As Lubricants For Aluminium-Steel

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1. Introduction

The use of aluminium alloys in technical applications is increasing due to their high strength-to-weight ratio, corrosion resistance and high thermal conductivity. Unfortunately aluminium alloys perform poorly in moving contact situations and suitable lubricants are yet to be discovered [1]. Room temperature ionic liquids (ILs) are organic salts that are liquid at room temperatures. ILs have a number of properties which make them suitable candidates as lubricants, such as low volatility, non-flammability and thermal stability. Since ILs have good solubility with organic compounds they can be used with current oils and additives [2].

2. Experimental

The performance of a series of novel room temperature ionic liquids (ILs) based on the trihexyl-(tetradecyl)phosphonium cation (P₆₆₆₁₄⁺) and a number of novel anions was studied as lubricants in pin-on-disk tests using a 6mm 100Cr6 steel ball on AA2024 aluminium disks. The anions coupled to the P₆₆₆₁₄⁺ cation included diphenyl phosphate (DPP-), dibutyl phosphate (DBP-), bis(2,4,4-trimethyl penty) phosphate (M₆₆₆₁₄PPh) and bis(2-ethyl hexyl) phosphate (BEH-). More traditional anions such as bis(trifluoromethanesulfonyl) amide (NTf₂) and bromide (Br⁻) were also investigated.

3. Results

Figure 1 shows the average coefficient of friction and Figure 2 the wear coefficient as a function of the load for the six ionic liquids and one fully formulated diesel engine oil tested.

For the ILs tested the friction coefficient results show that the P₆₆₆₁₄Br sample had the lowest friction and P₆₆₆₁₄DPP and P₆₆₆₁₄TFSA were consistently lower than P₆₆₆₁₄BEH and P₆₆₆₁₄M₆₆₆₁₄PPh and all were much lower than the P₆₆₆₁₄DBP and the engine oil.

Fig.2 Wear coefficient vs load

For the wear coefficient (Fig. 2) it can be seen that the aluminium disks lubricated with the P₆₆₆₁₄TFSA, P₆₆₆₁₄Br and P₆₆₆₁₄DPP ILs resulted in very low wear coefficients.

EDS of the wear scars from the 30N samples showed the presence of various IL elements, such as F, S and P on the P₆₆₆₁₄TFSA, P₆₆₆₁₄BEH and P₆₆₆₁₄DPP samples only, suggesting that a complex tribofilm had formed on these. The result for the P₆₆₆₁₄TFSA, shown in Figure 3, had the largest oxygen peak, as well as a substantial fluorine peak. This suggests that a relatively thick tribofilm may have formed.

4. Conclusions

Phosphonium based IL lubricants performed well in a pin-on-disk wear test using a steel ball on an aluminium disk. It is suggested that they may form complex tribofilms on the surface that act to reduce friction and wear.

5. References

Classification of Bone Texture for Detection of Early Knee Osteoarthritis

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1. Introduction

Osteoarthritis (OA) is a prevalent knee joint disease and the leading cause of functional disability in older adults. Since no cure for knee OA currently exists, it is of great importance to be able to detect the disease in its early stage. For the detection, OA changes in articular cartilage, trabecular bone (TB) or synovial fluid are assessed. Recently, it has been suggested that the bone changes occur first [1] and subsequently, TB can be used for detection of early knee OA.

In this paper, TB texture images extracted from knee radiographs are used (Figs.1a-b). OA changes in the images are assessed using a signature dissimilarity measure (SDM) method [2]. The use of the method in radiographic knee OA detection, based on texture analysis, is demonstrated.

2. Method

The SDM method produces roughness and orientation signatures for each TB texture image. For the roughness signature, the sharpness of microstructural bone elements (trabeculae) is quantified. For the orientation signature, the alignment of the trabeculae is measured. The signatures are invariant to imaging conditions such as magnification, exposure, noise, and blur. An example of TB texture image and its corresponding signatures are shown in Figs.1b-d.

3. Experiments

To evaluate the performance of the SDM method in knee OA detection, 137 TB texture images were extracted from 17 healthy and 34 OA subjects. The images were used to form healthy and OA classes (Table 1). The SDM method was combined with a support vector machine (SVM) classifier and a SDM-SVM classification system was constructed. The system was compared against a benchmark weighted neighbor distance using compound hierarchy of algorithms representing morphology (WND-CHARM) classification system [3].

Table 1 Details of healthy and OA classes

<table>
<thead>
<tr>
<th>Class</th>
<th># images</th>
<th>Mean age/BMI of subjects</th>
</tr>
</thead>
<tbody>
<tr>
<td>Healthy</td>
<td>68</td>
<td>40.8 / 24.6</td>
</tr>
<tr>
<td>OA</td>
<td>69</td>
<td>44.5 / 27.1</td>
</tr>
</tbody>
</table>

4. Results

Classification accuracy (percentage of correctly classified images), sensitivity (percentage of OA images classified as OA) and specificity (percentage of healthy images classified as healthy) were calculated for each system using a leave-one-out cross-validation method (Table 2). The SDM-SVM system showed the best performance.

Table 2 Performance of the two systems evaluated

<table>
<thead>
<tr>
<th>System</th>
<th>Accuracy/sensitivity/specificity</th>
</tr>
</thead>
<tbody>
<tr>
<td>SDM-SVM</td>
<td>85.4 / 87.0 / 83.8</td>
</tr>
<tr>
<td>WND-CHARM</td>
<td>64.2 / 58.0 / 70.6</td>
</tr>
</tbody>
</table>

5. Conclusions

The SDM-SVM classification system achieved the highest values of accuracy, sensitivity and specificity. The results obtained indicate the potential application of the system for detection of early knee osteoarthritis.

6. References

Optimization of Surface Texture Shapes in Hydrodynamic Contacts:
Two-Dimensional Bearings

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1. Introduction

In recent years, a number of studies have focused on the application of surface textures in hydrodynamic contacts. The results have shown that the use of textured surfaces leads to the improvement of the tribological characteristics of bearings and other mechanical elements by increasing their load carrying capacity and reducing frictional losses.

However, despite the numerous studies conducted, there has been no systematic and efficient approach to finding optimal surface texture parameter values. Most studies have been carried out by means of exhaustive search, enumerating all possible parameter configurations and checking their respective tribological characteristics.

In this paper, an approach is proposed for surface texture configurations governed by the two-dimensional Reynolds equation. This approach will enable treating bearing optimization problems as a parameter optimization problem using available optimization routines, such as Matlab functions.

2. Method

In this study, a new unified approach is proposed, basing on the optimal parameter selection theory combined with nonlinear programming optimization routines. Optimization is conducted with respect to surface texture parameters for the maximum bearing load capacity. Consequently, the objective function is the integral of the pressure over the entire pad area. The optimal parameter selection problem thus becomes:

\[ \text{maximize } W = \iint_{X,Y} p \, dxdy \]

Subject to following constraints:

- Pressure boundary conditions
- Reynolds equation in two dimensions

A bearing surface texture is modeled by a function of surface parameters \( h \). Parameters used in this investigation are following: dimple height ratio \( \frac{h}{h_0} \), area ratio \( \frac{A_{textured}}{A_{film}} \) and textured portion of the slider \( \alpha \) as shown in Fig. 1, where \( r_1, r_2 \) – dimple cell radii in x and y direction, respectively; \( r_b \) – dimple base radius; \( h_0 \) – dimple height; \( h_0 \) – film height. Surfaces with rectangular dimples described by same set of parameters were also analyzed.

3. Results and discussion

A series of preliminary results for bearing surfaces covered with both rectangular and elliptical dimples was obtained. The validity of the results was proved by comparing them to the results obtained from the exhaustive search. A partial comparison of results for elliptical dimples is shown in Table 1. The optimal load was found to be proportional to the dimple area ratio. The optimal textured length of the slider was about 0.6 for all dimple configurations; only the optimal values of dimple height ratio are listed.

<table>
<thead>
<tr>
<th>Number of dimples in y-direction</th>
<th>Optimization</th>
<th>Exhaustive search</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Load capacity</td>
<td>Load capacity</td>
</tr>
<tr>
<td></td>
<td>Optimal  ( H )</td>
<td>Optimal  ( H )</td>
</tr>
<tr>
<td>5</td>
<td>0.0287</td>
<td>2.1390</td>
</tr>
<tr>
<td>10</td>
<td>0.0285</td>
<td>2.1481</td>
</tr>
<tr>
<td>20</td>
<td>0.0292</td>
<td>2.1611</td>
</tr>
</tbody>
</table>

The results for the elliptical dimples were in good agreement to those presented by Brizmer et al.\textsuperscript{1,2,3} This has proved that the approach presented in this paper is capable of finding optimal bearing texture shape parameter values using minimum number of calculations.

4. References

Electrochemical Methods to Monitor Corrosive Wear of Ferrous Alloys

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1. Introduction

Electrochemical methods are powerful techniques used in both research and industry for corrosion investigation. When materials are subjected to simultaneous action of wear and corrosion, electrochemical methods can alert to an unusual damage rate. The purpose of this study is to point out critical factors that influence electrochemical characterization of three-body abrasion-corrosion of ferrous alloys in seawater.

2. Experimental

A newly designed rotating disc-on-plate tribometer that allows simultaneous investigation of abrasive wear and electrochemical corrosion (tribocorrosion) has been used1). Linear polarization resistance, Tafel polarization and current vs. time measurement techniques have been applied to monitor conjoint action of corrosion and wear. Commercial grade 316L stainless steel, 52100 ball bearing steel (annealed and quench & tempered (Q&T)) and high chromium cast iron (HCCI) (30%Cr, 4%C) were the alloys investigated.

3. Results and Discussion

Figure 1 shows how hard particles with different shapes affect current evolution while abrading 316L sample in seawater at corrosion potential. It can be seen that mild wear due to rolling and shallow indentation by round glass beads and semi-round garnet grits did not increase the current significantly. On the other hand, more severe wear due to indentation by angular alumina grits increased the current sharply. When abrasive wear was eliminated, the current decreased and corrosion resistance of 316L was restored.

For AISI 52100 samples that do not passivate in seawater, the corrosion current measured was similar for both corrosion only and abrasion-corrosion conditions. The current did not depend on the shape of abrading particles, as shown in Fig. 2. The only difference that could be measured was a level of corrosion potential that decreased with increasing severity of wear. Most of the wear synergism is the acceleration of wear due to corrosion.

HCCI sample with multiphase microstructure exhibits a weak passive layer in seawater that protects the ferrous matrix from dissolution. Figure 3 shows that three-body abrasive wear increased the corrosion current of HCCI in seawater. Doubling the applied load increased the current further. In contrast to a single-phase 316L sample, eliminating abrasive wear did not decrease the current for the HCCI sample. The continuing current increase can be explained as a detrimental effect of wear damage on the corrosion resistance of HCCI in seawater.

4. Conclusion

Application of electrochemical methods for monitoring corrosive wear of engineering alloys works best for passivating alloys and alloys with homogenous microstructures.

5. References

Surface Enhanced Micro-Raman Spectroscopy of Tribo-formed Boundary Films of Cold-Pressed Canola Oil

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1. Introduction
It was hypothesized that polyunsaturated triglycerides could polymerize and form thick boundary films under metallic sliding contact, where high temperature and pressure and metallic wear particles acting as catalyst are prevalent. In an earlier work, we tested three refined oils and nine unrefined cold-pressed plant oils, and showed for the first time, that the boundary film thickness of some unrefined plant oils are able to grow up to 40-100 nm in a ball-on-disk sliding experiment. However, no correlation could be found between the degree of unsaturation and the growth rate of these films.

The aim of this study is to determine the chemical structure of the thick films formed, and to ascertain whether the growth was indeed due to polymerization. Surface-Enhanced Raman Spectroscopy (SERS) was selected for this purpose. This is the first time SERS has been utilized in the study of plant oil boundary films.

2. Experimental Details
The bearing ball used in the sliding experiment was electroplated with silver prior to the test. Ball-on-disc sliding experiments (as described earlier) were performed for unrefined canola oil. The ball was then carefully removed, briefly immersed in hexane, air-dried and then analyzed under a LabRam 1B Raman spectrometer equipped with a 4 mW He-Ne laser, and coupled with a Olympus BX40 microscope. Raman spectra was collected at different points on the wear scar with a 632.8 nm line at the resolution of 4 cm⁻¹ and a laser spot size of ~2 μm in diameter.

3. Results
The ball wear scar and the Raman spectra obtained from 2000 to 200 cm⁻¹, are shown in Fig. 1a and b respectively:-

Spectrum B - The peaks at 226, 295, 411, 499 cm⁻¹ are characteristic of α-Fe₂O₃ (hematite), while the strong peak at 659 cm⁻¹ can also be found in γ-Fe₂O₃ (maghemite) and γ-FeOOH (lepidocrocite). This confirms our earlier postulation that the dark patches are mainly ferrous and ferric oxides.

Spectra C - The disappearance of the 1656 cm⁻¹ (cis C=C stretching) band, typical of unsaturated triglycerides, indicates that auto-oxidation or polymerization had occurred. Two broad bands are observed at 1290-1400 cm⁻¹ (C-H bending) and 1570-1600 cm⁻¹ (C=C ring stretching). These two bands are characteristic of hydrogenated amorphous carbon, which suggests that polymerization could have proceeded by conjugation of the double bonds and subsequent cross-linking of fatty acid chains forming more complex carbon ring structures.

4. Conclusions
The SERS results provide some evidence that the polyunsaturated triglycerides had undergone polymerization, possibly forming complex carbon ring structures. Confirmation of the presence of ferrous and ferric oxides supports the view that the steel surface was chemically attacked by hydroperoxides and lipid peroxides in the sliding contact area. Future tests, however, would be required to elucidate the mechanism of its growth.

5. References
1. Chua, W., Stachowiak, G. W., Tribol. Lett (accepted Nov 2010)
The Influence of Hydrostatic Recesses on Spring Supported Tilting Pad Thrust Bearings

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1. Introduction

This study investigates the effect of adding a hydrostatic groove to the plane pads of a hydrodynamic thrust bearing. Computer simulations show that the presence of a groove can improve the performance of an un-optimised thrust bearing.

2. Development

Numerous hydroelectric power stations around the world, many built in the 1950s and 60s, utilise tilting pad thrust bearings that are not furnished with hydrostatic assistance. This causes durability problems due to bearing wipe at low speeds and high temperatures1.

The problem of what recess to use for providing hydrostatic capability, without detracting from fully-developed hydrodynamic performance, is not a trivial one – especially when retrofitting existing installations owing to the greater number of design constraints. This work investigates the influence of different types of recesses, with identical footprint, and how their performance compares to the plane pad.

An isothermal finite difference simulation was used to assess parameters such as nominal oil film thickness $h_{nom}$, convergence $K = h_1/h_0 - 1$, average temperature rise (adiabatic) $\Delta \theta$, and power dissipation per pad, $\Pi_p$. Bearing parameters found in a typical hydroelectric installation were used in the model. The pressure distribution for a specific test case is shown in Fig.1.

Table 1 Basic dimensions for three recess profiles (mm)

<table>
<thead>
<tr>
<th>Type</th>
<th>$R_g$</th>
<th>$L_{slo}$</th>
<th>$L_{sli}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>60</td>
<td>0</td>
<td>50</td>
</tr>
<tr>
<td>B</td>
<td>55</td>
<td>10</td>
<td>45</td>
</tr>
<tr>
<td>C</td>
<td>45</td>
<td>20</td>
<td>35</td>
</tr>
</tbody>
</table>

Common dimensions: $R_e = 65$ mm; $A_r = 13,000$ mm²

3. Results and Conclusion

One surprising result is that the presence of a groove can improve the performance of an un-optimised thrust bearing by increasing the convergence towards the optimal value at lower values of viscosity and speed, as shown in Fig.2. The increased film thickness, combined with the presence of the groove, also reduces $\Delta \theta$ and $\Pi_p$. Caution is advised as increasing either lubricant viscosity $\eta$ or speed $N$ past a certain point can result in a deterioration of performance. This phenomenon is self limiting, however, as higher speed results in viscosity reduction due to thermal effects that are yet to be fully incorporated in the simulation.

It was found that a large external taper shifts the performance cross-over point to the left. Recess type C, for example, exhibited little benefit when compared to the plane pad. In fact, the isothermal simulation showed the oil film to collapse for $\eta \cdot N > 60$. Recess type A exhibited the best performance among the candidates tested.

4. References

The Effect of Buffer Solution Chemistry on Bovine Serum Lubrication of CrCoMo Alloy

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1. Introduction

In an earlier work we report a lubrication study of buffer solution (BS) and proteins dissolved in deionised water. The objective was to examine lubrication mechanisms in the absence of buffer salts as these are expected to contribute to surface film formation, thus obscuring the protein lubricating mechanisms. In the current paper we extend this approach to study the effect of BS chemistry (pH) on film thickness and wear. pH was varied from 7.4, which represents ‘healthy’ SF to 8.4 which represents diseased SF.

2. Experimental Programme

Central film thickness was measured in a ball-on-disc device (PCS Instruments) using thin film optical interferometry. The technique measures film thickness in the range 1-100 nm with a resolution of ±1 nm. The test used a commercial femoral resurfacing head (38 mm diameter, as-cast) as the stationary component. This was loaded against the glass disc to form a sliding contact (Fig. 1). The load (5 N) was held constant during the test and although this is not representative of the cyclic conditions occurring during the gait cycle, it does provide a measurement of the film thickness under the most severe loading and presumably high wear condition. The test temperature was held at 37°C throughout the test. The model synovial fluid solutions were bovine calf serum diluted to 25% w/w in deionised water and tris buffer (pH: 7.4, 8.0 and 8.4). Tests were carried out at variable (5-50 mm/s) and constant mean speed (0, 10 mm/s, 20 minutes). At the end of the constant speed tests the wear scar diameter on the femoral head was measured.

3. Results

Variable speed film thickness measurement results are shown in Fig. 2 for 25 BS in deionised water. The tests were run by measuring film thickness for increasing and then decreasing speeds. In Fig. 2 two speed sweeps (Up/Down 1 and Up/Down2) are shown.

The film behaviour is complex and usually time dependent with much thicker films forming at lower speeds as the test progresses. This behaviour appears to be partly due to the formation of a high-viscosity reservoir at slow speeds in the inlet region. We attribute this to aggregation of protein molecules in the inlet flow field. As this material passes through the contact it is deposited on the rubbed surface.

![Film thickness measurement method](image)

![Fig.1 Film thickness measurement method](image)

![Fig.2 Film thickness results for BS solutions](image)

![Fig.3 Effect of SF pH on film thickness](image)

4. Conclusions

It would appear from this study that patient SF pH does indeed play a role in implant wear and might contribute to premature failure.

5. References