Final Scientific Report

DOE Award:
DE-AI26-06NT42753

Recipient:
National Institute of Standards and Technology

An Integrated Surface Engineering Technology Development for Improving
Energy Efficiency of Engine Components

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Executive Summary

Frictional losses are inherent in most practical mechanical systems. The ability to control friction offers many opportunities to achieve energy conservation. Over the years, materials, lubricants, and surface modifications have been used to reduce friction in automotive and diesel engines. However, in recent years, progress in friction reduction technology has slowed because many of the inefficiencies have been eliminated. A new avenue for friction reduction is needed.

Designing surfaces specifically for friction reduction with concomitant enhanced durability for various engine components has emerged recently as a viable opportunity due to advances in fabrication and surface finishing techniques. Recently, laser ablated dimples on surfaces have shown friction reduction properties and have been demonstrated successfully in conformal contacts such as seals where the speed is high and the load is low. The friction reduction mechanism in this regime appears to depend on the size, patterns, and density of dimples in the contact.

This report describes modeling efforts in characterizing surface textures and understanding their mechanisms for enhanced lubrication under high contact pressure conditions. A literature survey is first presented on the development of descriptors for irregular surface features. This is followed by a study of the hydrodynamic effects of individual micro-wedge dimples using the analytical solution of the 1-D Reynolds equation and the determination of individual components of the total friction resistance. The results obtained provide a better understanding of the dimple orientation effects and the approach which may be used to further compare the friction reduction provided by different texture patterns.

Goals and Objectives of Project

Interagency Agreement No. DE-AI26-06NT42753 was established between the Department of Energy (DOE) and the National Institute of Standards and Technology (NIST) such that NIST would perform tasks on the project “An Integrated Surface Engineering Technology Development for Improving Energy Efficiency of Engine Components” over the period June 1, 2006 to May 31, 2009.

The project Principal Investigator, Dr. Stephen Hsu of the NIST Ceramics Division, Nanomechanical Properties Group, retired from NIST on January 3, 2007. In the absence of Dr. Hsu, the NIST Ceramics Division could not continue the project.

Dr Hsu took a position at George Washington University (GWU) and indicated that he wished to continue the project at GWU. NIST was willing to facilitate transfer of the project to Dr Hsu at GWU. A memorandum describing the actions taken by NIST to complete this transfer was submitted to DOE on July 14, 2008. These actions were accepted by DOE.

The scope of the project was reduced to single modeling element with the following milestones:

- Milestone 1: Conduct a literature survey on two new developments in descriptors of irregular surface features as they apply to frictional characteristics of textured surfaces: areal parameters derived from rheological-mechanical models; and 3-D parameters from the motif method based on the use of morphological watersheds. A list of publications in these two areas will be compiled and the results summarized as to the advantages and shortcomings of these two techniques in describing textured surfaces.
- Milestone 2: Develop an elasto-plasto-lubrication contact model to simulate friction reduction under high contact pressure conditions and develop a computer simulation model to predict optimum surface textures for friction reduction under a particular set of contact conditions.

At Dr. Hsu's request, the NIST Ceramics Division entered into a Cooperative Research Agreement with The Pennsylvania State University (PSU) for “Theoretical Modeling for Predicting Friction Reduction of Textured Surfaces.” DOE funds transferred as part of the Agreement were used to support a PSU post-doctoral research assistant assigned to NIST as an Associate. The Agreement extended over the period 4/2/07 to 3/31/08.

Technical direction and oversight of the research performed under the Cooperative Research Agreement was provided by Dr. Hsu. The Cooperative Research Agreement with PSU included Prof. Liming Chang as the Principal Investigator and Dr. Huan Zhang as a post-doctoral research associate.

The NIST Ceramics Division provided labor for *supervisory and administrative support* for transfer of the project to GWU, oversight of the NIST Associate, and creation of the Cooperative Research Agreement with PSU.

This report describes the research performed by Prof. Chang and Dr. Zhang as part of the Cooperative Research Agreement. The report was submitted to NIST by Prof. Chang and is included here unaltered.

Project Summary

An Integrated Surface Modification for Automotive & Trucks: Surface Feature Characterization and Model Development

1. Introduction

Frictional losses are inherent in most practical mechanical systems. The ability to control friction offers many opportunities to achieve energy conservation. Over the years, materials, lubricants, and surface modifications have been used to reduce friction in automotive and diesel engines. However, in recent years, progress in friction reduction technology has slowed because many of the inefficiency have been eliminated. A new avenue is needed.

Designing surfaces specifically for friction reduction with concomitant enhanced durability for various engine components has emerged recently as a viable opportunity due to advances in fabrication and surface finishing techniques. Recently, laser ablated dimples on surfaces have shown friction reduction properties and have been demonstrated successfully in conformal contacts such as seals where the speed is high and the load is low. The friction reduction mechanism in this regime appears to depend on the size, patterns, and density of dimples in the contact.

The objective of this project is to develop an application specific engineered surface (combining surface textures, thin films, surface chemistry) tailored for the optimum friction reduction and enhanced durability.

This friction reduction technology is aimed at various energy transmission components in an engine, particularly in a heavy duty diesel engine. Components such as cylinder liner ring contacts, cam and lifters, and overhead bearing components are being targeted. To meet the durability criteria, the texture patterns have to be protected by thin films/coatings and appropriate lubricating chemistry over a broad range of operating conditions. The final outcome of the project will be a general design principle and guidelines for various engine components under a wide range of operating conditions. To achieve the maximum energy conservation, the technology is being shared through an

international cooperative research under the International Energy Agency Implementing Agreement on Advanced Materials for Transportation (IEA IA-AMT). The organization effort under the IEA activities involves getting new countries to join, and UK, China, Australia, Finland, Sweden, Israel, and Japan have agreed to participate under the IEA Annex IV on surface technology.

From energy conservation point of view, most parasitic losses in engines take place under high load and medium to low speed conditions. Therefore, we need to extend the surface texturing designs from high speed low load conditions to low speed high load conditions. However, when we ran the same surface texture patterns under boundary lubrication conditions, friction actually increased. A collaborative effort with Northwestern University using a sophisticated elastohydrodynamic friction model confirmed our experimental observations. Basically, under boundary lubricated conditions (high load low speed), the edge stresses around the dimple increase frictional losses as if the surface is much rougher. So there is a trade-off in this regime.

Previously there had been no success in achieving significant friction reduction under high load low speed conditions. Therefore we have to develop the fundamental theory and practice from scratch. The approach taken is to engage in simultaneous theoretically modeling and experimentations to advance our understanding and subsequently use the understanding to further our development of new texture patterns and features to achieve friction reductions.

This report focuses on our progress in surface texture characterization and model development based on literature survey and friction and lubrication theories.

2. Characterization of Surface Features

One biggest challenge in the surface texture development for effective friction reduction is to describe an individual feature or surface by a set of parameters which can be measured quantitatively and related to the tribological characteristics. The topography of surfaces is directly related to lubrication, friction, wear, and adhesion. The previous search of parameters for surface functional characterization had led to a ‘parameter rash’

[1], more than 100 parameters defined based on 2D stylus-type measurement. Although these existing parameters are more than sufficient for most applications, there are still cases in which new parameters are required. A particular example is the textured surface which consists of a wide range of statistical and deterministic features.

Conventional spacing parameters include

- Density of summits: the stability of the definition is still a matter of discussion
- Auto-correlation: parameters such as the isotropy index or the fastest decay auto-correlation length originate from the ACF
- Texture direction: directionality of surfaces has good correlation with functional properties – Information can be extracted either from the angular spectrum diagram which originates from the power spectral diagram (the extreme point is known as the Std parameter) or from vectorial analysis
- Geometrical: wherever there is a predominance of deterministic features in a surface, information can be retrieved using plain Euclidean geometry characterization

The spacing parameters are problematic because there are many topological problems that arise when extending the definitions from a 2D profile to an areal measurement. For textured surfaces, the definition of topography characterization parameters needs to follow the next three steps:

- 1) Identify the key parameters that directly influence friction, wear, and lubrication in a 3-D perspective;
- 2) Develop a group or a collection of parameters that can adequately represent various textured surfaces for comparison, quality control, performance identification;
- 3) Identify and separate the function-relevant structures and describe their functional behavior by a set of structural parameters.

Such parameters may be derived from spacing parameters recently developed for characterizing closed and open lubricating zones. One example is the spacing parameters

proposed by Sacerdotti et al based on peaks and valleys consisting of interconnected points [2]. Once the desired height of the horizontal intersecting plane is defined, a list of the closed regions above the intersecting plane (peaks) is built according to a chosen neighboring criterion (NC). The list is then pruned of closed regions that can be considered as noise. With the pruned list, the closed peak area (Spa) parameter is defined as its average area. The closed peak density (Spd) parameter is defined as the ratio between the overall area above the intersecting plane (bearing area) and Spa then divided by the total area. The same calculation process applied to the closed regions below the intersecting plane leads to the definition of the average valley area (Sva) and valley density parameter (Svd).

The approach above is refined as the closed region curve (CRC). The surface is dissected at N evenly spaced heights from the maximum to the minimum value. At each height h_k , closed peak average area, edge peaks area (EPA), bearing area, edge valley area (EVA) and closed valley average area are plotted as a percentage. A variety of experiments were devised and performed to correlate the defined surface parameters to pressing performance of the panels. The results showed the importance of closed region parameters when characterizing the lubricant migration. The migration phenomenon appears to be driven by the size of the closed valleys, which can be described by the Sva parameter.

In comparison, Pfestorf et al defined another set of parameters: material area ratio, open void area ratio, and closed void area ratio from a mechanical-rheological model [3]. The closed void volume plays a decisive part in characterizing the tribological behavior in sheet metal-forming. Hydrostatic pressure is built up within the lubricant pockets. The reduction of the closed void volume must be connected with an increase of the hydrostatic pressure. In addition, micro hydrodynamic mechanism arises as the lubricant will be squeezed into the solid contact area building up microfilm lubrication. There is an optimum value for optimum friction conditions. In case the closed void volume is low, there is not enough lubricant to build up the micro film. On the other hand, if the closed void volume is very high, the hydrostatic pressure will be too low, not capable to produce the microfilm.

Although the above areal three dimensional (3D) measurements and parameters show potential for describing functional characteristics, they are far from being recognized by International Standards.

3. Model Development

3.1 Understanding the effects of micro-wedges formed by individual dimples on the overall behavior based on the analytical solution of the 1-D Reynolds equation

In this section, the contribution from individual dimples is estimated and role of these micro-wedges in determining the overall behavior is then examined. Assuming the micro-wedge effect is the only lift mechanism, a dimple illustrated by Fig. A1 can be approximately modeled as a tilting pad bearing shown in Fig. 1. In the case of infinite length (in the direction normal to the page), the hydrodynamic load capacity per unit length of the pad bearing is given by [4]

$$W_H = 6U\eta \left(\frac{L}{h_2 - h_1} \right)^2 \left[\ln \frac{h_2}{h_1} - \frac{2(h_2 - h_1)}{h_2 + h_1} \right] \quad (1)$$

where U is the sliding velocity and η is the lubricant viscosity. Equation (1) is determined from the analytical solution of the Reynolds equation and thus requires the following assumptions

- 1) The thickness of the fluid film is much smaller than the characteristic lengths in the sliding directions, $h_2 \ll L$. Then the curvature of film can be ignored;
- 2) Newtonian lubricant film undergoing laminar, incompressible flow;
- 3) There is negligible pressure variation across the film thickness;
- 4) No external forces act on the fluid film;
- 5) There is no slip at the film boundary;
- 6) Temperature, fluid inertial and squeeze film effects can be neglected.

Figure 1. An one dimensional tilting pad bearing.

Because of the finite length, the hydrodynamic load capacity of single dimple can be determined from the solution of the tilting pad bearing by introducing a correction factor to account for the side leakage of lubricant. The resulting

$$W_{H,dimple} = 6CU\eta B \left(\frac{L}{D} \right)^2 \left(\ln \frac{D+h}{h} - \frac{2D}{D+2h} \right) \quad (2)$$

where C is the correction factor and h is the separation between the ball and the flat sample. In the mixed or lubrication regime, h can be related to the composite surface roughness σ_{cmp} as

$$h = \lambda \sigma_{cmp} \quad (3)$$

with the value of λ ranging from 1 to 3.

As for the texture patterns listed in table 1, the values of factor C should be similar since all the dimples have similar shapes. Equation (2) may thus be used to compare the contribution of a single dimple of different patterns. The total hydrodynamic lift from micro-wedge effects can also be determined by summing up the contribution from individual dimples. Under a load of 20 kg, the lift is calculated for the three patterns with 106 dimples in the nominal contact zone by taking the value of λ to be 2 and Figure 2 plots the friction coefficient against the calculation results. The figure shows a weak correlation but them but significant deviation exists. The deviation may be caused by the following:

- 1) For individual dimples, assumption 1 for the Reynolds equation is no longer valid because the dimple depth is in the same order of magnitude as the dimple length, especially for deeper dimples.
- 2) The overall effects of micro surface dimples include the change of surface gap. As a result, the estimation based on the same gap shown in Fig. 1 is not accurate.
- 3) The micro-wedge effects may not be the only lift mechanism. The friction reduction is a result of multiple mechanisms.

Figure 2. Friction coefficient vs. total hydrodynamic lift from micro-wedge effects for patterns C2-400-2, C4-400, and C4-800 in table A1.

In summary, the analysis above demonstrates the potential to explain the micro-wedge effects of surface dimples based on the solution of the Reynolds equation. An insightful understanding of the experimental results may be obtained from a more accurate solution of individual dimples and a more careful consideration of their overall effects.

3.2 Orientation Effects and Size Limits of Micro-Wedges Formed by Individual Dimples

The experimental results show that the friction measurement is sensitive to the sliding direction with respect to the dimple orientation. In comparison, the orientation effects of micro dimples on friction and lubrication are also revealed by a numerical analysis of the lubricant flow across a surface cell containing a single dimple. This section first presents a simple model problem and a set of analysis results obtained and then explores the possibility to determine the size limits of micro-wedge effects using a similar numerical approach.

Figure 3. Model problem: a surface cell containing a single dimple.

As shown in Fig. 3 the model problem uses a rectangular dimple consisting of a pair of converging/diverging wedges to simulate the micro-wedge effects. The hydrodynamic pressure distribution is determined by solving the Reynolds numerically.

For very deep dimples or those with non-smooth edge, the first three simplifications in deriving the Reynolds equation may not apply because of too much disruption caused by them to the laminar flow. Computational fluid dynamics (CFD) codes have therefore been used to solve the Navier-Stokes equations directly [5-7]. The CFD simulation results are significantly different from the solution of the Reynolds equation when the Reynolds number defined below are larger than 40 [6].

$$\text{Re} = \frac{\rho U h_2}{\mu} = \frac{\text{Inertial effects}}{\text{Viscous effects}} \quad (4)$$

where ρ is the fluid density and h_2 the dimple depth. For the textured samples examined at the highest speed in our ball-on-three-flats tests, the corresponding Reynolds number is much smaller than the critical value suggesting that the orientation effects and other dimple hydrodynamic behaviors can be captured by solving the Reynolds equation.

Figure 4. Numerical solution of the Reynolds equation – dimensionless pressure distribution over the surface cell: sliding in the positive x direction.

Figure 4 shows the dimensionless pressure distribution over the surface cell under the condition of $a_1 : a_2 = 1 : 2$, $h_2 : a_1 = 1 : 10$, and $h_1 : h_2 = 1 : 100$ in which h_1 is the nominal film thickness. The hydrodynamic lift due to the micro-wedge effects is determined by integrating the pressure distribution over the surface cell. The lift is nearly doubled by reversing the sliding direction demonstrating a strong orientation effect. More specifically, the dimple performs better when the converging wedge is shorter than the diverging wedge. This is consistent with the experimental observation.

3.3 Determination of Estimation of individual friction components

The Reynolds equation-based models do not account for all the lifting mechanisms induced by surface textures during the lubricated contact. A new approach is to quantify the frictional resistance at selected experimental data points instead. This friction resistance-focused approach is developed by considering the balance of tangential forces. For untextured samples, the corresponding force balance equation is given by

$$F = F_{\text{film viscous shear}} + F_{\text{roughness}} + F_{\text{sliding friction}} + F_{\text{plowing friction}} \quad (5)$$

In this equation, the last three components are from “true” asperity contacts and associated with deformation, plowing, and tribo-chemical reactions between fresh metal, metal oxide, boundary additives. For textured samples, there might be additional friction resistance due to the stress concentration at dimple edges. The resulting force balance equation is given by

$$F = F_{\text{roughness}} + F_{\text{film viscous shear}} + F_{\text{sliding friction}} + F_{\text{plowing friction}} + F_{\text{edge stress}} \quad (6)$$

There are no simple solution to the above two equations because they contain more than one unknowns. However, the net friction reduction or lift force by the textures can be determined at the data points selected if we can estimate most of the frictional resistance components. The determination will be an iteration process to ensure the validity of the assumptions with which the friction resistance components are estimated. The results and the concept of controlled experiments can then be used to determine the friction reduction from cavitation, hydrodynamic and hydrostatic effects, and other factors. With the results of the contribution from individual mechanisms, the friction reduction of new texture designs can be predicted.

This section presents individual modules developed for estimating frictional components in Eqs (5) and (6). Since the friction resistance related to roughness and interfacial sliding are proportional to the real area of contact and the plowing friction depends on the degree of plastic deformation, the contact solution module is described first.

3.3.1 Determination of the real contact solution

With the digital topography, the real area of contact and the fraction in plastic deformation are determined by solving the following numerical contact equations [8].

$$u_{ij} = \sum_l \sum_k K_{ik,jl} p_{kl}, \quad (7a)$$

$$\sum_i \sum_j p_{ij} = P_0, \quad (7b)$$

$$g_{ij} = h_{ij} + u_{ij} - u_0, \quad (7c)$$

where u_{ij} are the surface displacements at grid points (i, j) under the normal applied force F_0 , p_{kl} are the contact pressures acting over elements of $\Delta x \times \Delta y$ centering at grid points (k, l) , u_0 is the rigid normal approach of one body to the other, P_0 is defined by $F_0 = \Delta x \Delta y P_0$, and h_{ij} are the gaps between surfaces at individual grid points just prior to any deformation and g_{ij} the gaps after the deformation. In Eq. (7a), the surface displacements are related to contact pressures via the influence coefficient $K_{ik,jl}$ which is defined as the normal displacement at the grid point (i, j) due to a unit pressure acting on the element which centers at (k, l) . The value of $K_{ik,jl}$ is calculated by integrating over the element on which p_{kl} is acted the Boussinesq relationship between the surface normal surface deflection and the concentrated point force, i.e.

$$K_{ik,jl} = \frac{1}{\pi E'} \int_{y_l - \Delta y/2}^{y_l + \Delta y/2} \int_{x_k - \Delta x/2}^{x_k + \Delta x/2} \frac{dx dy}{\sqrt{(x_i - x)^2 + (y_j - y)^2}}, \quad (8)$$

where E' is the reduced elastic modulus.

3.3.1 Effects of surface roughness

Both our measurement with the ball-on-three-flats tests and the simulation results of a mixed lubrication model [9] show that the friction and lubrication behavior of surfaces strongly depends on the surface roughness. However, the surface roughness parameter, the root mean square (RMS) of surface heights, is not the only controlling factor. The microgeometry characteristics resulting from specific surface finish methods, such as

orientation and wavelengths, are also critical. For the textured surfaces, the roughness effects become more complicated. The friction resistance from roughness of a textured surface consists of the contribution from original roughness and that due to the stress concentration of dimple edges. Considering that the roughness contribution is proportional to the apparent contact area, the force balance equation of a textured surface can be simplified as

$$\text{Total Friction} = \text{Friction from roughness of the untextured surface} \times (1 - \text{fractional area coverage ratio}) + \sum \text{friction from individual texture features in the contact} + \sum \text{negative friction reduction} + \text{Viscous shear resistance} \quad (9)$$

With this equation, the net friction reduction of a texture pattern can be determined once the roughness contribution of the corresponding untextured surface, the additional resistance due to dimple effects, and the viscous shear resistance can be estimated.

3.3.1 Plowing friction

The plowing friction resistance between two surfaces is estimated by considering their topographic conformity achieved at the steady state. Take a ground surface (sample C0-1 in table A2.1) as an example. Figure 5 shows the topography of a fractional surface of this sample. The plowing friction may come from the plowing at grooves shown in the figure and also asperity contacts. The contribution of these two types of plowing is related to the energy required to remove material to form the grooves and valleys. For a single groove shown in Figure 6, the volume of the material volume removed can be estimated as

$$\text{Volume of removal} = \text{Average cross-sectional area} \times \text{Sliding distance} \quad (10)$$

For valleys formed due to asperity plowing, the volume of material removed can also be estimated from the digital topography. The energy consumed during the removal of material is given by

$$\text{Energy consumed for plowing} = \text{Volume of removal} \times \text{Material hardness} \quad (11)$$

The corresponding resistance is then given by

$$\text{Plowing friction} = \text{Energy consumed for plowing} / \text{total sliding distance} \quad (12)$$

Figure 5. The topography of a fractional surface of a ground sample (C0-1 in table A1).

Figure 6. Formation of a single groove due to plowing.

3.3.1 Viscous shear resistance

The friction resistance from viscous shear can be approximated as

$$F_v \approx (A_n - A_r) \bar{\tau}, \quad (13)$$

where A_n is the nominal area of contact, A_r is the real area of contact, and $\bar{\tau}$ is the average viscous shear stress and is given by

$$\bar{\tau} = \eta_{\text{eff}} \left[\frac{\partial V}{\partial z} \right]_{\text{average}} \approx \eta_{\text{eff}} \frac{V}{\bar{h}}, \quad (14)$$

where V is the sliding velocity, $\bar{h}$ is the average film thickness, and η_{eff} is the effective viscosity and is estimated by using the following viscosity-pressure relation

$$\eta_{\text{eff}} = \eta_0 e^{\alpha p}, \quad (15)$$

where η_0 is the viscosity under the ambient pressure, α the pressure-viscosity coefficient, and $\bar{p}$ the average hydrodynamic pressure.

Figure 7. Surface contact deformation and film thickness.

With the numerical contact solution determined under a certain assumption on the load support, as shown in Fig. 7, the average film thickness can be estimated as

$$\bar{h} = \bar{s} + \bar{u} - u_{\max}, \quad (16)$$

where $\bar{s}$ is the average separation before the contact, $\bar{u}$ the average surface contact displacement, and $u_{\max}$ the maximum contact displacement.

In addition, the average hydrodynamic pressure $\bar{p}$ is approximated as

$$\bar{p} = \frac{\text{Load}_{\text{supported by hydrodynamic pressure}}}{(A_n - A_r)}, \quad (17)$$

With the estimation of $\bar{h}$ and $\bar{p}$ based on different assumptions on the load support and the corresponding numerical contact solution, the viscous shear friction resistance can be calculated by using Eq. (13) along with Eqs (14-15).

3.3.1 Additional resistance from dimple edge effects

Figure 8. Finite element simulation results of the stress distribution around a dimple under normal contact.

The introduction of dimples on a surface may also cause additional friction resistance due the dimple edge effects: including both the resistance resulting from the normal stress concentration and that due to the sliding. For a given dimple, the finite element simulation of the normal contact can be conducted to determine the stress distribution at the dimple as shown in Fig. 8. The results can be used to construct equivalent positive roughness at the same position of the dimple and the friction resistance due to the edge effects can be estimated based on the experimental data of ground and polished surfaces. On the other hand, the finite element simulation can also be carried out for the sliding contact involving the dimple for a given interfacial sliding friction coefficient. The additional friction force can be determined by averaging the tangential force over a single pass. Figure 9 shows the typical stress distribution around the dimple during the sliding.

Figure 9. Finite element simulation results of the stress distribution around a dimple during sliding contact.

3.3.1 Interfacial sliding resistance

In addition to the resistance due to viscous shear, plowing friction, and edge effects, the roughness and sliding friction terms in Eqs (5) and (6) can be estimated based on the real area of contact and the corresponding coefficient selected from a range determined in experimental measurement. Between them, the sliding friction is related to tribochemical reactions between fresh metal, metal oxide, and boundary additives, and thus cannot be to model directly. However, the coefficient of interfacial sliding may be in the range of 0.08-0.12 according to experimental data [10, 11]. Similarly, the friction coefficient corresponding to $F_{\text{roughness}}$ is around 0.03-0.05, according to the experimental results of a series of indentation tests [12]. In estimating the friction resistance components for the data points selected from the experimental database available, the values of these two coefficients will be specified to ensure the convergence of the results.

4. Summary

The load and velocity-dependence revealed by the correlation analysis indicates that textured samples most likely work in the mixed lubrication regime. In addition, the analysis based on the solution of the Reynolds equation is used to understand the role of individual dimples in determining the sample lubrication behavior. Models with a geometry configuration more similar to the real dimples designed may be used to further determine the optimum L/D ratio and size limit of micro-wedge mechanism. Moreover, the roughness factor may be incorporated into the model by relating the nominal film thickness to surface roughness parameters and examining the change of the fractional load support from the hydrodynamic lift and asperity contact with the film thickness.

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Appendix: Friction Measurement and Texture Patterns

Figure A1 shows the ball-on-three-flats modification of a 4-ball wear tester, which is used to simulate the sliding under high contact pressure conditions. The sliding is generated as the ball (standard 52100 bearing steel, 12.48 mm in diameter) rotates against the three flats (hardened 52100 steel, diameter of 6.24 mm in diameter and 1.59 mm in thickness). In order to avoid wear-in at the start of the experiment, the flat samples were pre-worn in with the tester for 30 minutes. Dimples were then produced on the surfaces of the flats by nanomechanical scratching. Finally, the friction tests were carried out for the untextured and textured samples under various load and speed conditions. In these tests, pure paraffin white oil (Fischer Scientific 25.1cst at 40°C) with 2% anti-wear and 1% anti-oxidation additives was used as the lubricant to avoid severe wear. This procedure has been used to generate experimental results for different groups of textured samples to reveal the effects of texture size, depth, and other parameters on the friction behavior.

Figure A1. Ball-on-three-flats contact geometry of a 4-ball wear tester

From the experimental database available, the friction measurements of 9 sets textured and untextured flat samples have been examined to develop the friction prediction model. The dimples created on 7 sets of textured samples with built-in micro-wedges. The 7 sets of textured samples are obtained by changing the dimple size, area ratio, and number density. Table A1 shows the details of these texture patterns, which are designed to determine the factors critical to the friction behavior. The textured samples were tested with the four-ball wear tester under loads of 5, 10, 20, and 30 kg. The resulting nominal contact pressures were 192.3, 242.3, and 277.4 MPa, respectively, calculated from Hertzian contact equations between the ball and a flat. At each load, the test sequence

was: (1) 500 rpm (0.2m/s) for 5 minutes; (2) 1 minute for friction measurement; (3) increasing the rotational speed and measuring the friction at 1000 rpm (0.38 m/s), 1500 rpm (0.57 m/s), 2000 rpm (0.77 m/s), 2500 rpm (0.96 m/s), 3000rpm (1.15 m/s), and 3500rpm (1.34 m/s) with the same time intervals. In addition, friction was measured for a set of polished samples to serve as the baseline of comparison and also for a set of unpolished or ground samples to demonstrate the effects of surface roughness. Table 2 lists the roughness parameters of these two sets of untextured samples. Within this range of contact pressure and speed, the bearing steel-on-bearing steel system is mostly in mixed lubrication where the normal load is shared by the oil film and asperity contacts. In the most severe situations (high load and low speed), wear occurred on sample surfaces suggesting that the system is under boundary lubrication and almost all the load is supported by asperity contacts. In these two regimes, our previous tests with the system showed that the friction would increase due to the introduction of surface texture patterns which was proved effective in reducing friction under low load-high speed conditions. The results of a sophisticated elastohydrodynamic friction model developed by Northwestern University confirmed the experimental observations. The increase of friction may be primarily due to the high edge stresses around dimples and the negligible hydrodynamic effects at low speed.

Table A1. Parameters of 7 texture patterns examined for the model development.

Texture pattern	Number of dimples in contact	Dimple length (μm)	Dimple width (μm)	Dimple depth (μm)	Ratio of area coverage (%)	L/D ratio
C1-200	426 $\pm$ 30	9.81 $\pm$ 0.32	4.98 $\pm$ 0.10	0.61 $\pm$ 0.05	6.41 $\pm$ 0.34	10
C2-400	206 $\pm$ 15	14.91 $\pm$ 0.41	7.14 $\pm$ 0.17	1.04 $\pm$ 0.11	6.77 $\pm$ 0.35	10
C4-800	106 $\pm$ 8	22.31 $\pm$ 0.62	9.80 $\pm$ 0.18	1.86 $\pm$ 0.12	7.14 $\pm$ 0.33	9
C2-400-2	106 $\pm$ 8	14.72 $\pm$ 0.34	7.11 $\pm$ 0.12	1.05 $\pm$ 0.08	3.42 $\pm$ 0.14	10
C1-100	426 $\pm$ 30	8.95 $\pm$ 0.23	3.88 $\pm$ 0.08	0.38 $\pm$ 0.02	4.6 $\pm$ 0.21	15
C2-200	206 $\pm$ 15	13.75 $\pm$ 0.58	5.25 $\pm$ 0.08	0.61 $\pm$ 0.08	4.6 $\pm$ 0.26	15

C4-400	106±8	20.40±0.56	7.37±0.15	1.12±0.10	4.9±0.23	13
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Figure A2. Illustration of wedge effects in the hydrodynamic lubrication.

According to the hydrodynamic lubrication theory, as shown in Fig. A2, the presence of a geometric wedge between two surfaces causes the change of velocity gradient along the path when the lubricant is drawn through the convergent space [A1]. As a result, the hydrodynamic pressure is developed under the action of viscous force leading to the formation of a fluid film and its load-carrying capacity sufficient to separate the surfaces. The dimples examined were thus designed based on the consideration that each dimple may work as a microscopic bearing because of the presence of the converging/diverging wedges. Figure A3 shows the top-view and cross-sectional profiles of a dimple made this way. From experiments, we have identified texture patterns consisting of this type of dimples that are capable of providing significant friction reduction (40%) in the bearing steel-on-bearing steel system shown in Fig. A1. This implies a possible micro-wedge effect in the mixed and boundary lubrication regimes. One of the critical issues of this type of features is the effect of the slope at the bottom of the feature. In bearing applications, this typically is called the L/D ratio (bearing length over radius of the bearing) to estimate the hydrodynamic lift force. The dimple size and depth are also critical to the potential wedge effects and we are exploring the limit size above which the effects become significant. Another important factor is the number of surface features over the contact area. In boundary lubrication conditions, since the edges of the surface features serve as rough spots increasing the friction, therefore, the optimum number of

features might be much smaller than the high speed-low load regime. The optimum number of features is also related to the ratio of area covered by the features.

(a) Top view

(b) X-profile

Figure A3. Top view and 2D profiles of a dimple produced by nanomechanical scratching.

References

A1.Khonsari, M.M., and Booser, E.R., *Applied Tribology: Bearing Design and Lubrication*, John Wiley & Sons, New York, 2001.