CO-OPERATIVE PROGRAMME ON INTEGRATED ENGINEERED SURFACE TECHNOLOGY TO REDUCE FRICTION AND INCREASE DURABILITY

A Topical Report

INTERNATIONAL ENERGY AGENCY

IMPLEMENTING AGREEMENT FOR A CO-OPERATIVE PROGRAMME ON ADVANCED MATERIALS FOR TRANSPORTATION APPLICATIONS

The report is prepared by Annex IV leaders from USA, China, Australia, Israel.
An integrated surface technology design, modeling, and representation

A topical report

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EXECUTIVE SUMMARY

The demand for higher fuel economy of internal combustion engines to satisfy our transport needs are ever increasing due to limited availability of cheap oil. This coupled with the need to reduce carbon emissions has created a tremendous driver to develop fuel efficiency technologies to extend the oil reserve for the foreseeable future while the deployment of electric vehicles and plug-in hybrids are progressing slowly. In 2011, US enacted a new CAFÉ standard of 54.5mpg for cars and light trucks by 2025. This has provided an added impetus to accelerate fuel efficiency technology development.

This Annex was formed based on a simple concept that if surfaces can be designed and fabricated at a cost-effective way, then it may be possible to control the interfacial friction in engine components to a level that would significantly increase the fuel economy of internal combustion engines.

Surface texturing is not new. But the science and technology of it took off in the early 1960s in metal forming processes. In the 1990s, Izhak Etsion published a series of papers demonstrating that discrete dimples fabricated by laser ablation can reduce friction while improving durability in seals (low load, high speed conformal contacts) in both theoretical models and field tests.

The Annex combines the expertise and effort of four countries in harnessing this concept. During the course of this effort, each country contributes unique expertise and facility. At the end, we have developed a design guideline for extending the texturing into non-conformal contacts, developed lithographic techniques coupled with electrochemical etching of steel surfaces, and succeeded in demonstrating significant friction reduction on actual engine components such as ring and liner interface, cam and lifter interface, etc. The technology is being evaluated by major original equipment manufacturers (OEMs) for potential implementation.

This report is organized by chapters delving into model prediction, texture design, diamond like carbon thin films to protect the texture from wear, and bonded chemical film to further enhance the friction reduction capability as well as enhancing durability. There are two chapters on surface representation and optimization. This report represents the current state of the art understanding and practice of this technology.
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Within each country, the government is represented by the Department of Energy in the US, the Department of Natural Resources, Energy and Tourism in the Australia, the Chinese Academy of Sciences, and the Ministry of National Infrastructure in Israel. The technical leaders performing the research are: Dr. Stephen Hsu of George Washington University in the US, the Chair of this Annex; Dr. Gwidon Stachowiak of the Western Australia University, Australia; Dr. Junyan Zhang of Lanzhou Institute of Chemical Physics, Chinese Academy of Sciences, China; and Dr. Izhak Etsion of Technion, Israel. Within each country, many people are involved in this research. We wish to acknowledge the financial support of our sponsors and the many industrial companies providing parts, advice, and guidance.

The authors wish to acknowledge the efforts of many graduate students, Postdoctoral Fellows, and other collaborators in industry, government, and national laboratories in each country, without their contributions, this report would not be possible.
1. Introduction

Surface texturing as a means for enhancing tribological properties of mechanical components is well known for many years. Perhaps the most familiar and earliest commercial application of surface texturing is that of cylinder liner honing. Fundamental research work on various forms and shapes of surface texturing for tribological applications is carried out worldwide and various texturing techniques are employed in these studies. Of all the practical micro-surface patterning methods it seems that laser surface texturing (LST) offers one of the most promising concepts. This is because the laser is extremely fast and allows short processing times; it is clean to the environment and provides excellent control of the shape and size of the texture, which allows realization of optimum designs. Indeed, LST is gaining more and more attention in the Tribology community as is evident from the growing number of publications on this subject. LST produces a very large number of micro-dimples on the surface (see Fig. 1) and each of these micro-dimples can serve either as a micro-hydrodynamic bearing in cases of full or mixed lubrication, a micro-reservoir for lubricant in cases of starved lubrication conditions, or a micro-trap for wear debris in either lubricated or dry sliding.

Fig. 1. LST regular micro-surface structure in the form of circular micro-dimples each is having a diameter of the order 100 µm and a depth of the order of 1-10 µm.
LST started at Technion in Israel in 1996 [1, 2], at about the same time that laser surface texturing was started in Germany (most of the work in Germany was published in the German literature, hence not well known). Some papers from the Geiger’s group at the University of Erlangen-Nuremberg were published in English, e.g. [3, 4]. They used an eximer laser through a mask projecting on metal surfaces. This method was applied to a punch, used in a backward cup extrusion process for the production of rivets, and showed a substantial increase of up to 169% in cold forging tool life. These as well as many other papers on LST are described in a review of the state of the art of LST covering this subject until 2005 [5]. A recent review on laser surface texturing and applications covers the period through 2007 [6] and another recent review focuses on surface texturing for in-cylinder friction reduction [7].

Fig. 2. Number of papers on surface texturing published in English each year from 1996 through 2010.

The number of papers on surface texturing shows a dramatic growth over the last three years, as shown in Fig. 2. To date, surface texturing have been explored in many applications including; automotive, bearings and seals, elastohydrodynamic lubrication (EHD), magnetic storage etc. While the micro-hydrodynamic bearing function of textured features in cases of full lubrication has been extensively studied theoretically and experimentally, and is reasonably understood, application in the EHD low film thickness regime, and boundary lubrication regime (BL) (micro-reservoirs for lubricant under starved lubrication conditions, and micro-traps for wear debris) are still far from completion and much research work is needed in these areas.

In the following sections, the current state of the art in surface texturing will be described focusing mainly on modeling in hydrodynamic and hydrostatic lubrication applications.

2. Background
Surface texturing is a powerful means of enhancing hydrodynamic lubrication between parallel surfaces in relative sliding, which otherwise when un-textured cannot provide any significant hydrodynamic load carrying capacity. This is demonstrated in Fig. 3 showing two parallel surfaces, the lower one moving at a relative sliding velocity $U$ with respect to the upper surface.

![Fig. 3. Schematic description of parallel sliding surfaces: a single protrusion (a), and the hydrodynamic pressure distribution over the single protrusion (b).](image)

The surface texturing is represented in Fig. 3(a) by a single protruding asperity attached to the upper surface. The ambient pressure surrounding the two surfaces is $P_a$. In the absence of any texturing the relative sliding velocity results in viscous shear only without any effect on the pressure between the two flat surfaces and, hence, zero load carrying capacity $F=0$. The introduction of the protruding asperity changes the local film thickness into a converging diverging one, which generates a hydrodynamic pressure distribution as shown in Fig. 3(b). Due to the relative velocity $U$ the pressure increases above $P_a$ in the converging portion and decreases below $P_a$ in the diverging portion of the clearance. At low velocities the maximum pressure is smaller than the absolute value of the cavitation pressure $P_c$ and the pressure distributions is anti-symmetric about $P_a$. Integrating the pressure distribution along its axial span results in zero load capacity since the above and below ambient pressures cancel each other. As the velocity $U$ increases the pressure distribution becomes asymmetric about $P_a$ since the minimum pressure value is bounded from below by the cavitation pressure $P_c$ while the maximum pressure is not limited. The integration of the pressure now results in a net positive value $F > 0$. When the textured surface contains a large number of protruding asperities the total load carrying capacity is the sum of their individual contributions. Exactly the same effect with an asymmetric pressure distribution as shown in Fig. 3(b) can be obtained with indented dimples instead of protruding asperities, only in this case the diverging portion of the clearance precedes the converging one. It should be noted here that because of the micro-scale of the asperities or dimples (see Fig. 1) the total load capacity of textured parallel surfaces is relatively small compared to the load capacity that can be generated in conventional hydrodynamic slider bearings. Hence, surface texturing is mostly...
beneficial in cases where conformal mating surfaces at very small uniform clearances are required as, for example, in various sealing applications. Also, as shown in Fig. 4 the dimples configuration is a better choice for surface texturing compared to that of protrusions.

![Protrusions vs Dimples](image)

- Complicated etching technology
- High wear
- High leakage (seals)

- Simple & cheap laser technology
- Low wear
- Low leakage/spacing

**Fig. 4. Advantages of a dimples configuration in surface texturing compared to that of protrusions.**

This is mainly because when the surfaces are brought into contact the real contact area with dimples is much larger than that with protrusions. Hence, the average contact pressure in the dimples case is much smaller and the wear is much lower. Another advantage of the dimples configuration is the smaller separation that can be obtained between the mating surfaces, which allows better sealing and much smaller leakage. Indeed, laser surface texturing has been proven extremely beneficial in various sealing applications for liquids and gas, see e.g., [8-12]. In some of these applications where high pressure liquid has to be sealed, such high pressure may eliminate the cavitation in the individual dimples and thus hamper the generation of hydrodynamic load carrying capacity. The solution for this problem is a partial surface texturing adjacent to the high pressure side as shown, for example, in Fig. 5. When the clearance between two parallel surfaces has a step change (see Fig. 5(a)) and the larger clearance $h_{\text{max}}$ is facing the higher pressure $p_0$, a hydrostatic load carrying capacity can be generated due to the restriction in the Poiseuille flow caused by the smaller clearance $c$ facing the lower pressure $p_a$. A similar effect is obtained by "partial" surface texturing where high density dimples at the high pressure end form an equivalent "step" of height $h_{\text{eq}}$ as shown in Fig. 5(b). This solution has been used successfully in high pressure seals as described in [13].

![Fig. 5](image)

**Fig. 5. A comparison of two equivalent sets of parallel surfaces: a step change in the clearance (a), and partial surface texturing (b).**

Both the original "full" texturing and its modification form of "partial" texturing, as well as textured micro-grooves, were successfully applied to piston rings and cylinder liners [14-17]. The partial texturing concept was also found very useful in generating substantial
hydrodynamic load capacity in hydrodynamic slider bearings and thrust bearings of the simplest form of parallel sliding disks [18,19].

![Cross Section of Partial Laser Surface Textured Parallel Slider Bearing](image)

Fig. 6a. A cross section of a partially laser surface textured parallel slider bearing.

Here the effect of the equivalent step is similar to the very efficient Raleigh step in a slider bearing. Through extensive theoretical modeling it was found that the most important parameter in surface texturing for full fluid film lubrication is the aspect ratio of the dimples (depth over diameter ratio). It is this parameter that can be optimized in order to provide maximum load carrying capacity, maximum film stiffness and minimum friction coefficient. The area density, $S_p$, of the texturing also affects the efficiency of surface texturing and this parameter too can be optimized. Yet another important parameter that can be optimized in cases of partial texturing is the textured portion ratio $\alpha=B_p/B$, see Figs. 6a and 6b.

![Graph of Textured Portion Effect](image)

Fig. 6b. The effect of the textured portion, $\alpha=B_p/B$, on the dimensionless load carrying capacity of an infinitely long parallel slider at various dimensionless dimple depths, $H_p=h_p/h_0$ (see Fig. 6a)

3. Basic Modeling for Different Applications
3.1. Mechanical Seals

We shall use a typical textured seal application [8] to demonstrate the basic modeling of surfaces texturing for mechanical seals. The geometrical model of the textured surface is displayed in Figs. 7 and 8.

*Fig. 7. Schematic description of a textured seal ring: (a) dimples distribution, (b) a single dimples column with its coordinate system and boundary conditions, (c) individual dimple cell.*
Each dimple is modeled by an axi-symmetric spherical segment with a base radius $r_p$, and depth $h_p$ (see Fig. 8). The dimples are distributed uniformly over the annular surface with an area density $S_p$. Each dimple is located in the center of an imaginary square cell of sides $2r_1 \times 2r_1$ (see Fig. 7c) where:

$$r_i = \frac{r_p}{2} \sqrt{\frac{\pi}{S_p}}$$

(1)

It is assumed that the clearance between the nominally parallel mating surfaces (see Fig. 8) is fully filled with an incompressible viscous (Newtonian) fluid having a constant viscosity $\mu$. The ratio $R_i$ between the inner and outer radii, $r_i$ and $r_o$, of the seal ring under consideration is larger than 0.7. This allows to neglect curvature effects and consequently, a circular sector containing one dimples column in the radial direction (see Fig. 7a, and b) is assumed to be rectangular, subjected in the lateral $x$ direction to a relative sliding velocity $U$, corresponding to the tangential velocity at the mean radius of the seal ring.

The two-dimensional, steady state form of the Reynolds equation for an incompressible Newtonian fluid in a laminar flow is given by

$$\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( h^3 \frac{\partial p}{\partial z} \right) = 6\mu U \frac{\partial h}{\partial x}$$

(2)

where $z$ and $x$ are the radial and circumferential directions Cartesian coordinates, respectively, $h$ and $p$ are the local film thickness and pressure, respectively, at a specific point of the seal. In order to reduce Eq. (2) to a dimensionless form the dimensionless Cartesian coordinates $X$ and $Z$, dimensionless local film thickness $H$ and dimensionless pressure $P$ are defined as:

$$X = \frac{x}{r_p}; \quad Z = \frac{z}{r_p}; \quad H = \frac{h}{C}; \quad P = \frac{p}{p_a}$$

(3)

where $p_a$ is the ambient pressure and $C$ is the seal clearance. After substitution of Eqs. (3) into Eq. (2) the Reynolds equation in its dimensionless form becomes:
\[ \frac{\partial}{\partial X} \left( H^3 \frac{\partial P}{\partial X} \right) + \frac{\partial}{\partial Z} \left( H^3 \frac{\partial P}{\partial Z} \right) = \Lambda \frac{\partial H}{\partial X} \]  \hspace{1cm} (4)

where the dimensionless parameter \( \Lambda \) is given by

\[ \Lambda = \frac{6 \mu U r_p}{p_a C^2} \]  \hspace{1cm} (5)

The dimensionless local film thickness \( H \) over one imaginary dimple cell as a function of the local dimensionless Cartesian coordinates \( X \) and \( Z \), measured from the center of the imaginary dimple cell is given by:

\[
H(X, Z) = \begin{cases} 
1 & \text{for } X^2 + Z^2 > 1 \\
1 + \left( \frac{\varepsilon}{2\delta} + \frac{1}{8\delta^2} \right)^2 - \left( X^2 + Z^2 \right) \left( \frac{1}{4\delta^2} - \frac{1}{8\delta^2} \right) & \text{for } X^2 + Z^2 < 1 
\end{cases} \]  \hspace{1cm} (6)

where \( \varepsilon = h_p / (2r_p) \) is the dimple's aspect ratio, \( \delta \) is the dimensionless seal clearance \( C / (2r_p) \), and the axes of \( X \) and \( Z \) (not shown in Fig. 7) are parallel to these of \( x \) and \( z \), respectively.

By specifying \( \varepsilon \) and \( \delta \) and hence, the film thickness distribution \( H(x,z) \), the dimensionless parameter \( \Lambda \), and the relevant boundary conditions, Eq. (4) can be solved for the pressure distribution in the seal clearance. Integrating the pressure over the seal area gives the opening force acting in the axial direction to prevent contact between the rings for reliable operation of the mechanical seal.

Since the micro-dimples are evenly distributed it is assumed that the pressure distribution is periodic in the circumferential direction with a period equal to the imaginary square cell size \( 2r_j \). Because of this it is sufficient to consider only one radial dimples column as shown in Fig. 7b, with the following boundary conditions:

\[ p(x, z = r_1) = p_{in}; \quad p(x, z = r_o) = p_{out} \]  \hspace{1cm} (7)

Dimensionless values of the pressure at the two edges of one dimples column are obtained by normalization according to Eqs. (3). Periodicity condition of the pressure is applied in the circumferential direction so that

\[ p(x = -r_1, z) = p(x = r_1, z) \]  \hspace{1cm} (8)

The boundary conditions in dimensionless form are given as follows:

\[ P \left( X = \frac{r_1}{r_p}, Z \right) = P_{in}; \quad P \left( X = \frac{r_o}{r_p}, Z \right) = P_{out}; \]  \hspace{1cm} (9)
The boundary conditions at the inner and outer radii of the seal, Eq. (7), and the periodicity condition, Eq. (8), should be complemented by the conditions at the boundaries of possible cavitation regions associated with each individual dimple. As explained in the Background section above, these cavitation regions are responsible for the asymmetric hydrodynamic pressure distribution and, hence, are the only source for load carrying capacity in parallel surface sliding. The relatively simple Reynolds boundary condition, also known as the Swift Stieber condition, (see, for example, Ref. [20]) or any other more advanced cavitation boundary conditions, (e.g., [21]) can be assumed. The Reynolds condition implies that on the cavitation boundary the pressure gradient normal to the boundary is zero and the pressure inside the cavitation region is retained constant close to zero.

The Reynolds equation, Eq. (4), with its appropriate boundary conditions can be solved by a finite difference method using a non-uniform grid over the radial dimples column, shown in Fig. 7b, where a denser grid is applied within the dimple areas. Numerical tests show that, for dimple density values $S_p$ in the range between 10% to 50%, the best accuracy of pressure calculation is obtained when the grid applied in the area of the dimples is about five times denser than that outside of the dimples.

The finite difference method leads to a set of linear algebraic equations for the nodal values of the pressure which should be solved with the boundary conditions at the inner and outer radii of the seal, Eq. (7), and the periodicity condition, Eq. (8). This linear equations set may be solved by various standard methods. The successive over-relaxation Gauss-Seidel iterative method [22] is one possible method. It requires an initial approximation of the solution and, in the case of a mechanical seal, for example, a linear hydrostatic pressure distribution may be used for this purpose if a more precise solution is unknown. Although the iteration algorithm of Gauss-Seidel is not always the most effective one, this method is convenient for the determination of the previously unknown cavitation regions.

The basic modeling described above for mechanical seals can be easily modified to fit different applications. These modifications include, for example, solving a non-linear Reynolds equation for compressible fluids [11], solving the Reynolds equation simultaneously with a dynamic equation for piston rings [14] or with the equation of elasticity for elastomeric seals and bearings [23,24]. In the following few of these modifications are presented in more details.

3.2. Gas Seals

The non-linear Reynolds equation for compressible fluid applications such as gas seals [11] or gas bearings has the form:

\[
\frac{\partial}{\partial x} \left( ph^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( ph^3 \frac{\partial p}{\partial z} \right) = 6\mu U \frac{\partial}{\partial x} (ph) \tag{10}
\]

After substitution of Eqs. (2) into Eq. (10) the Reynolds equation in its dimensionless form becomes:

\[
\frac{\partial}{\partial X} \left( PH^3 \frac{\partial P}{\partial X} \right) + \frac{\partial}{\partial Z} \left( PH^3 \frac{\partial P}{\partial Z} \right) = \frac{\lambda}{\delta^2} \frac{\partial}{\partial X} (PH) \tag{11}
\]
where $\lambda = 3\mu U / 2r_p P_a$ is the dimensionless seal parameter and $\delta = c / 2r_p$ is the dimensionless seal clearance.

A Finite Element Method using a non-uniform grid over the radial dimples' column, shown in Fig. 7b, can be used to solve the Reynolds equation, (11), with its appropriate boundary conditions, where a dense grid is applied within the dimple areas. Note, that for the case of compressible fluids there is no cavitation to deal with, which somewhat simplifies the boundary conditions. A variational Galerkin formulation can be utilized in order to apply the Finite Element Method to the Reynolds differential equation [25]

$$\int_{\mathcal{A}} \left[ \frac{\partial}{\partial X} \left( PH^3 \frac{\partial P}{\partial X} \right) + \frac{\partial}{\partial Z} \left( PH^3 \frac{\partial P}{\partial Z} \right) - \frac{\lambda}{\delta^2} \frac{\partial}{\partial X} H \right] N_i(X) N_j(Z) \, dX \, dZ = 0 \tag{12}$$

where $A$ is the area of one dimples' column, $N_i(X)$ and $N_j(Z)$ are Lagrange polynomials and all their possible products $N_i(X) N_j(Z)$ are the weight functions. The solution $P(X, Z)$ is approximated by the double series

$$P(X, Z) = \sum_{i,j} P_{ij} N_i(X) N_j(Z) \tag{13}$$

where $P_{ij}$ are the approximate pressure values at the nodal points of the finite element grid. Note that the Lagrange polynomials used for the approximation of the pressure distribution in the seal are the same as these of the shape functions. The Finite Element Method leads to a set of nonlinear algebraic equations for the nodal values of the pressure that should be solved with the boundary conditions. This set of equations can be solved by the Newton gradient method. It requires an initial approximation of the solution, which can be a linear hydrostatic pressure distribution if a more precise solution is unknown. Although the convergence of the Newton gradient method is sensitive to the initial approximation, the convergence rate, in the case of successful initial approximation, is very high.

### 3.3. Parallel Thrust Bearings

A schematic representation of a parallel thrust bearing [18] is shown Fig. 9.
A plain disk (D) is rotating relative to a number of identical stationary pads (P). Each pad, when properly textured, develops the same hydrodynamic force. Hence, in order to evaluate the load carrying capacity of the complete parallel thrust bearing it is sufficient to determine the hydrodynamic pressure distribution over a single pad. A simplified geometrical model of a single pad in the form of a rectangular parallel slider is displayed in Fig. 6a. The dimples are regularly distributed over a portion, $0 \leq \alpha \leq 1$ (where $\alpha = \frac{B_p}{B}$), of the slider width, $B$, in the sliding direction, $x$, and over the full slider length, $L$, along the direction $z$ (not shown in Fig. 6a).

In order to reduce Eq. (2) to a dimensionless form, the dimensionless coordinates $X$ and $Z$, dimensionless local film thickness, $H$, and dimensionless local pressure, $P$, are defined as follows:

$$X = \frac{x}{r_p}; \quad Z = \frac{z}{r_p}; \quad H = \frac{h}{h_0}; \quad P = \frac{1}{\Lambda} \left( \frac{P}{p_a} - 1 \right)$$

(14)

where $p_a$ is the ambient pressure, $h_0$ is the bearing clearance (see Fig. 6a) and $\Lambda$ is the dimensionless bearing number, given as:

$$\Lambda = \frac{3 \mu U}{2 r_p p_a}$$

(15)

Substituting Eqs. (14) and (15) into (2) yields the Reynolds equation in its dimensionless form:
\[
\frac{\partial}{\partial X} \left( H^3 \frac{\partial P}{\partial X} \right) + \frac{\partial}{\partial Z} \left( H^3 \frac{\partial P}{\partial Z} \right) = \frac{1}{\delta^2} \frac{\partial H}{\partial X}
\]  

(16)

where \( \delta \) is the dimensionless clearance, defined as: \( \delta = \frac{h_0}{2r_p} \). The pressure along the slider boundaries is equal to the ambient pressure \( p_a \) that by the definitions in Eqs. (14) corresponds to zero dimensionless pressure.

The analytical model is valid for all values of slider length, \( L \), and width, \( B \). However, if the slider is long enough in the \( z \) direction (normal to the sliding velocity), with a ratio \( L/B > 4 \), the end effects in this direction can be neglected. In this special case the pressure distribution is periodical in the \( z \) direction with a period equal to the imaginary cell size \( 2r_1 \). Hence, because of this periodicity, it is sufficient to consider a single column of dimples along the \( x \) direction. Due to symmetry of the dimples column about its \( x \) axis, the pressure distribution will be also symmetric about this axis. Therefore, for the complete pressure distribution it is sufficient to consider only one half of the dimples column with \( z \) varying from 0 to \( r_1 \). From the periodicity, symmetry and continuity of the pressure distribution, it follows that:

\[
\frac{\partial P}{\partial Z}(X,0) = \frac{\partial P}{\partial Z}(X,r_1) = 0
\]  

(17)

The dimensionless pressure obtained from a solution of the Reynolds equation (16) with its appropriate boundary conditions, for a finite or for an infinitely long slider is numerically integrated over the slider area yielding the dimensionless load carrying capacity \( \bar{W} \), which is related to the dimensional load carrying capacity \( W^* \) in the form:

\[
\bar{W} = \frac{2W^*}{3\mu Ur_p}
\]  

(18)

3.4. Piston rings

Fig. 10 shows a piston ring segment [26] with partial LST applied at the middle portion of the ring width.

![Fig. 10. A segment of partially textured piston ring.](image)
However, the textured portion can be located elsewhere as shown schematically in Fig. 11. Here $w^*$ is the piston ring width, $b_p$ is the axial length of the textured zone, $x$ is the axial direction of the cylinder liner and $z$ is the circumferential direction of the piston ring. In Fig. 11(a) the textured zone is symmetrically located at the center of the ring; in Fig. 11(b) it is located symmetrically at both ends of the ring and in Fig. 11(c) at an arbitrary distance $d$ from the ring center.

A detailed description of a partial central symmetrical texturing is presented in Fig. 12. The dimples are uniformly distributed, with an area density $S_p$, within the strip $b_p$ of the piston ring full width $w^*$. The textured zone is bounded with two un-textured strips of width $(w^* - b_p) / 2$ on each of its side. Because of the periodicity of the surface texturing in the circumferential direction $z$, and the symmetry in each axial dimple column of width $2r_i$ about its longitudinal axis, it is sufficient to consider the pressure distribution within just one half of one dimple column.

![Fig. 11. Different locations of the textured zone: (a) symmetrically in the center; (b) symmetrically at both ends; (c) arbitrarily at a distance d from the ring center [26]](image)

Assuming a slider-crank mechanism that drives the piston [14], the sliding velocity $U$ is time dependent and the clearance between ring and cylinder liner varies with time during each cycle of the ring reciprocal motion. Hence, a simultaneous solution of the Reynolds equation with squeeze effect and the equation of piston ring radial motion is required. These equations are given in the form:

$$\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( h^3 \frac{\partial p}{\partial z} \right) = 6\mu U \frac{\partial h}{\partial x} + 12\mu \frac{\partial h}{\partial t}$$

(19)
\[ \rho h_r \frac{\partial^2 c}{\partial t^2} = p_h - p_e \]  

(20)

Fig. 12. A geometrical model of central symmetrical textured surface of a piston ring [26]

where \( h \) is the instantaneous local film thickness at a specific point \((x, z)\) (see Fig. 13), \( p \) is the instantaneous local hydrodynamic pressure, \( \mu \) is the dynamic viscosity of the fluid, \( c(t) \) is the instantaneous nominal clearance, \( \rho \) and \( h_r \) are the piston ring density and height, respectively, \( p_e \) is the total external pressure on the ring consisting of gas pressure and piston ring elasticity, and \( p_h \) is the instantaneous average hydrodynamic pressure between the ring and liner.

A simultaneous solution of Eqs. (19) and (20) provides the time behavior of both the clearance and pressure between the piston ring and cylinder liner surfaces. The dimensionless form of the differential equations (19) and (20) is given by:

\[ \frac{\partial}{\partial X} \left( H^3 \frac{\partial P}{\partial X} \right) + \frac{\partial}{\partial Z} \left( H^3 \frac{\partial P}{\partial Z} \right) = 2\Lambda_1 \Phi(\tau) \frac{\partial H}{\partial X} + 12 \frac{\partial H}{\partial \tau} \]

(21)

\[ \Lambda_2 \frac{\partial^2 C}{\partial \tau^2} = p_h - p_e \]

(22)

where \( X \) and \( Z \) are the dimensionless Cartesian coordinates, \( H \) is the dimensionless instantaneous local film thickness, \( \Phi(\tau) \) is a trigonometric function governing the kinematics of crank mechanism (see e.g. [14]), and \( P, P_h \) and \( P_e \) are the corresponding dimensionless pressures. These various parameters are normalized in the form:

\[ X = \frac{x}{r_c}; \quad Z = \frac{z}{r_c}; \quad H = \frac{h}{r_c}; \quad P = \frac{p - p_a}{p_a} \]

(23)

Note that \( r_c \) the crank radius is half the piston stroke, and \( p_a \) is an ambient pressure.

The other dimensionless parameters, \( \Lambda_1, \delta, \ v_c \) and \( \Lambda_2 \) are given by:
\[ \Lambda_1 = \frac{3\mu\omega}{p_a}; \quad \delta = \frac{2r_p}{r_c}; \quad \varepsilon_c = \frac{h_p}{r_c}; \quad \Lambda_2 = \frac{\rho h_cr_p}{\mu^2} \] (24)

Fig. 13. Piston ring, cylinder liner and film thickness cross-section [26]

More details as well as the formulation of the friction force between the piston ring and cylinder liner, due to viscous shear stresses can be found in Ref. [26].

3.5. Soft Elasto-Hydrodynamic Lubrication (SEHL)

Theoretical modeling of surface texturing for applications involving elastomeric components such as: O-rings and various other geometries of reciprocating sealing rings, lip seals, elastomeric bearings etc. is relatively new, e.g. [23,24]. Hence, differently from the previous applications, no experimental evidence has yet been accumulated to validate these models. There are two options in SEHL applications regarding the surface to be textured. Texturing can be applied either on the rigid surface or on the soft elastomeric one. While texturing elastomeric surfaces may present a technological challenge it has one obvious advantage at least in reciprocating sealing where the area to be textured is much smaller if the short elastomeric seal rather than the long reciprocating shaft is textured. Another advantage of texturing the soft rather than the rigid surface may be the prevention of wear of the soft elastomer by the harder surface features (protrusions or indentations) during unfavorable lubrication conditions. In this section the modeling of a textured elastomeric sleeve in rotary sliding [24] is described as an example for SEHL application.

A schematic illustration of the SEHL model is shown in Fig. 14a. A stationary elastomer sleeve with rectangular cross-section is fitted on a rotating shaft. A regular surface texturing is applied to the inner surface of the sleeve as shown schematically in Fig. 14b and in more details in Fig. 14c. The dimples are uniformly distributed on the elastomer surface with an area density \( S_p \). The dimples are arranged in a grid of longitudinal columns (see Fig. 14d for a single column) and circumferential rows. As in all previous applications here too all
the dimples are identical spherical segments with a base radius, $r_p$, and a maximum depth, $h_p$. Each dimple is located in the center of an imaginary square cell of sides $2r_1 \times 2r_1$.

The longitudinal cross-section of the elastomer sleeve before (dashed line) and after (solid line) deformation is shown in Fig. 15a. The sleeve is fitted on the shaft with an interference fit, $c$, while its outer circumference is fixed to a rigid foundation. Due to the surface texturing a hydrodynamic pressure is generated between the rotating shaft and the elastomer sleeve, causing deformation of the latter. This pressure may be sufficient to separate the initially interfering mating surfaces by a thin fluid film with a thickness distribution $h(x_1, x_3)$, which is given by:

$$h(x_1, x_3) = -c + h_0(x_1, x_3) + \delta(x_1, x_3) \tag{25}$$

In Eq. (25) $h_0(x_1, x_3)$ is the local depth of an un-deformed dimple (see Fig. 15b), and $\delta(x_1, x_3)$ is the variation of the local film thickness due to radial deformation of the elastomer inner surface.

![Diagram](image_url)

*Fig. 14 - (a) A schematic of the SEHL model; (b) an elastomer sleeve with regular surface texturing; (c) a geometrical model of a textured surface. (d) A single column of dimples [24].*
For analyzing the surface texturing effect a full hydrodynamic lubrication between the textured elastomer and rigid shaft is assumed during operation, i.e. no contact between the mating surfaces is allowed. The fluid film thickness is assumed much smaller than the shaft radius, thus curvature in the circumferential direction $x_3$ can be neglected. This allows using Cartesian instead of cylindrical coordinates and replacing rotational by translational velocity in the $x_3$ direction. Roughness of the shaft and the elastomer surfaces is neglected except for the regular surface texturing. The fluid is assumed Newtonian, incompressible, and with a constant viscosity $\mu$. Further assumptions related to relevant properties of the fluid and the elastomer can be found in Ref. [24].

Based on these assumptions the Reynolds equation for the local pressure in the fluid film, and the set of elasticity equations for the elastomer deformation are given in the form (see [23]):

$$\frac{\partial}{\partial x_1}\left( h^3 \frac{\partial p}{\partial x_1} \right) + \frac{\partial}{\partial x_3}\left( h^3 \frac{\partial p}{\partial x_3} \right) = 6\mu U \frac{\partial h}{\partial x_3}$$

(26)
\[
\left[ \frac{1}{2}(u_{i,j}^* + u_{j,i}^*) + \left( \frac{v}{1 - 2v} \right) u_{k,k}^* \delta_{ij} \right]_{,j}^* = 0 \quad (27)
\]

where \( p \) is the local hydrodynamic pressure, \( U \) is the relative sliding velocity, \( u_i^* \) represents the components of the elastomer displacement vector distribution, and \( \delta_{ij} \) is the delta-Kronecker. The index notation of the general form \( u_{i,j} \) denotes partial derivative of \( u_i \) with respect to \( x_j \), and a repeated index denotes summation (e.g. \( u_{k,k}^* = u_{i,i}^* + u_{2,2}^* + u_{3,3}^* \)). In order to evaluate the load carrying capacity and the viscous friction force, a simultaneous solution of Eqs. (26) and (27) is required [24].

The dimensionless form of the problem parameters consists of normalizing all length dimensions by \( r_p \) (see also [23]) in addition to:

\[
\varepsilon = \frac{h_p}{2r_p}; \quad P = \frac{(p - p_a)r_p}{6\mu U}; \quad E = \frac{E^* r_p}{6\mu U} \quad (28)
\]

where \( p_a \) is the ambient pressure and \( E^* \) is the Young’s modulus of the elastomer.

The dimensionless forms of the local film thickness (Eq. (25)), the Reynolds equation (Eq. (26)), and the elasticity equations (Eq. (27)) are given as:

\[
H(X_1, X_3) = -C + H_0(X_1, X_3) + \Delta(X_1, X_3) \quad (29)
\]

\[
\frac{\partial}{\partial X_1} \left( H^3 \frac{\partial P}{\partial X_1} \right) + \frac{\partial}{\partial X_3} \left( H^3 \frac{\partial P}{\partial X_3} \right) = \frac{\partial H}{\partial X_3} \quad (30)
\]

\[
\left[ \frac{1}{2}(u_{i,j} + u_{j,i}) + \left( \frac{v}{1 - 2v} \right) u_{k,k} \delta_{ij} \right]_{,j} = 0 \quad (31)
\]

Because of periodicity of the surface texturing in the circumferential direction (see Fig. 14c), it is sufficient to consider the pressure distribution within just one single dimples column with its proper boundary conditions (see Fig. 14d).

The deformations of the elastomer sleeve are also periodic in the circumferential direction \( x_3 \) with the same period of \( 2r_1 \) as the hydrodynamic pressure. Hence, it is sufficient to solve Eq. (31) for a single elastomer slice of width \( 2r_1 \) (shown in Fig. 16a) that is associated with the single column of dimples shown in Fig. 14d. The boundary conditions for Eq. (31) relate the dimensionless local shear and normal stresses, respectively, to the dimensionless elastic displacements (see Ref. [23]) at the inner surface of the elastomer sleeve. These shear and normal stresses are equal to the Couette viscous shear and the hydrodynamic pressures in the fluid, respectively.
Fig. 16 - Elastomer model: (a) a single elastomer slice (associated with a column of dimples) and its boundary conditions; (b) the finite element mesh of the elastomer slice [24].

The numerical procedure for the simultaneous solution of the Reynolds equation and the elasticity equations consists of the following steps: An initial guess of the film thickness distribution is made and the Reynolds equation (30) with its proper boundary conditions is solved by a non uniform grid finite difference method (see e.g. Ref. [8] and section 3.1 above). This provides a first approximation of the pressure and shear stress distributions, which are used as boundary conditions at the inner surface of the elastomer sleeve.

The elastomer deformations are calculated from Eq. (31) and its boundary conditions (see Ref. [24]) by finite element commercial software ANSYS with higher order 3-D 10-node tetrahedral solid element and non uniform mesh as shown in Fig. 16b. The finite element mesh is finest at the textured surface and becomes coarser towards the built-in plane. Local fluid pressure and shear stress are applied at element nodes by using surface elements. The finite element solution provides the deflections of the elastomer and the dimensionless variation in the local fluid film thickness, $\Delta(X_1,X_3)$, which in turn are used to correct the fluid film thickness distribution by Eq. (29). The new film thickness distribution is returned to the Reynolds equation and this iterative process is repeated until a desired convergence is achieved.

The dimensionless load carrying capacity, $W$, and friction force, $F_t$, are obtained by:

$$W = \frac{1}{A} \int \int P(X_1,X_3) dX_1 dX_3 = \frac{r_p}{6 \mu U} W$$

$$F_t = \frac{1}{A} \int \int \frac{dX_1 dX_3}{6H} = \frac{r_p}{6 \mu U} F_t$$
where $A=2R_1L_1$ is the dimensionless area of a single dimples column and $w$ and $f_t$ are the dimensional average pressure and shear on the elastomer slice inner surface, respectively. Finally, the coefficient of friction is defined as the ratio of the friction force over the load capacity:

$$\eta = \frac{f_t}{w} = \frac{F_t}{W}$$

The model dimensionless parameters required to investigate the load capacity and friction are the following: The operating conditions are represented by: the SEHL stiffness index, $E=E^{r_p}/(6\mu U)$, the interference fit, $C=c/r_p$, and the pressure differential $P_r$. The elastomer geometry is given by the sleeve dimensions $L_1=l_1/r_p$ and $L_2=l_2/r_p$. The texturing parameters include the dimple aspect ratio, $\varepsilon=h_p/2r_p$ and the area density, $S_p$.

Differently from all the previous applications that are characterized by rigid mating surfaces, in SEHL there is only a weak optimum of the texturing parameters regarding the load capacity. This can be explained by the opposite effects of hydrodynamic pressure and fluid film thickness on the load capacity. An effective texturing which would increase the hydrodynamic pressure also increases the elastomer radial deformation, which increases the film thickness and hence, offsets the effect of the texturing on the load capacity. However, this increase of the film thickness has a beneficial effect on reducing the viscous shear and therefore, it reduces the friction force and the coefficient of friction.

4. The Validity of the Reynolds Equation for Textured Surfaces

Although the theoretical models based on solving the Reynolds equation showed good agreement with experimental results, e.g., [8,13,15,19] as shown, for example, in Fig. 17 taken from Ref. [15], it was argued on several occasions, e.g., [27] that the Reynolds equation may not be valid when applied to textured features that have large aspect ratio (the ratio of depth over diameter or width) and that the full Navier-Stokes (NS) equations should be employed. In order to clarify this issue, both the full NS equations and the Reynolds equation were solved for the case of a compressible fluid at no sliding but with a pressure differential to simulate a hydrostatic gas seal [28]. A comparison between the two solution methods illustrated that in spite of potential large differences in local pressures the differences in load carrying capacity are small for realistic geometrical parameters of LST. Hence, the Reynolds equation can be safely used for most LST applications.
Fig. 17. Correlation between experimental and theoretical results of friction-time variation for simulated textured piston rings at 1000 rpm under full lubrication condition [15]

Fig. 18 Schematic of (a) plane inclined slider bearing and (b) textured slider with pitch angle $\alpha$ [29]
It should be noted here that, as discussed in [6], surface texturing was also attempted in non-parallel sliding with full hydrodynamic lubrication applications such as thrust bearings and journal bearings. However, in these cases the texturing is beneficial only when the global film convergence is small enough as is shown, for example, in Ref. [29] for textured magnetic recording sliders, see Figs. 18 and 19.

![Fig. 19 Non-dimensional average pressure as a function of pitch angle $\alpha$ for the un-textured and textured sliders of Fig. 18, showing that surface texturing is beneficial only at small pitch angles [29]](image_url)

In yet another non-conformal contact application such as ball bearings it was found that surface texturing can be beneficial only with very shallow dimples [30]. These limitations underline again the most beneficial use of surface texturing in parallel surfaces hydrodynamic and hydrostatic applications.

5. Optimization

In order to fully benefit from surface texturing a proper optimization of the geometrical parameters must be performed in accordance with the application in hand. This includes the aspect ratio of the dimples, their area density and, in cases of partial surface texturing, the textured portion. The preferred and most efficient way to optimize surface texturing is by parametric analysis in a theoretical model. This was done in several models for bearings [18,29], various seals [8,11,13,23,24,31], piston rings [14,26] and magnetic recording tapes [32] to obtain maximum load capacity, minimum friction, maximum film stiffness and minimum leakage. Many experimental studies attempting optimization by trial and error approach can also be found in the literature (see, e.g., [6]). These include, for example, different texturing geometries and dimple shapes like squares, triangles, ellipses, grooves etc. Unfortunately, quite often wrong conclusions are arrived at in these studies due to insufficient experimental data. Another typical mistake, which should be avoided in optimizing surface texturing, concerns a comparison of different texturing shapes based on a common parameter such as dimple size or area density. Here too, wrong conclusions are usually made regarding the optimum shape for best performance. The correct procedure for
finding an optimum texturing among different shapes is first to optimize each shape individually in terms of its own optimum aspect ratio and area density and only then compare the individual optimums to find the ultimate one. Such a procedure is described in Ref. [17] where an optimum conventional un-textured barrel-shaped piston ring was compared with an optimum surface textured cylindrical piston ring in a firing diesel engine resulting in about 4% improvement in fuel consumption with the optimum flat face textured ring. In previous studies, when the texturing was applied to the conventional barrel-shaped piston ring no difference was observed between the textured and un-textured cases, leading to the wrong conclusion that surface texturing has no benefit in piston rings.

6. Summary

Surface texturing, and more specifically laser surface texturing (LST) technology, has great potential in improving tribological performance of various mechanical components over a wide range of different operating conditions. The micro-dimples produced on the surface can act as micro-hydrodynamic bearings in cases of full or mixed lubrication with either incompressible or compressible lubricants.

Surface texturing is most beneficial in cases of parallel sliding with full fluid films. In these cases the effect of texturing geometry on the tribological performance can be easily modeled and optimized for best required performance. Such optimization can provide substantial reduction in friction losses and result in much desired energy savings. In non parallel/non conformal lubricated contacts the benefit of surface texturing is rather limited and may even become detrimental i.e. [29,33].

References


Chapter 2
Surface texture design and applications
Stephen Hsu, GWU, USA

INTRODUCTION

The basic concept of controlling the surface topography to improve friction and wear performance has been recognized since the early 1900s, and has been practiced in specific instances. The extent of the use of controlled surface topography is limited by economic considerations: i.e. the cost of fabricating the features versus the benefits gained. Cross-hatching of diesel cylinder liner to reduce scuffing and seizure was introduced in the 1940s and they are still in-use today (1). Dimples have been introduced on the surface of golf balls to reduce the aerodynamic drag so that they can fly longer and straighter. Modern tires use intricate surface textural designs to control traction and friction with road surfaces, and this has spawned a whole new area of technological innovations (2). As fabrication methods and materials continue to improve, surface engineering and textural control are increasingly being recognized as potential tool to overcome specific performance challenges.

As technology evolves, advanced surface processing techniques such as controlled grinding, chemomechanical polishing, and single diamond turning have become increasingly available for surface topography control. Laser texturing using focused laser beams has also been developed (3-5) for magnetic hard disks and seals. Micro-photolithography and subsequent reactive ion etching have been developed to fabricate sophisticated surface features at relatively low cost (6). With the availability of nanoindenters and scratch testers, nanomechanical means to carve textural features on surfaces are now feasible to produced precisely controlled geometric sizes and shapes at the micrometer scale. Focused ion beam techniques and sputtering have also been used to fabricate surface textures (7). The availability of all these relative low cost fabrication tools has rekindled surface texturing research.

Significant tribological success with engineered surface texture has been reported for a number of different areas: mechanical face seals with laser ablation dimples to improve durability and reduce friction (8); metal forming operation with introduced pits to prevent adhesion and seizure (9); and magnetic hard disk with laser textured bumps to minimize stiction (10). The examples described above highlight the increasing use of surface texturing as part of surface engineering technology to achieve desirable outcomes through empirical trial-and-error. However, the application of the surface texturing technology is limited by the lack of understanding of the fundamental processes involved in friction reduction under various lubrication regimes. Some hypotheses of lubrication improvements by applying surface textures may summarized as follows:

1) the dimples serving as lubricant reservoirs that provide
   a. extra lubrication, or
   b. cooling,
2) trapping of wear particles in the grooves or dimples,
3) development of pressure spikes in the dimples due to cavitation, inertia, or development of mini pad bearings,
4) combinations of these mechanisms.
The understanding of the interplay between operating conditions and surface texture size, shape, and patterns remains rudimentary. Further advances in the surface technology require a breakthrough in the fundamental understanding of how discrete surface textures influence the fluid mechanics, contact mechanics, and tribology.

In this report, the effects of discrete geometric shapes on friction were studied. Various shapes were fabricated by using photolithography followed by electrochemical etching of metal surfaces. The textured surfaces were measured for friction under a step-loading test method under fully lubricated conditions using a flat-on-flat geometry on a pin-on-disk wear tester. The results suggest that both the shape and the orientation with respect to the sliding direction have significant influence on friction.

**BACKGROUND**

Surface textures have been used to decrease friction and increase seizure load for metal and ceramics surfaces lubricated [8, 11, 12, 13]. The mechanisms proposed include: generating additional fluid pressure under hydrodynamic lubrication [14], providing lubricant under highly loaded contacts to prevent seizure [15], and trapping wear particles to minimize third body abrasion under non-lubricating conditions [16]. These studies highlight the benefits of surface textures under poorly lubricated contacts under boundary lubrication conditions and well lubricated contacts under hydrodynamic lubricated conditions. The surface textural features studied include symmetrical round dimples, grooves, spirals, and cross-hatched grooves of various dimensions.

Etsion [17] developed hydrodynamic models for lased textured surfaces with symmetrical circular dimples and found several important factors in dimple design. The hydrodynamic lift depended on the size, depth, and number of dimples in the contact area. The number of dimples in the contact area can be quantified by the area ratio defined by the area of dimpled surface over the total area in the contact. Other studies also suggested that an area ratio of 5-20% was effective [17, 18, 19, 20]. The ratio of dimple depth over the diameter of the dimple also has significant influence on hydrodynamic lift [21].

Based on these studies, one can conclude that the effect of circular dimples appears to provide additional hydrodynamic lift in conformal surface contacts under relatively low apparent contact pressures such as seals and journal bearings. However, for components, such as gears and rolling element bearings, the lubrication condition is quite different from the seals and journal bearings. As illustrated in Figure 1, the lubrication regimes can be qualitatively classified based on somewhat arbitrary divides of industrial components/systems operating under various speed and load combinations. Each region represents different degree of influence by the three basic lubrication modes: hydrodynamic, elastohydrodynamic (EHL or mixed), and boundary lubrication regimes.
Since each of these regimes represents different combination of lubrication mechanisms, surface texture patterns that work in one regime do not necessarily work in another regime. The optimum surface texture design in terms of size, shape, depth, and orientation can differ significantly from regime to regime. A discussion of the current understanding of surface texturing in these three regimes is provided in the following paragraphs.

**High speed low load conditions**

In this regime, ideally, under steady state conditions, the load is fully supported by the hydrodynamic film pressure. The surfaces are separated by a continuous fluid film and the thickness of the film is controlled by the contact geometry (conformal and non-conformal contacts), speed, load, and viscosity of the lubricant. Surface textures in this regime are often used to promote the onset of the hydrodynamic lubrication mechanism, and in turn, reduce the friction.

The most notable studies in this regime are the pioneering papers by Etsion and his group (5, 8, 17, 21-22). A focused laser beam is used in pulsating mode to generate micro-dimples rapidly on various metal surfaces. These dimples enable durable energy efficient operations of many mechanical seal designs (nominal apparent contact pressures from 0.1 MPa to about 15 MPa and speed range from 0.5 m/s and up). He cited 40%-50% reduction in frictional torques and nearly doubling seal service life by various manufacturers (23). He reported that within this range of speed and load for conformal contacts, the friction reduction mechanisms were: a) enhanced hydrodynamic lubrication by early entry into the hydrodynamic regime (21); b) possible cavitationally lift effects for some systems; and c) reverse flow inside the dimples or induced by the dimples (22). Etsion also developed hydrodynamic models for laser textured surfaces with symmetrical circular dimples and proposed several parameters for the design of dimples in this regime (17). The hydrodynamic lift depended on the size, depth, and number of dimples in the contact area. The number of dimples in the contact area can be quantified by the area ratio defined by the area of dimpled surface over the total area in the contact. He suggested that for typical seal applications, a surface texture using circular dimples (100 µm diameter and 10 µm in depth) at 20% area coverage may be a good starting point (5). Further refinements are needed depending on specific operating conditions and surface materials. The ratio of dimple depth over the diameter of the dimple also has significant influence on hydrodynamic lift (21).

**High speed medium load conditions**

![Figure 1 Division of the operating regimes of industrial components and systems](image-url)
The success achieved in the seals area kindled interest in studying further the possibility of textures in other more severe contact applications. However, for the textured surfaces that worked in the low load-high speed regime, friction tends to increase rather than decrease when the apparent contact pressures (for typical engineering material surface roughness) is increased to a level that elastohydrodynamic lubrication takes place. In this regime, the asperities are under elastic deformation and there is still a continuous fluid film flowing through the contact.

There are contradictory reports in the literature on the effects of textures on friction under EHL regime (24-26). Our own studies suggested when the apparent contact pressure exceeds 180 MPa, friction actually increased. Both theoretical modeling and experimental results from references 24 and 26 confirm this. This is primarily due to the edge stresses induced by the elastic contacts around the dimple edges. Effective design of textures and patterns in this regime are being studied currently by many groups around the world.

**Low speed high load conditions**

Most of the load is supported by the asperity contacts under low speed, high load conditions. Under such conditions, local plastic deformation of asperities and wear can occur (boundary lubrication regime). Studies in this regime are characterized by contradictory reports (27-32). Suh reported friction reduction by introducing parallel grooves in relatively poorly lubricated systems such as titanium alloys to trap the wear particles, hence reducing significantly the third body abrasion effects (27-28). Pantelis reported to achieve benefits of increased anti-galling by surface texturing (29). Persersson reported some benefits under very limited conditions using different geometries but also reported instances where friction actually increased (30, 31).

The above review clearly suggests that for high speed low load conformal contacts, significant friction reduction and increase in durability can be obtained through the use of surface texture dimples. However, there is no consistent conclusion can be drawn for the effect of surface texture under EHL and boundary lubrication conditions. This may also interpreted as the lack of research and development in the theory and practices for these two regimes. In terms of energy conservation, of course, the majority of energy conversion devices and components operate in these two regimes. Therefore, research for textural design is clearly needed under these regimes.

**SURFACE TEXTURE DESIGN**

**Effect of texture feature and directionality**

Throughout our review of the use of surface textures to reduce friction, various investigators have used grooves, circles, squares, triangles as surface textures, but conclusion and comparison are difficult to make. The main reason is that many of the parameters involved were not controlled in the experiments: including surface roughness, area ratio of the textured area to the total area, precise orientation of the textures, surface patterns of the textures, and depth and diameter of the textural features. There is no systematic study that compares different features on an equal area ratio basis and measures the effect of orientation with respect to the sliding direction. Geiger et al (33) studied triangles, circles, and rectangles of various diameters (25 µm to 150 µm) and depths from 5 µm to 20 µm of ceramic pairs in a ring and block configuration with a load of 325 N and a speed range of zero to 1.85 m/s linear
velocity. He reported that many small diameter features were more effective and the friction reduction was small. He further concluded that the influence of the shape and depth of the surface features were relatively small in his system. Jeng (34) used a pin-on-disk tester to study various forms of small grooves in the form of controlled roughness. He reported the effect of parallel roughness on friction and concluded that grooves generally increase friction and perpendicular roughness was better than parallel lines with respect to the sliding directions. In both of these studies, there was no equivalent basis of comparison and conclusion on geometric shape and size could not be drawn.

This report focuses on the fundamental issue of the effect of geometric shape and orientation of surface texture designs. For this purpose, we compared the untextured surface with surfaces textured with circular dimples, triangles, and ellipses in a pin-on-disk wear tester, which simulated the friction in lubricated steel on steel system.

**Surface texture design and fabrication**

In order to make a comparison of dimple shape and sliding direction effects on friction characteristics, the surface texture designs are controlled as such that the dimple depth, the area of one dimple, the area ratio (percentage of dimple area to the total area), as well as the pitch (the distance between two dimples) remain the same for different dimple shapes. The radius of circular shape is 75\(\mu\)m. The radiuses of major and minor axes of elliptical shape are 150\(\mu\)m and 37\(\mu\)m, respectively. The side length of the triangular shape is 188\(\mu\)m. The resulting area of one dimple is the same for all shapes as 17670\(\mu\)m\(^2\). The area ratio yields as 7\% with a pitch of 500\(\mu\)m. The depth of the dimple is 8\(\mu\)m. The geometry parameters of the surface texture design are summarized in Table 1. All features were arranged in a hexagon array with the same depth and the same area ratio as shown in the Table 1.

<table>
<thead>
<tr>
<th>Pattern &amp; sliding directions</th>
<th>D depth ((\mu)m)</th>
<th>Radius ((\mu)m)</th>
<th>Area of a dimple ((\mu)m(^2))</th>
<th>Pitch ((\mu)m)</th>
<th>Area ratio (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Circle</td>
<td>8±1</td>
<td>75±4</td>
<td>17670±880</td>
<td>500±15</td>
<td>7±0.4</td>
</tr>
<tr>
<td>Ellipse</td>
<td>8±1</td>
<td>150±7 37±2</td>
<td>17670±1000</td>
<td>500±20</td>
<td>7±0.4</td>
</tr>
</tbody>
</table>

Table 1 Geometry parameters of the patterns of dimples
The texture designs were fabricated with electrochemical etching. Photoresist film was coated on substrate by spin coating. Micro-photolithographic technique was used to fabricate photoresist masks with pre-fabricated photomask that designed with specific geometric shapes and dimensions. The texture pattern of the photomask was then been transferred onto the photoresist mask and electrochemical etching was followed in 1 M NaCl solution, at 0.2 amp DC current and room temperature. Carbon steel was used as substrate for this study. Since the carbon steel is a multiphase material, the etched surface is rough in appearance, as shown in Figure 2. The schematic diagram of the etching process and the micrographs of the etched textures are also shown in Figure 2. The etched wall of the feature inclines at about 50 ~ 70° angle with respect to the vertical axis.

EXPERIMENTS

Experimental apparatus and test procedure
pin-on-disk wear tester (shown in Figure 3) was used to measure the friction characteristics of untextured and textured surfaces. A flat pin was modified slightly to enable a flat-on-flat test geometry as shown in Figure 3. A small disk (6.25 mm in diameter) was attached to a ball-joint so that the top surface automatically would align with the bottom flat disk. The lower disk (rotating disk) was made of carbon steel ANSI 1017-1018. It was ground and polished to an average roughness of 0.039 µm Ra but not textured. The top counter surface was polished to 0.008 µm Ra and textured.
The experimental conditions were designed to obtain friction measurement as a function of the Sommerfeld number (dimensionless parameter, viscosity × speed × contact width / load). So a two dimensional experimental design of varying speed at constant load, and varying load at constant speed was used to generate friction data as a function of speed and load. The range of speed and load are shown in Table 2.

<table>
<thead>
<tr>
<th>Table 2</th>
<th>Experimental conditions</th>
</tr>
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<tbody>
<tr>
<td>Diameter of flat pin specimen: 6.35 mm</td>
<td></td>
</tr>
<tr>
<td>Load range: 1-35 N</td>
<td></td>
</tr>
<tr>
<td>Apparent contact pressure: 0.03-1.1 MPa</td>
<td></td>
</tr>
<tr>
<td>Speed range: 0.023-0.23 m/s</td>
<td></td>
</tr>
<tr>
<td>Lubricant: Paraffin oil (Saybolt viscosity at 100°F: 125/135)</td>
<td></td>
</tr>
<tr>
<td>Temperature: room temperature</td>
<td></td>
</tr>
</tbody>
</table>

The samples were lubricated with pure paraffinic oil without additives to ensure that there is no friction modifier in the oil to interfere with the experimental results. Wear of the specimen was potentially a problem but under the experimental conditions, the apparent contact pressures were less than 2 MPa and the wear was negligible.

The same test protocol was also applied to untextured samples were to provide baseline data for comparison. The tests were conducted using a step-loading or step-speeding to measure the friction. The step change was one minute duration or time to reach steady state friction trace, the step loading was 1 N per step at low loads and 5 N per step at loads higher than 10N. The step speed was 0.023 m/s (10 rpm) at low speeds and 0.12 m/s (50 rpm) at speeds above 0.12 m/s. In this series of experiments, the test starts with low speed at a load and the speed was ramped up step by step. Under a specific load, the low speed start up conditions was actually the most severe contact conditions, and occasionally, wear was observed at the end of the test series. This test sequence probably contributes significantly to the test uncertainty.

Surface textures of circular, elliptical and triangular shapes were applied to the flat pin specimens as illustrated in Table 1. In addition to the geometric shape differences, orientation of the sample with respect to the sliding direction was also investigated. Two sets of test specimens with the same geometric shapes were used for this purpose. The elliptical dimples were aligned either parallel to or perpendicular to the direction of sliding. As marked in the
result figures presented in the following paragraphs, “Ellipse ⊥” and “Ellipse //” represent the cases that sliding direction is perpendicular and parallel to the major axis of ellipse, respectively. In addition, “Triangle ∆↑” and “Triangle ∆↓” correspond to the cases in which the lubricant is driven toward the apex and the base of triangles, respectively.

**Experimental results and discussion**

Under the test protocol developed, seventeen measurements were carried out. The resulting Sommerfeld number, $\eta VB/W$ ($\eta$ is the viscosity of the lubricant, $V$ is the speed, $B$ is the contact width, and $W$ is the nominal load) varies from $9.2 \times 10^{-5}$ to $1.7 \times 10^{-7}$ to obtain the Stribeck Curve.

The friction data were first plotted as a function of the Sommerfeld number to show the transition from hydrodynamic lubrication to mixed lubrication and then into boundary lubrication. By comparing the results of various surface textures at the same area ratio, the effect of geometric shapes on friction should be clearly observed in terms of friction transitions from one regime to the other and also the intrinsic friction level in each regime.

Figure 4 shows the Stribeck curves of untextured and textured specimens with the major axis of ellipse oriented parallel to the sliding direction. All of the curves show clear transitions from hydrodynamic to mixed lubrication. There are considerable data scatter due to the dynamic nature of the friction measurement and the potential deviation of geometrical dimple shapes, even though the deviation is within the statistical uncertainties. The friction curves all have a minimum friction level, indicating the change from mixed lubrication mode to the hydrodynamic regime at higher Sommerfeld numbers. For example, the untextured surface has its lowest friction coefficient of 0.0083 at the value of $\eta VB/W$ about $3.2 \times 10^{-6}$. Compared to the untextured surface, both circular and elliptical dimples show lower friction coefficients.
Figure 4  Stribeck curves of the various geometric shapes (major axis of ellipse oriented parallel to the direction of sliding), where $\eta$ = viscosity [Pa·s], $V$ = sliding speed [m/s], $B$ = contact width [m], and $W$ = load [N].

Figure 5 shows the friction data for the untextured, circles, and ellipses with the major axis oriented perpendicular to the sliding direction. The testing results clearly indicated that a significant friction reduction for elliptical dimples when the sliding direction is perpendicular to the major axis of the ellipse at low Sommerfeld number. Compared to the results shown in Figure 4 for the case of ellipse parallel, it is more favorable to adopt the perpendicular orientation of ellipse dimples in low speed, high load conditions.

Figure 5  Stribeck curves of various geometric shapes (major axis of ellipse oriented perpendicular to the direction of sliding), where $\eta$ = viscosity [Pa·s], $V$ = sliding speed [m/s], $B$ = contact width [m], and $W$ = load [N].

Figure 6 presents the Stribeck curves of the baseline and the samples with triangular dimples arranged in the two directions defined in the previous section. Again, significant friction reduction observed with triangular dimple when the lubricant is driven toward the base of the triangle at low sommerfeld numbers. Comparison of Figs 4-6 shows that the effect of feature shape and orientation on the friction results is dramatic. As previous hydrodynamics-based theoretical models are based on circular (isotropic, symmetric) dimples, such strong dependence on geometric shape or orientation is unexpected.
Energy consumption due to frictional loss

Given the degree of uncertainty in the exploratory experimental results, we decided to conduct a more controlled set of friction tests with the load ranging from 1 N to 35 N, and speed ranging from 0.023 m/s to 0.23 m/s. This time, the test sequence was reversed: for a given load, the start-up condition was at the highest speed, 0.23 m/s (100 rpm), from which the speed was initially reduced by a step of 0.12 m/s (50 rpm). After the speed reached 0.12 m/s, then the step of change was set at 0.023 m/s (10 rpm). This change in test procedure improved the test precision and the results reveal a different degree of dimple geometry effects. For all the samples tested, Figures 7(a) and 7(b) compare the total energy consumed by friction in the duration of tests based on the experimental results obtained with the two different procedures, respectively. In generating the two figures, the total energy consumption is calculated with the following equation:

\[ E_{\text{total}} = \sum_{\text{load}} \sum_{\text{speed}} \text{friction force} \times \text{sliding distance} \]  

(1)
Both figures clearly show that, among all the shape and orientation combinations, the elliptical dimples perpendicular to the sliding direction are most effective in reducing friction over the range of operating conditions tested. However, Figure 7(a) shows that the energy consumption or friction is strongly dependent on both dimple shape and orientation while the measurement from the more precise procedure only shows a strong sensitivity to the dimple shape.

**Friction mapping as a function of speed and load**

Figures 4-6 show that there is a transition from mixed lubrication to hydrodynamic lubrication with the associated friction minimum. If one starts from hydrodynamic regime, under constant speed, that is a load beyond which the friction will increase, indicating the transition into the mixed lubrication regime. Figure 8 (a) illustrates such a transition by plotting the friction coefficient as a function of load at $V = 0.058$ m/s. We can call this load the critical transition load (the highest load under a specific speed that will sustain low friction). Similarly, Figure 8 (b) presents the friction coefficient as a function of speed under a constant load. As shown in this figure, starting from the high speeds, there will be a speed
below which the hydrodynamic film thickness can no longer be sustained. We will call this critical transition speed (the lowest speed under a specific load that can sustain the low friction). Figure 9 shows the critical speeds of each of the samples examined under loads of 15 and 20 N. The critical speed and load can provide a simple indicator for the effective operating range of a specific surface textural geometry and pattern.

Figure 8 Friction coefficients as a function of load and speed at a constant condition

Figure 8 Friction coefficients as a function of load and speed at a constant condition
DISCUSSION

This study examines the interplay between geometrical features under equal area coverage and the onset of hydrodynamic lubrication, lowering the friction between sliding contacts. The apparent contact pressures between the flat-on-flat contact surfaces are very low, in $10^0$ MPa range. In the experiments, we have observed occasional tilting of the pin on the surface. This artifact will be even more severe under reciprocating motions. We modified the sample loading device to introduce self-aligning ball and socket joint to minimize the potential tilt under high loads.

The elliptical textural features perpendicular to the direction of sliding provided the best friction reduction capability. The dependence of friction under this fluid film supported regime on the geometric shape and orientation is unexpected. Controlled experiments were conducted in our laboratory by using a flat sample with a transparent single dimple sliding on a flat surface under a high speed camera. The video suggested potential cavitation mechanism in providing pressure lift, enabling the early entry into hydrodynamic lubrication. If this mechanism is dominant, then triangles with the base facing the sliding direction and the exit at the apex position would give the best friction reduction, yet this is not the case. In contrast, as shown in Fig. 7(b), the triangles with the base at the exit position show slightly better friction reduction. Clearly mechanisms other than cavitation may be at play as well.

We speculate at this time that in order for the hydrodynamic lift to occur; the edge pressure around the dimple has to be high enough to provide sealing of the trapped liquid. So it is a balance between the leakage rate and the hydrodynamic pressure build-up within the surface feature. For features such as circles, ellipses parallel to the sliding direction, fluid leakage...
may be more pronounced hence lower lift. The elliptical narrow shape would behave like an
infinitely long bearing under sliding conditions to provide maximum sealing pressures to
allow for the maximum lift force per unit time. This speculation will have to be proven in the
future.

CONCLUSIONS

These early results have demonstrated that dimple surface texture can have significant
benefits for lubricated sliding. It can lower the hydrodynamic-to-mixed lubrication transition
to lower friction and Sommerfeld number values (higher load/lower velocity), and it can
reduce the boundary lubrication friction by around 35%. The major axis of elliptical dimple
perpendicular to the direction of sliding, and the fluid is driven toward the base of the
triangular shape appear to be critical. The results do not at this stage permit us to
qualitatively distinguish between several proposed mechanisms – lift due to cavitation, inertia
or mini wedge formation in the early mixed lubrication regime and mini wedge/oil
distribution in the boundary lubrication case. Additional experimentation will be required to
elucidate these factors.

The conclusions can be summarized as follows:

1) A low cost fabrication technique has been developed for surface texturing on steel or
other metals. The dimensions can be precisely controlled in terms of size, shape, and
distance between features.

2) Dimples of various shapes under the same area ratios are compared in a series of
friction tests with a flat on flat contact. The geometric shape has strong influence on
friction under relatively high speed low load conditions.

3) Orientation effect was also observed, ellipses placed perpendicular to the direction of
sliding showed the best result of friction reduction with the lowest friction transition
point and overall low friction.

4) Friction mapping of the textures as a function of speed and load suggested that we can
use the critical speed (lowest speed that surface texture would work) and critical load
(highest load that surface texture will work) to compare various surface textural
features and orientation effects.

5) More studies to explain the observed phenomena and new theories are needed to
advance this new area of research.

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Chapter 3
Diamond-like Carbon protective films

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1. THE NATURE OF DLC FILMS/COATINGS

Diamond-like carbon (DLC) films are hydrogenated or hydrogen-free amorphous carbon materials consisting of sp\(^3\) and sp\(^2\) bonds (with only a tiny amount of sp hybridized sites). In general, the sp\(^3\) C-C connectivity controls the mechanical properties, while the sp\(^2\) sites determine the electrical and optical properties of DLC films [1]. The unique combination of exceptional chemical, electrical, optical, mechanical and tribological properties (e.g., low friction, high wear resistance, chemical inertness, relatively high optical gap and high electrical resistance) make them very attractive to both scientific and industrial communities [2]. However, DLC films prepared by various PVD and PECVD techniques present a wide range of structure, composition and properties, as seen in Fig. 1. For instance, the sp\(^3\) C-C content can reach 80% or more in certain non-hydrogenated DLC films and leads to hardness as high as 80 GPa. Such specific DLC films are generally designated as tetrahedral amorphous carbons or ta-C. H incorporation into the carbon matrix could break the interlinking of C-C network and thus result in the deterioration of hardness. However, hydrogenated carbon films with a high total sp\(^3\) fraction (e.g., >70%) and low H content (<30%) are still dense and hard (~50 GPa), and are referred to as ta-C:H (tetrahedral hydrogenated amorphous carbon). Only a further increase in H content and decrease of sp\(^3\) fraction that leads to a remarkable reduction of film hardness (~10-20 GPa). Such films are termed as a-C:H. As even more hydrogen is added into the carbon matrix, the H-terminated C-H bonds would completely destroy the three-dimensional interlinking and, consequently, prevent the formation of films. It is noteworthy that although H incorporation is harmful to mechanical property, it is tribologically preferred because it is able to passivate surface dangling bonds and provide an ultralow friction performance. Therefore, a-C:H films are more often used in tribological applications as friction-reducing and wear-resistant coatings.

Fig. 1. Ternary phase diagram of the C, H system [3]

For both H-containing and H-free DLC films, the maximum sp\(^3\) content and density was
obtained at an ion energy of about 100 eV per carbon atom, as shown in Fig. 2. This optimum energy range enables the maximum subplantion of incident energetic carbon ions which leads to a high local stress and density required for the formation of thermodynamically instable sp³ phase. The ions with lower energy could hardly penetrate the surface and just stick to it to form a sp²-bonded graphitic structure (for a-C:H, a more polymeric structure would be expected due to surface growth and radical reactions). Ion energy higher than 100 eV also leads to reduced sp³ content because the excess ion energy could not only increase the atomic mobility (even migrating back to surface) and relieve local stress, but also produce a strong heating effect within the thermal spike volume, thus prompting transformation from metastable sp³ bonding to stable graphitic sp² bonding.

The above relationship between sp³ fraction and ion energy is valid, however, only at a deposition temperature of lower than 100 °C. If the chamber temperature is higher, a sp³-to-sp² conversion will occur even at an ion energy of ~ 100 eV [4-6], Fig. 3.

For most DLC films, the hardness is directly linked to the content of sp³ C-C bonds. However, in some cases, certain nonhydrogenated carbon and carbon nitride films containing a large number of sp³ bonds also present high hardness (up to 55 GPa) and extreme high elasticity (elastic recovery of 85%) concurrently. This unique mechanical property has been attributed to the fullerene-like (FL) microstructures (e.g., curved graphene planes or closed
shells, Fig. 4) because the interlocked and curved graphite planes could extend extraordinary strength of a planar sp²-coordinated carbon network into three dimensions. Such FL structures are more readily to be observed in N-incorporated DLC films just because N atoms could provide the required out-of-plane bonds for curvature. Fullerene-like CNx has been characterized as a new class of extremely resilient materials, where deformation energy is predominantly stored elastically [7], as shown in Fig. 5. This can be understood by the cross-linked planar structure of fullerene-like films, where the sheet-like domains wrap and buckle under load, while the strong C–C cross-links prevent gliding. Thus, plastic deformation occurring in graphite is inhibited. Additionally, the interplanar strength of a predominantly sp²-hybridized material is considerable, which in turn reducing the possibility of cracking or bond-breaking by plastic yielding. Therefore fullerene-like CNx deforms by bond angle deflection rather than breaking and acts as a super-elastic material.

![Fig. 4. Schematic drawing and plan-view HRTEM image of FL carbon film.](image)

![Fig. 5. Deformation of fullerene-like structure under load.](image)

**A) DEPOSITION TECHNIQUES IN GENERAL**

DLC films can be prepared by various energetic deposition techniques including physical vapor deposition (PVD) and plasma-enhanced chemical vapor deposition (PECVD). The PVD techniques include ion beam (IB), mass selected ion beam (MSIB), unbalanced magnetron sputtering (UMS), filtered cathodic vacuum arc (FCVA), pulsed laser deposition (PLD), and so on.

In IB deposition process, carbon or hydrocarbon ions are generated by sputtering the graphite cathode or ionization of hydrocarbon gases in plasma. These ions are then accelerated to form an ion beam to deposit on the substrate surface, Fig. 6. However, in this
case, a large percentage of unionized neutral species is simultaneously present and strongly affects the film quality. To avoid this, a MSIB method is employed by introducing a magnetic filter to filter out those neutral species and select ion species with defined ion energies for film deposition. Nevertheless, the film growth rate of MSIB is very low.

Filtered cathodic arc process, Fig. 7 (a), is used extensively to deposit tribological and wear resistant thin films. The cathodic arc process produces particles with a wide range of sizes including macroparticles, ion and neutrals. The macroparticles could degrade coating quality and performance. As a result, a filtered cathodic vacuum arc (FCVA) technique was developed. A toroidal magnetic filter duct is applied to filter out the macroparticles and neutral species when the plasma passing through it because these particles cannot follow the magnetic field and will hit the walls. The electrons and ions, however, could spiral around the magnetic field lines and follow them around the filter axis, Fig. 7 (b). This filter could lead to a very high degree of plasma ionization (nearly 100 %) at the exit. At the cathode end, the ions have a mean energy of about 10-30 eV. However, the incident energy can be considerably increased after the accelerating effect of a bias voltage applied to the substrate. So the energy of the ions arriving at the substrate surface can be tuned from 20 to 3000 eV, similar to that of an ion source.

Sputtering has been the most common industrial process for the deposition of DLC films. In the basic sputtering process, a target (or cathode) plate is bombarded by energetic ions generated in a glow discharge plasma, Fig. 8. The bombardment process causes the removal or sputtering of target atoms, which may then condense on a substrate as a thin film. However, the process is limited by low deposition rates, low ionization efficiencies in the plasma, and high substrate heating effects. These limitations have been overcome by the development of unbalanced magnetron sputtering (UMS). In the UMS system, the magnets are arranged behind the target in such a way that one pole is positioned at the central axis of the target and the second pole is formed by a ring of strengthened magnets around the outer edge of the target. As a result, the secondary electrons can be not only constrained to the vicinity of the target to substantially increase the probability of ionizing, but also allowed to flow out towards the substrate to enhance the deposition rate. The most common form of depositing DLC is using the dc or rf sputtering of a graphite target by an Ar plasma. a-C:H or a-C:Nx films can be prepared by reactive sputtering using a plasma of Ar and hydrocarbons, or Ar and N2. However, like IB technique, the UMS technique only has a low ratio of energetic ions to neutral species and usually unable to produce hard DLC films with high sp³ fraction.
Fig. 7. Schematic of cathode source and a single bend FVCA.

Fig. 8. Illustration of a typical magnetron sputtering system.
No matter what deposition technique is employed, a common feature exists that the highest sp$^3$ content is achieved at moderate ion energy of ~100 eV per carbon atom. Therefore, to prepare ta-C films with high sp$^3$ content, plasma beams with a high ion fraction, a narrow energy distribution and a single energetic species are required. These applicable methods include mass selected ion beam (MSIB) deposition, filtered cathodic vacuum arc (FCVA) and pulsed laser ablation deposition (PLD).

Diamond-like a-C:H films (with H content as high as 60%) are generally produced by plasma-enhanced chemical vapour deposition (PECVD) of hydrocarbon molecules, or by the reactive sputtering of graphite in a H-containing atmospheres. The most frequently used PECVD method for a-C:H is radio frequency PECVD (RF-PECVD).

With further advancement of PECVD techniques, highly tetrahedral hydrogenated amorphous carbon or ‘ta-C:H’ films can be produced in high-density plasmas provided by techniques like electron cyclotron resonance (ECR), inductively coupled plasma (ICP), the plasma beam source (PBS) or electron cyclotron wave resonance (ECWR). The preferred source gas is acetylene rather than methane. The high-density plasmas are achieved by operating at a pressure lower than the usual PECVD and by using magnetic fields to give long electron path lengths, which encourages a high plasma ionization.

Deposition conditions for fullerene-like carbon films should be carefully adjusted because, on the one hand, low or high ion energy is required to achieve a high sp$^2$ fraction, and on the other hand, graphitization of sp$^2$ sites should be prevented, Fig. 9. As a result, a low-energy reactive sputtering or high-energy inductively coupled RF-PECVD with assistant pulsed biasing and temperature control should be employed for film deposition. H-free fullerene-like carbon films were generally prepared by PVD techniques. For instance, pure carbon films have been synthesized by an anodic jet carbon arc, while carbon nitride films have been fabricated by reactive dc magnetron sputtering and pulsed laser deposition. However, H-free fullerene-like carbon films usually exhibit poor friction behavior. So researchers further synthesized hydrogenated fullerene-like carbon (FL–C:H) films by a PECVD technique. These FL–C:H films are found to display an ultra-low friction coefficient of 0.009 under ambient conditions when sliding against Si$_3$N$_4$ materials [8].

2. DEPOSITION TECHNIQUE AND THE PROPERTIES OF FILMS
Studies on formation of FL structured carbon in ultrafast laser ablation experiments and in molecular dynamics simulations have revealed that large, closed and caged structures can be formed from a variety of precursors including the amorphous carbon and even the nanodiamond if sufficient annealing temperature and annealing time are provided [9]. This transformation process was thermodynamically driven on the basis of energy minimization of fullerene structure. Therefore, a preferred method to grow FL carbon films is to create an annealing process, more exactly a periodical annealing, during energetic deposition. This can be realized in a pulse-discharged plasma in which the whole deposition process can be considered as repetitions of primary amorphous carbon ‘growing’ during pulse-on/discharged process and subsequent annealing to FL carbon during pulse-off process, as shown in Fig. 10.

Fig. 10. ‘Growing-annealing’ model for formation of FL nanostructure in periodical discharged plasma

Fig. 11 shows the HRTEM plane-view images of the samples grown under different pulse duty cycle (20%, 50%, and 100%). Ordered domains of several nanometers in size consisting of straight and curved planes can be obviously distinguished for the film grown with duty cycle of 20%, Fig. 11(a), and the layer spacing is approximately 0.34 nm, which is well consistent with the layer spacing of the graphite face (0002). These features are typical of FL arrangements and are more clearly observed at the edge where an individual structure of concentric FL shells can be distinguished. As the pulse duty cycle increased to 50%, similar microstructure could be observed in the films except that the ordering degree became smaller and the ordering range was shortened (Fig. 11 (b)). with the pulse duty cycle further increasing to 100%, namely the continuous discharge, no ordered graphite planes can be clearly detected any more (Fig. 11 (c)) and the film was completely amorphous.
Further evidence can be found from the Raman spectra, Fig. 12. In each Raman spectrum, four subpeaks corresponding to different vibration modes of five-, six-, and seven-membered rings can be extracted, Fig. 12 (b). It is seen from Fig. 12 (c) that the fraction of 5\text{A}1 and 7\text{A}1 increase monotonously with increasing duty cycle, especially from 65\% to 100\%, with the fraction of 6\text{A}1\text{g} varying in an inverse tendency. For pure carbon structures, curvature is induced by the incorporation of odd member rings in an otherwise hexagonal structure. Therefore, the decrease of the fraction of five- and seven-membered rings indicates the reduced curved graphite sheets and FL nanoarrangements in films as the pulse duty cycle increases. Both the HRTEM and Raman results reflect the dependence of formation of FL nanostructures on the annealing process (namely the duty cycle in pulsed discharge). And a relatively low pulse duty cycle, corresponding to a relatively long annealing time, is considered to promote the formation of fullerene-like carbon films. On the basis of this conclusion, we have successfully prepared a series of FL-C:H films in several different pulse-discharged plasmas [10-20].

1). Pulsed dc PECVD
Amorphous carbon films were deposited by pulsed dc PECVD using a mixture of methane and hydrogen as feedstock. A pulsed dc power source was supplied to the unheated smaller
electrode (substrate holder) to control the ion energy, Fig. 13. A base pressure of $8.0 \times 10^{-4}$ Pa was attained in the chamber with a turbomolecular pumping system. The deposition conditions are as follows: (1) applied pulsed dc negative voltage: -1000 V; (2) pulsed frequency: 20 kHz; (3) duty cycle: 0.6; (4) gas flow rate: CH$_4$ = 7.3 sccm and H$_2$ = 20 sccm; (5) deposition pressure: 20 Pa; (6) deposition time: 2.5 h. Prior to deposition, the substrates were cleaned ultrasonically in an acetone bath and dried in N$_2$ followed by etching at a bias voltage of −800 V, duty cycle of 0.6 and argon plasma pressure of 3 Pa for 30 min to remove the native oxide on the Si surface.

**Fig. 13.** The schematic diagram of the pulse DC PECVD system.

**Fig. 14.** HRTEM images of the FL–C:H: (a) plan-view image of the film, the inset is the corresponding diffuse rings; (b) image at the fracture edge.
Fig. 14 shows the HRTEM images of prepared carbon films which deposited on unheated substrates (the substrate temperature was $75 \pm 10^\circ$C). The fullerene-like nanostructure characteristic of curved graphite planes with intervals of approximately 0.34 nm could be clearly observed. The electron diffraction pattern showed three diffuse rings, with the outer two rings corresponding to $\sim 0.12$ and $\sim 0.2$ nm interplanar distance, similar to those of amorphous carbon; and the innermost ring corresponds to the lattice spacing of $0.334 \pm 0.002$ nm, in good accordance with the layer interval of bulk graphite ($d_{002} = 0.334$ nm). At the very edge of the film, Fig. 14 (b), nanometersized structure of concentric fullerene-like shells with the same characteristic lattice spacing ($\sim 0.334$ nm) could be directly distinguished. The HRTEM results clearly indicate that the normally flat sp$^2$-hybridized planes in graphite were buckled and led to basal planes intersecting each other. Thus, the basal planes in this structure were interlocked by covalently linked tetrahedral sp$^3$ bonds to form a three-dimensional network structure. Nanoindentation tests revealed that the prepared FL–C:H films had a hardness of 17-19 GPa, with the elastic recovery up to 85%, Fig. 15.

2). Pulse bias assisted Reactive Magnetron Sputtering
Films with a thickness of 700 nm were deposited using an unbalanced magnetron sputtering technique in a multifunction deposition system, Fig. 16. The substrate holder was placed parallel to the target and the substrate-to-target distance was 150 mm. The vacuum chamber (900 mm in diameter) was evacuated to a base pressure of less than $4.0 \times 10^{-3}$ Pa using a turbomolecular pumping system; the film was deposited at a pressure of $4.0 \times 10^{-1}$ Pa using methane and argon gases as the precursors with flow rates of 60 sccm and 120 sccm, respectively. Substrates were cleaned ultrasonically in acetone and alcohol subsequently before being introduced into the deposition chamber. Prior to deposition, the substrates were subjected to ion bombardment in an argon glow discharge under a negative bias voltage of -1000V for 10 min to remove the native oxide layer. During deposition, the target current was limited at 2.0A at a discharge voltage of 350-400V. The negative bias voltage was generated by a pulse power supply (4 KHz, 15% duty cycle). The films were grown for 60 min and the film thickness was measured by a surface profilometer at a step formed on the film by masking the substrate. Although there was no deliberate substrate heating during deposition, local temperature could rise up to a maximum of 120 $^\circ$C.
The HRTEM image of the prepared FL-C:H film is shown Fig. 17. An image of a-C:H films produced using magnetron sputtering at −800V bias is also given for comparison. Obviously, there were no detectable graphite plane fragments in the HRTEM image of the a-C:H films due to their completely amorphous character. The image of FL-C:H film, however, is quite different. Ordered domains of several nanometres in size consisting of straight and curved graphene planes can be observed. Fig. 18 gives the load–displacement curves of the FL-C:H film and the a-C film deposited by magnetron sputtering of a graphite target. Noticeably, the FL-C:H film has a much higher hardness and elastic recovery (20.9 GPa and 85%, respectively) than the a-C film which has a hardness of 11.8 GPa and elastic recovery of 67%. Tthe excellent mechanical properties should be attributed to the unique fullerene-like structure which is just like a ‘molecule spring’ squeezed in the carbon matrix, reserves the elastic energy during distortion through reversible bond rotation and bond angle deflection. While the amorphous carbon matrix restrains the relaxation of the rigid C–C network and compressive stress, then restricts the slip of graphene sheets.

Fig. 17. HRTEM plan-view images of (a) FL–C:H and (b) a-C:H films.
3). Pulse bias assisted RF PECVD

Films with a thickness of 600 nm were deposited on the AISI 52100 steel substrates by bias-assisted radio frequency (RF) plasma enhanced chemical vapour deposition (PECVD), using a mixture of methane and argon as the feedstock, Fig. 19, where the FL-C:H film was produced by pulsed bias-assistance and the non-FL film was produced by direct current (dc) bias assistance. A base pressure of less than $4.0 \times 10^{-3}$ Pa was attained in the chamber. Prior to deposition of the carbon films, a Si interlayer (150 nm in thickness) was deposited using unbalanced magnetron sputtering so as to improve the adhesion between the carbon films and steel substrates. Then, the carbon films were deposited by a mixture gas of CH4 and Ar with a ratio of 1/1. The deposition pressure was 0.6 Pa and the RF supplier output power was 400W. A pulsed (duty cycle of 20%) bias of $-400$V and a dc bias of $-400$V was applied to the substrate for the FL-C:H film and the a-C:H film, respectively.

![Fig. 19. Schematic illustration of the deposition system](image)
Fig. 20 shows the HRTEM plane-view image of the carbon film grown with pulsed bias. As seen, the ordered domains several nanometres in size consisting of straight and curved planes can be obviously distinguished, and the layer spacing is approximately 0.34 nm, which is in good agreement with the layer spacing of the graphite face (0002). All these features demonstrate the presence of FL arrangements, especially at the edge of the film where an individual structure of concentric FL shells could be clearly distinguished. These FL-C:H film also present better mechanical properties than the a-C:H films without FL nanostructures. As shown in Fig. 21, the FL-C:H has an elastic recovery as high as 80%, much higher than that of the a-C :H film which is only 67%. Similarly, the FL-C :H film has a hardness of ~ 15.6 GPa, much larger than the ~11.2 GPa of a-C :H film.

Fig. 20. HRTEM micrograph of the FL-C:H film grown with pulse bias.

Fig. 21. Typical nanoindentation load–displacement curves for FL-C:H and a-C:H films.

Excellent tribological performance

The unique fullerene-like nanostructure not only endows the carbon films with excellent mechanical properties, but also improves the tribological performance of hydrogenated DLC films including better fracturing resistance under external loads, enhanced anti-oxidization performance in ambient air, reduced friction sensitivity to humidity and independence of friction on counterpart materials [12, 18, 20, 21].

Fig. 22 shows the friction behaviors of FL-C :H and a-C :H films in three atmospheres. Both the FL-C:H and a-C :H films exhibited almost identical super-low friction (< 0.01) in dry N₂,
which is considered to result from two factors, one being that the H-terminated carbon surfaces have fewer tangling $\sigma$-bonds and low interfacial adhesion; the other is that the dry N$_2$ gas can restrain the incursion of O$_2$ and H$_2$O molecules at the sliding interfaces, thus avoiding strong tribochemical reactions (e.g., surface oxidation) of the hydrogenated carbon films. In humid air, however, the FL-C:H film exhibited obviously lower friction than the a-C :H film. The friction coefficient at RH=35% is 0.036 for a-C:H but only 0.018 for FL-C:H. Even at a high relative humidity of 70%, the FL-C:H still holds a low friction coefficient of 0.023, much smaller than the 0.061 for a-C:H. Fig. 22 (b) shows the corresponding wear behaviors of FL-C:H and a-C :H films. It can be seen that, irrespective of the environments, the FL-C:H film always present much lower wear rates than the a-C :H film. In particular, the FL-C:H film had a low wear rate of $9.2 \times 10^{-9}$ mm$^3$N$^{-1}$m$^{-1}$ even at RH=70%. This strongly implies that the friction behavior of FL-C:H films is less sensitive to the water and oxygen molecules in ambient atmosphere as compared with the a-C :H film.

![Fig. 22. (a) Comparison of friction behavior of FL-C :H and a-C :H films in dry N$_2$, air (RH: 35%) and humid air (RH: 70%); (b) corresponding specific wear rates.](image)

![Fig. 23. (a) and (b) 3D micrographs and corresponding cross-section profiles of wear tracks of FL-C:H and a-C:H after tests in air (RH = 35%) for 1 h; and (c) the changes in O/C+O ratio (wt%) on the original and worn surfaces of the FL-C :H and a-C:H films.](image)
Fig. 23 (a) and (b) show the 3D micrographs and the corresponding cross-section profiles of the wear tracks of the FL-C:H film and the a-C:H film after tests in air (RH=35%) for 1 h. The wear track on FL-C:H film is obviously shallower but wider than that of the a-C:H film, which may be due to its good elastic recovery. Fig. 23 (c) summarizes the changes in C and O concentration (wt %) determined by EDS analyses, on the original and worn surfaces of both films. It was observed that compared with the a-C:H film, the O/C+O ratio of FL-C:H film is much lower and only a little increase after friction. This indicates less oxygen adsorption on the original film surface and also slighter oxidation during friction process in humid air for the FL-C:H film. This might be explained from two aspects. Firstly, the first-principle calculations indicate that the energy cost for the formation of dangling bonds in fullerene-like carbon films is considerably higher than that in amorphous CN and pure carbon films. So fewer dangling bonds exist in FL-C:H film and thus leads to the less water or oxygen adsorptions. Besides, from the perspective of energy loss, the energy can be stored elastically and released upon unloading during the friction process, which not only reduces energy loss but also avoids the intense generation of friction heat, thus restraining the tribochemical reactions between frictional pairs or between film and atmosphere molecules of H2O and O2. 

The unique property of fullerene-like carbon films, especial modified by incorporation of functional elements, also leads to a counterpart material-independent ultralow friction performance. Fig. 24 shows the friction behaviors of Ti-doped FL-C:H films sliding against different counterpart materials (Si3N4, Al2O3 and bearing steel) in ambient air (RH ~ 30%) and dry nitrogen (RH < 5%) environments. Generally, the friction coefficient is almost independent of the counterpart material and the test atmosphere. In air, the steady-state friction coefficient varies in the range of 0.01 to 0.015 for different counterpart materials. In dry nitrogen, it further decreases to vary in the range of 0.008 to 0.01.

![Fig. 24. Friction behaviors of Ti-doped FL-C:H film sliding against different counterpart materials in (a) ambient air (RH ~ 30%) and (b) dry nitrogen (RH < 5%).](image)

3. Characterization of films
Apart from observation of fullerene-like nanostructures through HRTEM, plane space calculation by electron diffraction pattern, and elastic recovery by load–displacement curves in nanoindentation tests, fullerene-like carbon films could still be distinguished from a-C:H films without FL nanostructure by Raman, IR and XPS spectra.

1) Raman spectra
Fig. 25 - 27 show the Raman spectra of FL-C:H films prepared in different pulsed plasmas.
As clearly seen, the FL-C:H films prepared with different plasma methods present almost identical Raman scattering curves. In Raman spectra, three characteristic features can be observed: two low/intermediate wave number bands near 400 cm\(^{-1}\) and near 700 cm\(^{-1}\) and a broad overlapped D and G band similar to that of DLC in the region 1000–2000 cm\(^{-1}\). The low/intermediate bands from figure are shown five times magnified in Fig. 25 (a) and 26 (a). The assignment of the bands near 400 and 700 cm\(^{-1}\) is controversial. Roy et al attributed the two peaks to arise from the relaxation of the Raman selection rule due to the curvature in graphene planes [22]. Similar bands also appeared in the Raman spectrum acquired from carbon nano-onions [23]. The bands that appeared near 400 cm\(^{-1}\) and near 700 cm\(^{-1}\) from carbon nano-onions have been assigned to transverse optic and transverse acoustic vibration at the M point [26]. From the HRTEM image of the film, we can see that there are curved graphene planes embedded in an amorphous carbon matrix. These mbedded curved graphene planes can exhibit characteristics similar to carbon onions; therefore, the two low/intermediate wave number bands similar to those from carbon nano-onion are also expected to appear in the film. The Raman bands of the film in the region 1000–2000 cm\(^{-1}\) were fitted with Gaussian functions. Usually, the Raman spectra of amorphous carbon films are characterized by the presence of two peaks in the region 1000–2000 cm\(^{-1}\), which are labelled as the D and G peaks. The G peak at around 1560 cm\(^{-1}\) originated from optical zone centre vibrations (E\(_{2g}\) mode) of all pairs of sp\(^2\) C atoms in aromatic rings and olefinic chains and a shoulder peak at around 1380 cm\(^{-1}\) originated from the breathing modes of sp\(^2\) atoms in clusters of sixfold aromatic rings [27–29]. But in the film we cannot obtain good fits just by considering the G and D peaks. Acceptable fits could only be reached by adding two extra peaks at 1220 and 1460 cm\(^{-1}\). And hence, four Gaussian peak positions are 1220, 1357, 1470 and 1540 cm\(^{-1}\) (Fig. 26 (b)). With reference to the recent study of our research group, the two bands at 1220 and 1470 cm\(^{-1}\) can be attributed to fullerene or onion structure. Meanwhile, the peak at 1470 cm\(^{-1}\) can also be considered as the vibration peak for the C\(_{60}\) molecule. In conclusion, the peaks near 400, 700 and 1220 cm\(^{-1}\) in this work are attributed to the fullerene or onion structure, and the peak near 1470 cm\(^{-1}\) is attributed to C\(_{60}\) and fullerene or onion structure. Considering the presence of curved graphite planes in the films from the HRTEM observations, it is rational to use four vibrational bands to simulate the Raman spectra at 1260, 1380, 1470, and 1570 cm\(^{-1}\), respectively (three with A-type symmetry (from five-, six-, and seven-membered rings) and one with E-type symmetry (from six-membered rings)).

![Fig. 25. (a) Raman spectra of FL-C:H film prepared by dc-pulse PECVD: (1) Raman spectrum, and (2) magnified wave number region from 0 to 1000 cm\(^{-1}\), and (b) deconvoluted wave number region from 1000 to 2000 cm\(^{-1}\).](image-url)
Fig. 26. (a) Raman spectra of FL-C:H film prepared by pulse bias assisted reactive sputtering: (1) Raman spectrum, and (2) magnified wave number region from 0 to 1000 cm\(^{-1}\), and (b) deconvoluted wave number region from 1000 to 2000 cm\(^{-1}\) [13].

Fig. 28. (a) Raman spectra of FL-C:H film prepared by pulse bias assisted RF PECVD, and (b) deconvoluted wave number region from 1000 to 1800 cm\(^{-1}\).

2) IR spectra
Fig. 29 shows the FTIR spectra of FL-C:H, a-C:H and a-C films. The absorption peak at 1380 cm\(^{-1}\) represents the sp\(^3\) CH\(_3\) or (CH\(_3\))\(_n\) stretching mode. The appearance of a peak centred at around 2920 cm\(^{-1}\) indicates that hydrogen is predominantly bonded to saturated (sp\(^3\)) carbon atoms. A strong peak at around 1600 cm\(^{-1}\) can also be observed, it originates from the Raman active G band (graphite) at ~1570 cm\(^{-1}\) and the D band (disorder) at ~ 1360 cm\(^{-1}\), which can be attributed to olefinic or aromatic vibrations of C=C bonds in the film. The result of FTIR analysis indicates that the fullerene-like carbon film is hydrogenated and has a complex configuration consisting of sp\(^3\) olefinic C=C bonds, aromatic C=C bonds and sp\(^3\) C–H bonds. So strong bands centered at both ~1600 and ~2920 cm\(^{-1}\) can be observed in FL-C:H films. In contrast, a-C:H or a-C films just display strong band at either ~1600 or ~2920 cm\(^{-1}\).

3) XPS spectra
The mechanical properties of diamond-like a-C:H films are usually controlled by the sp\(^3\) C-C connectivity, so a-C:H films with high hardness usually have a high sp\(^3\) C fraction. For FL-C:H films, however, the mechanical properties are dependent not only on the amount of sp\(^3\) bonding but also on the sp\(^2\)-coordinated fullerene-like nanostructure which could extend the extraordinary strength of a planar carbon network to three dimensions and prevents
interplanar slip and bond breaking by reversible bond rotation and bond angle deflection. So the dominant bonding in FL-C:H films is sp² rather than sp³. As a result, the XPS C1s peak position for FL-C:H films should be lower than that of the a-C:H films, as seen in Fig. 30 and 31, because the binding energies for C-C and C=C are ~ 285.3 eV and ~ 284.8 eV, respectively.

Fig. 29. Infrared spectra of FL-C:H and reference a-C and a-C:H films for comparison

Fig. 30. XPS C 1s core-level spectra for the a-C:H and FL-C:H films prepared by dc-pulse PECVD

Fig. 31. XPS C 1s core-level spectra for the a-C:H and FL-C:H films prepared by pulse bias assisted RF PECVD
4. SPECIFIC REQUIREMENTS FOR ENGINE COMPONENTS DEPOSITION

Fuel-saving technologies have become more important recently, especially for automobiles, in order to avoid global environmental destruction and resource depletion. Technologies for reducing friction are direct ways of improving vehicle fuel economy [24]. Due to the unique combination of excellent properties including high wear resistance, ultralow friction coefficient, high corrosion resistance, insulation resistance, gas diffusion resistance, and high refractive resistance, DLC coatings have present huge potential as lubricating and protective coatings of engine components in automotive industry [25]. In particular, application of DLC coating on cam followers has led to a super low friction coefficient of 0.006 in ester containing PAO oil [24]. However, due to some inherent drawbacks, deposition of DLC films on engine components, especially on curved surfaces, must be specially processed. Firstly, there is a very large compressive stress in DLC films that exists irrespective of thin film growth technique, such as magnetron sputtering, pulsed laser deposition, ion beam deposition, etc. This large compressive stress, when accumulates to a certain level, causes the film to bulge and peel off from the substrate, thus restricting the applications of thin DLC films. As a result, elimination or minimization of compressive stresses in DLC films offers major challenge for technological applications of DLC coatings. Traditional approaches to obtain DLC films with low internal stress involved increasing deposition temperature or decreasing the energies of carbon species arriving at the substrate surface. Unfortunately, all these are achieved at the expense of reducing $sp^3/sp^2$ ratio. Actually, the elimination or minimization of compressive stresses in DLC films by alloying with metals and the use of metallic interlayers between the film and the substrate have been currently extended to reduce the compressive stress in the DLC films. Fig. 32 presents schematically different possibilities of doping DLC with different elements and metal to modify their nature and properties without changing the amorphous phase [26]. In previous work, we have also deposited DLC films with reduced stress by doping Mo, Ti, S, N and Si as well as employing a pulsed plasma technique to allow film relaxation during growth process [21, 27-31]. However, our further studies reveal that the lowest stress does not necessarily lead to good tribological performance due to the simultaneously decreased hardness. Therefore, the stress of DLC films should be adjusted according to application conditions.

![Fig. 32. Scheme of typical doping elements introduced into the DLC compositions for achieving improved properties [26]](image)

66
Another important issue for application of DLC films on engine components is the adhesion between DLC and substrates. Strong interfacial bonding or adhesion can be attained easily between DLC and carbide- and silicide-forming substrates (such as Si, Ti, W and Cr). The adhesion of DLC coatings to other metallic and ceramic substrates may not be as strong but can be improved by the deposition of an initial bond layer on these substrates prior to DLC deposition. These bond layers are typically selected from those elements that are known to be strong carbide- or silicide-formers such as Si, Ti, Cr, W and Nb. These elements can chemically react with the atoms of the substrate materials and thus insure strong bonding [2].

To improve the adhesion of DLC films on steel substrates, a Cr-substrate intermixing layer was employed by ion bombardment of the substrate surface before depositing a Cr intermediate layer by magnetron sputtering [26]. Using W or WC interlayers on steel substrates, the adhesion of DLC films was found to be strongly dependent on the thickness of sputter-deposited W layers and CH₄ fraction in the gas phase during sputter-deposition of WC layers. Furthermore, Ti/TiN multilayers, and Ti/TiN/TiCN transition layers were also investigated and deposited sequentially to create an interface between DLC films and steel substrates. But Ti interlayers alone fail to significantly improve DLC film adhesion. The Ti/TiN/TiCN transition layer led to significant improvements of the adhesion of DLC films. Gradient a-SiCx intermediate layers have been deposited by magnetron sputtering and RF plasma-enhanced CVD on steel substrates prior to deposition of DLC films. The critical load determined by scratch tests was improved significantly [32].

5. DEPOSITION OF FILMS ON STEEL SURFACES

After optimization of deposition procedure and plasma parameters, we have successfully prepared a series of FL-C:H films on different engine components, as shown in Fig. 33 - 35. The deposited FL-C:H films display excellent adhesion to the steel substrates, with the critical load determined by scratch tests as high as 62 N, Fig. 36. Friction behaviors of components with and without FL-C:H coatings were also investigated. As seen, the FL-C:H coating not only remarkably reduced the friction coefficient to 0.107 from 0.309, but also stabilize the friction behavior during the whole test duration.
Fig. 34. Workpieces with FL-C:H coatings

Fig. 35. Automobile engine components with FL-C:H coatings.
SUMMARY

To sum up, DLC films have emerged as a new class of important lubricating and protective materials in recent years due to their extraordinary high hardness, good chemical inertness, low friction coefficient and wear rates. Especially for hydrogenated DLC films, a superlow friction and wear performance has been observed. However, tribological applications of them are usually restricted due to the high compressive stress produced in energetic deposition processes and the humidity-dependent friction behavior, as well as the inherent high brittleness induced local fracture under external load. Further studies reveal that the mechanical and tribological behaviors of carbon films are not only related to bonding configuration, but also dominated to some extent by microstructure. And introduction of certain carbon allotropes, such as fullerene, nanotube and expanded graphite, could not only decrease internal stress and increase hardness and elasticity of amorphous carbon films, but also modify the tribochemical properties such as enhanced oxidization resistance, reduced friction sensitivity to humidity and independence of friction on counterpart materials.
We have successfully prepared a series of FL-C:H films using several different pulse-discharged plasma deposition techniques, including pulsed dc PECVD, pulse bias assisted reactive sputtering and pulse bias assisted RF PECVD. Irrespective of deposition method, all the FL-C:H films prepared exhibit similar microstructure and mechanical properties (hardness 16-21 GPa, elastic recovery > 80%). Tribologically, FL-C:H was able to provide an ultralow friction and wear performance (with friction coefficients < 0.01 and wear rate $10^{-9}$ mm$^3$/Nm) and also insensitivity to humidity, oxygen and counterpart materials. After careful optimization and design in interfacial adhesion strengthening and internal stress reduction, we have successfully deposited FL-C:H films on various metallic substrates and applied them in the engine components of cars to act as lubricating and protective coatings.

REFERENCES


Chapter 4
Test methods to evaluate diamond like carbon thin films on textured surfaces
Stephen M. Hsu, GWU, USA

Performance Evaluation of DLC films

The goal here is to develop a thin film or coating to protect the surface textures under high load slow speed conditions where wear may damage or eliminate the textures. There are several critical parameters to consider:

1) The nature and composition of the film
2) The thickness of the film
3) The adhesion characteristics of the film to the substrate
4) The hardness of the film in relation to the substrate
5) The compatibility of the film with lubricant chemistry

Adding a film to a complicated textured surface may alter the dimension and shape of the textured dimple; this may alter the friction reduction ability of the textures, so the thickness of the film and coating is a crucial parameter. If the thickness of the film is too thin, then the antiwear durability may be compromised. The nature and composition of the film control the adhesion characteristics of the film to the substrate. Thin films have the ability to conform to the textured surface with minimum residual stresses. A considerable amount of screening of various materials and deposition techniques will be necessary to find the right film that will satisfy all these requirements. There are no known films today capable of meeting all these requirements.

Sources of the films

We have approached many companies and research groups for films and coatings. Some are willing to provide coatings; some are not due to various proprietary reasons. At the end, we obtained film/coatings from Morgan Ceramics, Beamalloy, and several research groups.

Nature of the films

A collection of high hardness wear resistant films were deposited on textured surfaces, these include Chromium nitride, titanium nitride, carbides, and diamond-like carbon films. The film thickness ranges from 90 nm to 200 nm thick. Some samples were coated with one or two micrometer thick. As a rule, the thick films all failed early.

Test procedure development

At the beginning, all films failed rapidly under high load and low speed conditions, some resulting in high friction within a single cycle. As we progress to more tailored-made films, then how to evaluate the durability of the film becomes an issue. A new test procedure was developed to test the degree of protection and the durability of the film.
Table 1. Four ball wear tester test procedure for film durability

<table>
<thead>
<tr>
<th>Load</th>
<th>speed</th>
<th>Load</th>
<th>speed</th>
</tr>
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<tbody>
<tr>
<td>5 kg - 30 kg</td>
<td>1.15 m/s</td>
<td>5 kg - 30 kg</td>
<td>1.15 m/s</td>
</tr>
<tr>
<td>30 kg</td>
<td>0.96 m/s</td>
<td>5 kg - 30 kg</td>
<td>0.96 m/s</td>
</tr>
<tr>
<td>30 kg</td>
<td>0.77 m/s</td>
<td>5 kg - 30 kg</td>
<td>0.77 m/s</td>
</tr>
<tr>
<td>30 kg</td>
<td>0.57 m/s</td>
<td>5 kg - 30 kg</td>
<td>0.57 m/s</td>
</tr>
<tr>
<td>30 kg</td>
<td>0.38 m/s</td>
<td>5 kg - 30 kg</td>
<td>0.38 m/s</td>
</tr>
<tr>
<td>30 kg</td>
<td>0.19 m/s</td>
<td>5 kg - 30 kg</td>
<td>0.19 m/s</td>
</tr>
</tbody>
</table>

The old test procedure is severe in that the test severity increases with every step after it reaches the 30 kg load. Decreasing the speed makes the contact condition more severe and most films fail to finish the single cycle. Since we still have to add the friction-reduction chemistry to the system, too high a severity will not yield reasonable screening results.

Failure mechanisms

Contact mechanics analysis of a dimple suggests that the edges of a dimple under high load low speed conditions induce very high contact stresses, as shown in Fig. 1. Under such stresses, the film covering the dimple will tend to crack under static loading. When shear stresses are added on top of that, the film delaminates. Once the film delaminates, the film debris will act as an abrasive in the contact, hence the friction increases.

Fig. 1. Contact stress around the edge of the dimple

Fig. 2. Delamination of hard film at the edge perpendicular to the direction of sliding

Final selection of wear protective film

After many film failures, a diamond-like carbon (DLC) film was selected. The film thickness ranges from 90 nm to 150 nm and is deposited using a closed field unbalanced magnetron.
sputtering ion plating system. An interlayer was engineered to increase adhesion between the 52100 steel substrate and the DLC film.

![Coated side Uncoated side](image)

**Fig. 3.** SEM image of DLC thin film on steel sample

**Test results**
The DLC film was deposited on baseline 1 (polished sample only), baseline 2 (polished sample coated with the DLC film), baseline 3 (small deep textures with DLC film), multiscale (MS) with DLC film. The results are shown in Fig. 4.

![Comparison of friction coefficient of DLC protected textures with Baseline (BL) under 30 kg load with velocity varying from 1.34m/s to 0.19 m/s speed.](image)

**Fig. 4.** Comparison of friction coefficient of DLC protected textures with Baseline (BL) under 30 kg load with velocity varying from 1.34m/s to 0.19 m/s speed.

With this thin film, the wear protection increases three times. At this stage, we are beginning to understand how a film should be designed to protect the multiscale textured surfaces. It should not be too hard, and it cannot be too thick, and it should be able to resist plastic deformation shear. At this time, the system is not optimized. With compatible surface chemistry to provide additional lubrication protection and friction reduction, durability of this film can be extended significantly.
Chapter 5

Synergistic surface chemistry and durability test procedure

Stephen Hsu, GWU, USA

Working with additive companies and OEMs, we will test and screen various chemistries including solid lubricants, liquid lubricants, and their combinations to achieve the synergistic friction reduction of a multiscale surface texture, protective thin film, and friction reduction surface chemistry to reach the maximum attainable parasitic loss reduction.

Deliverables: A report on the selection process and the nature chemistries tested. Depending on the thin film selected, novel chemistry combining organic molecules and inorganic molecules will be explored.

Introduction

While diamond-like-carbon films are increasingly being used in engine components to control friction and wear, their impact on fuel economy is minimal. This may be due to several factors: conventional DLC films have low surface energy hence have much lower reactivity with friction reducing additives and antiwear additives contained in the lubricant; the film is hard hence it reduces contact area while increase the contact pressure which may increase friction; thickness of film, if not appropriate for the contact conditions, may spall and cause damage.

To introduce this surface texture technology into a production engine, assuming full friction reduction benefits can be achieved; durability of the surface technology is of paramount importance. Previously, we have used a 150 nm thick DLC film, a commercial lubricant, and an oxygenated compound and achieved 10x longer durability in our bench tests. But this approach may not be practical since engine manufacturers do not have any control of lubricant formulations. So in this activity, we will try to use a chemically bonded film (organic and inorganic) to achieve enhanced durability. The concept is simple, deposit mixed monomolecular films on the surface, anneal it to induce chemical bonding, and then coat a thin layer of monomers on top of the chemical film, use UV or heat to polymerize the monomers to form a polymeric canopy on top. This way, the film becomes part of the surface structure and will not be leached out by the lubricant used in the engine and can function independently in an engine.

Since the focus in this activity is on chemical film development and not surface textures, we decided to use a simple circular dimple texture fabricated using microlithography and chemical etching. This texture is easy to fabricate without nanomechanical scratching to create multiscale textural patterns, which is time consuming and would delay the evaluation of chemically bonded film concept. Since the circular dimple would not reduce the friction that much, it would increase the severity of the test procedures to see whether the chemical films can really provide long lasting protection. The DLC films used in this activity is the same as in activity 6, a more graphitic DLC film of 150nm thick, and 12 GPa nanohardness.
Test procedure

Test sample preparation using textured steel disks is time consuming and labor intensive because of the micrometer size of the textural patterns. Chemically bonded films at least initially are also complicated and require careful selection of chemistry, deposition conditions, and thickness control. In order to explore more chemical compounds, we decided to develop a simplified screening test procedure before conducting the normal ball on three flats tests on the Four Ball Test Apparatus. We have a large quantity of DLC coated aluminum based disks (magnetic hard disks used for information storage technology) in our inventory. These disks were then used for chemical deposition, encapsulation, thickness measurement, and tested for friction on a pin-on-disk tester.

Deposition of samples

The chemical compounds were deposited on DLC coated hard disks using a dip-coating technique. Solutions of 0.5% to 2.0% by weight of various compounds were prepared in cyclohexane or hexane. The solutions were mixed for a minimum of 10 minutes in an ultrasonic bath prior to dip coating to ensure a complete solubilization of the compounds. Deposition was carried out in a dipcoater using a dip speed of 40 to 60 mm/min. The resulting films were then annealed at 165ºF for 10 minutes under argon atmosphere to bind the molecules to the DLC surface while avoiding oxidation. After annealing, the films were washed with 10 ml of cyclohexane or hexane, to remove any molecules that were not chemically bound on the surface. The film was then covered with a thin layer of polymeric monomers and then UV irradiated to encapsulate the chemical film underneath. The friction properties of both the encapsulated films and not encapsulated films were tested with the pin on disk machine using a step loading procedure developed under this activity.

In order to simulate the test condition used by the four ball machine, a step loading test sequence was developed. The load was varied from 1 N to 25 N, and the speed was varied from 0.015 m/s to 0.20 m/s to cover the hydrodynamic and boundary lubrication regimes. To extend the load range of the test, two diameters of the pin (balls) were used (1.5mm and 6.125mm). A step loading test was used to evaluate the deposited film. At each sliding speed, 8 different loads were applied and the steady state friction for that film was recorded.

![Frictional characteristics of chemical films as a function of contact pressures](image)

Fig. 1. Frictional characteristics of chemical films as a function of contact pressures
As can be seen in Fig. 1, friction drops from boundary lubrication conditions (0.08-0.15) to 0.03, typical of hydrodynamic lubrication regime. Mixed compounds appear to be able to control the friction over the speed and load range much better than a single compound.

Durability test procedure modifications

These chemical films are much more effective than the added chemicals we tested in Activity 6. So when we test these chemical bonded films according to the test procedure used in activity 6, we cannot observe any failures. In an attempt to reach the time to failure criteria, we modify the test procedure to increase the severity of the test conditions by increasing the load from 20kg, 30kg, and then 40kg but still cannot observe any failures.

Test procedure using a ball-on-three flats geometry of a Four Ball wear tester was modified as follows:

Stage I: Minicycle step loading sequence:

First cycle: start at 1.15 m/s linear speed (3000rpm), increase loads from 5, 10, 15, 20, 25, 30 Kg per every 3 minutes (or longer until a steady state friction trace is obtained). The typical time is 3 minutes for each load.

Second cycle: lower the speed from 1.15 m/s to 0.96 m/s (2500rpm) and repeat the loading from 5 Kg to 30 Kg steps.

For a total cycle, the speed changes from 1.15 m/s, 0.96 m/s, 0.77 m/s, 0.57 m/s, 0.38 m/s, to 0.19 m/s in an increasingly severe test condition (low speed will not allow the textures to generate sufficient hydrostatic lift force, therefore more severe). Minicycle 7 and 8 will basically repeat the condition of cycle 1 and 2.

Stage II: Durability test (time to failure test sequence)

To clearly separate the various chemistries and textures, a time to failure test sequence is used. After the 8 minicycles of testing, if no failure is observed, then the test starts at 1.15m/s speed and 2 Kg load for 3 minutes, the load is increased to 5, 10, 20 Kg. At that time, the test continues for one hour until failure. If no failure is observed, the load is increased to 30Kg for an hour, then 40kg for an hour.
Fig. 2. Test results of various bonded chemical films undergoing the cyclic step loading mini-cycles. Baseline cases failed early as indicated by red arrows. Simple circular dimples were used in #5 and #6. No dimples were used in tests 1, 2, 4. The dimples are effective in high speed low load conditions.

**Durability test results**
Following the above mentioned test procedure, if no failure was observed, the sample was subjected to stage two durability test as illustrated in Fig. 3.

As can be seen from Fig. 2, a typical time to failure friction trace shows a sudden friction increase after a long steady friction level. When data from various test sequence are combined, they can be represented in two figures (Fig. 4 and Fig. 5)
Fig 4. Stage I test results for a specific combination of chemistry and textures, showing no failures for Mobil 1 lubricant and the 3 chemical films while the baseline blank in paraffin oil failed.

Fig. 5. Stage 2 durability test results showing times to failure data for the three bonded chemical films after they have gone through the previous 7000 seconds cycling testing.

As can be seen from Fig. 4 and 5, after the samples have gone through the 7000 seconds of stage I testing, the test continues at 20kg and 3000 rpm speed for an hour. The baseline case using a synthetic motor oil without bonded chemical film failed during the 30kg load at about 13000 seconds. Film 1 continued to 30kg load for one hour, 40kg load for another hour, and 50kg load for an additional hour without failure. Films 5 and 6 exhibited lower friction and appeared to be able to continue for much longer time. We then stopped the test at the end of 30kg load cycle. The concept of built-in bonded film has been successfully demonstrated.
Conclusions
We have developed bonded chemical film technology on DSC surfaces and demonstrated that these films are capable of extraordinary protection of textured surfaces. When combined with multiscale textures, the friction would be much lower for sustained duration.
Chapter 6
Surface representation of textured surfaces

Mingwu Bai, Stephen Hsu, GWU, USA

Background
To reduce the parasitic losses of cars and trucks, surface texturing for friction reduction has been proven as one of the promising technologies. As technology evolves, advanced surface fabrication techniques such as laser texturing [1], microlithography coupled with electrochemical etching [2] have been developed to fabricate sophisticated surface features. How to describe complicated multiscale surface textures to enable manufacturing process control has become an important issue in modern surface technology [3-8].

Surface roughness is the residual unevenness left on the surface after the part has been machined and manufactured. Historically, the machining process and polishing operations are aimed at producing isotropic, Gaussian distributed randomly rough surfaces. These characteristics are important for fitting and mating surfaces together for parts assembly. Since the surfaces are isotropic (no directionality), surface roughness can be easily measured and represented by the average deviation from the mean, such as Ra, (average deviation from the mean of the distance between the peak and the valley of the surface roughness). Ra can be measured by using a stylus instrument (profilometer), the stylus traces the surface peaks and valleys in a single direction and record the vertical distance. In the presence of discrete textures, the surface is no longer isotropic (result depends on the direction of the scan). Because the depths of the dimples are order of magnitudes larger than the average roughness of the untextured surface, a scaling issue emerges. How to measure and how to combine the two different length scales together as a roughness parameter becomes a major challenge.

Under IEA, we have surveyed the literature [3-8] and pooled the participants on surface description techniques on discrete dimple patterns for manufacturing specification control. We found no existing technique that can describe 3-D discrete multiscale textured surfaces. This was somewhat surprising. The international round robin has been organized and is proceeding forward. Meanwhile, we need to develop a new way to describe textured surfaces that are capable of reducing friction.

The surface roughness can be characterized as many parameters, and they are usually grouped into amplitude parameters [3-4], spacing parameters, and hybrid parameters [3]. Amplitude parameters characterize the surface based on the vertical deviations of the roughness profile from the mean line. Many of them are closely related to the parameters found in statistics for characterizing population samples. The typical parameters include average roughness, root mean square, skewness, kurtosis, peak-peak, ten point height, max valley depth, max peak height, surface bearing index, texture direction (texture here means directional roughness within the same order of magnitude). The definitions of these terms are presented in Appendix I.

In order to develop surface parameters to describe textured surfaces for manufacturing specifications, it is necessary to investigate series of 2D surface roughness parameters, and to determine the most sensitive roughness parameters that reflect the presence of dimples.
Problem definition

When a textured surface is scanned with a profilometer, as shown in Fig.1, the roughness at position A, \( R_a(A) \), reflects the roughness of the untextured surface, the profilometer system software automatically filters out the depth of the dimple and measures the roughness at the bottom surface of the dimple at position B. The overall Ra value is then calculated combining the \( R_a(A) \) and \( R_a(B) \) over the length of the scan. In this way, the resulting Ra does not represent the presence of the dimple, the depth and width of the dimple on a 2-D basis. This problem is universal in that all surface roughness measuring apparatus are equipped with automated data analysis software with filtering features. Surface dimples whose dimensions are outside of the range of roughness are not included in the measurement.

![Diagram of surfaces with two dimples. Sections at position A (red circle) and position B (red rectangle) are enlarged.](image)

Technical Approach

To include the depth of the dimple into the roughness calculation, we need to export the raw data from the profilometer and import them into a computer programme for processing. Since most of the roughness parameters are summation of the peaks and valleys of the surface, the final parametric value depends on the frequency of the peaks and valleys. We have developed a method to transform the dimple depths into the roughness calculation. Since all 2-D roughness parameters fundamentally are wave functions, we can take the average wave intervals from the untextured surface to transpose the dimple depths into the roughness calculation. This provides a consistent way to calculate the equivalent roughness (inclusion of dimple depths into the roughness calculation). So the final surface roughness parametric value will include the dimple depth and exclude the dimple bottom roughness. For Ra (defined as the average deviations from the mean height), Fig. 2 illustrates the transformation process.
Fig. 2 (a) Schematic diagram of a dimple with same interval of baseline interval, (b) Transform a dimple into waves with the same interval of interval of baselines.

With above methodology, an example is shown to detail the procedure. As shown in Fig. 3, the sample has dimple diameter of about 83 µm. The dimples were generated by micro-lithographic technique followed by chemical etching of the 52100 steel.

Fig. 3 Optical images of surfaces with dimples. The bar in the figures indicates 20 µm.

Results and discussions

Transformation

We use a surface Profiler (Model: Aplha-Step IQ, Manufacturer: KLA-Tencor). The scan length can be as long as 10 mm, scan speed 2 µm/s – 200 µm/s, vertical range can be ± 10 µm at 0.012 Å, ± 275 µm at 0.24 Å, or 1000 µm at 1.2 Å. It has horizontal resolution of 0.01 µm. Based on the proposed method, the 2-D profile is transformed and redrawn as shown in Fig.4. Matlab programs are written to process the data. The procedure is, as shown in Fig.4, ungroup non-textured sections (green in color) and textures sections (red in color), transform the depth of the dimple as illustrated in Fig. 2 based on the specific roughness parameter definition (formula). Then a composite roughness including the dimple depth can be drawn.
The transformed Ra equivalent roughness in Fig. 4c is different with that of Rpv in Fig. 4e, this is because, during transformation, all dimple’s data were transformed according to the definition of Ra and Rpv. In the case of Ra, the baseline is the mean height, and for Rpv, the baseline is the bottom of the valleys.

**Comparison of 2-D roughness parameters**

In order to compare different roughness parameters in terms of their sensitivity to the presence of dimples, we calculated various roughness parameters based on fixed length of horizontal scan, then include up to four dimples within the fixed length. Table 1 shows composite roughness of transformed roughness.

<table>
<thead>
<tr>
<th># dimples</th>
<th>Lud (µm)</th>
<th>LD (µm)</th>
<th>Lud+LD (µm)</th>
<th>LD/(Lud+LD) (%)</th>
<th>Ra (µm)</th>
<th>Rrms (µm)</th>
<th>Rpv (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>662.5</td>
<td>0</td>
<td>662.5</td>
<td>0.00</td>
<td>0.018</td>
<td>0.002</td>
<td>0.057</td>
</tr>
<tr>
<td>1</td>
<td>579.7</td>
<td>82.8</td>
<td>662.5</td>
<td>13.04</td>
<td>0.299</td>
<td>0.026</td>
<td>0.968</td>
</tr>
<tr>
<td>2</td>
<td>496.9</td>
<td>165.6</td>
<td>662.5</td>
<td>27.27</td>
<td>0.580</td>
<td>0.049</td>
<td>1.878</td>
</tr>
<tr>
<td>3</td>
<td>414.1</td>
<td>248.4</td>
<td>662.5</td>
<td>42.85</td>
<td>0.860</td>
<td>0.073</td>
<td>2.789</td>
</tr>
<tr>
<td>4</td>
<td>331.3</td>
<td>331.2</td>
<td>662.5</td>
<td>60.00</td>
<td>1.141</td>
<td>0.096</td>
<td>3.699</td>
</tr>
</tbody>
</table>

| Ld: length of dimple, Lud: length of untextured surface |
| Dimples: width 82.8 µm, depth 4.61 µm |

Table 1 compares Ra, Rrms, and Rpv as an illustration of the transformed roughness. As can be seen, Rrms has the lowest sensitivity towards the presence of the dimples, Rpv has the highest sensitivity towards the presence of dimples. We will then use Rpv (average peak to valley distance) to describe textured surfaces based on 2-D roughness parameters.
Three dimensional representation

3-D representation of the dimpled surface is needed to define the dimple location, shape, and size of dimples. The locations of the dimples can be determined by using reference points on a textured surface. Because the textures are mostly repeated patterns, a single unit cell area can be used to represent the basic dimple features per unit area.

In order to describe the size and shape accurately, repeated scans in the x direction as shown in the top ellipse in Fig. 5 provide very few points in describing the elliptical shape. So a x, y cross-scan will yield sufficient points to define the shape accurately without increase the scan density. However, current profilometers scan the x-axis with a resolution to 0.010 µm, but the y-axis scan precision is poor. So we added a y-axis mini-stage with a resolution of 0.010 mm (10 µm). This way, both x-axis scan and y-axis scan can be done with sufficient precision. A Matlab program was written to fold the x-axis scan and y-axis scan by selecting common reference points.

After selecting the point of origin, multi-scans of the unit cell was done. Fifty scans in both the x an y axes were performed 50 times by gradually changing the intervals. Then the data were plotted in 3D.

Fig. 6 shows multiple scans in the x direction of a surface with texture, the data were transformed to include the dimple depth into the roughness. In this case, Rpv was used. A 3-D Rpv map was then drawn to show the geometry, size, shape and positions of textures. Fig. 6b, we inverted the dimples so that the internal slope and other features can be viewed easily.
Fig. 7 a) multiple 2-D surface x-axis scan; b) y-axis scan; c) Rpv 3-D map of the surface after the data are transformed. The elliptic dimple diameters are 60µm in the long axis and 15 µm in the short axis.

For the elliptic dimples, there is a significant difference between x-axis scanning and y-axis scanning, shown in Fig. 7a and Fig. 7b. After data processing using a computer program written in Matlab, the 3-D Rpv map was constructed as shown in Fig. 6c. It clearly indicates the elliptical shape and size clearly.

Conclusions

Current existing methods can not describe the surface with textures, because they do not take into account of the multi-length scales of textured surfaces. A new technique has been developed to transform the dimple depth data into a composite surface roughness based on the same wave form intervals. Examples of Ra and Rpv were shown how this technique works and how dimples can be described accurately. Since the detailed mechanisms of how dimples work in friction reduction, representation techniques of surfaces which have demonstrated friction reduction will yield important information for future modeling efforts.

3-D construction of maps using surface roughness parameters can define geometry, size, and shape accurately.

References

Chapter 7
Directional fractal signature methods and unified optimization approach for surface textures

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Abstract
Surface texturing has a potential to be a cost effective and easy way to improve the tribological performance of lubricated interfacing surfaces. Effects of surface textures on the performance have been investigated in the past two decades. However, a limited number of analytical solutions have been proposed, the majority of studies have been experimental and results obtained have been not optimal. This is because the commonly used surface characterization methods are not able to characterize surface textures over a range of scales at different directions and optimization methods used work for relatively simple textures and specific constraints imposed on pressure, film thickness, sliding ratio and lubricant rheology. Previous studies have addressed these issues by developing directional fractal signature methods and unified computational approach for texture optimization. In this paper, recent advancements in the development of fractal methods and optimization of surface textures are presented.

1. Introduction
Surface textures are used as a means of reducing friction, increasing load capacity and wear resistance, obtaining good electrical contact, formability, paintability, specific optical properties and others [1,2]. This has been long known and is beyond question. However, characterization and optimization of surface textures that are essential to achieve the improvements still remain unresolved problems. This is because surface textures have complex spatial arrangements of topographical features ranging from hundreds of micrometers to sub-nanometers. They can be grouped as follows:

- structured: geometric shapes/objects (e.g. dimple, groove, chevron, pillar, finger, pyramids) are located on the surface according to rules which are explicitly defined to obtain particular patterns [1,3],

- self-structured: features are arranged on the surface through multiple interactions among components (molecules) according to certain self-organization rules which are not explicitly tied to any particular pattern [4,5],

- stochastic: spatial relationships of features is governed mainly by probability laws (e.g. Gaussian distribution of heights) [6,7]

- irregular: distribution of features is irregular and does not follow any known pattern, there are no periodic sequences of “valleys” and “peaks” and texture has cloud-like appearance (e.g. peening, shot blasted) [6,7].

1.1. Characterization
Because of the variety and multiscale nature of texture patterns, there is not a uniform standard, a universal method or an optimal method for the numerical characterization of geometry of surface textures. Each method was developed for a particular purpose and has its own advantages and disadvantages [2]. Existing methods for characterization of texture images are non-fractal and fractal. Non-fractal methods characterize local changes in texture and spatial interrelationships between them using statistical and/or structural properties of the surface image. They are roughness parameters such as average roughness $S_a$, peak-to-peak height $S_{\text{max}}$, average roughness asperity spacing $S_m$ (to name a few) routinely used for surface quality control. Studies showed that these methods can characterize accurately many surface textures [6,8]. However, when surface textures do not have periodic/quasi-periodic structure, take “strange” shapes such as lotus-leaf like [9], flora-like crystal [10], fibre mat [11], self-organized nanolayers [4,5] and mutually grafted nanolayers [12], or have roughness and directionality that vary with scales the methods do not work well. The reasons are scale-dependency, i.e. values of texture descriptors they produced depend on the measurement scale, and no measurement of texture anisotropy at different scales. Fractals are a promising alternative approach [7]. Fractal methods are scale-independent, do not need periodicity in texture and quantify texture roughness and directionality at different scales. Recent developments are the calculation of fractal dimensions (FD) in different directions and at individual scales and the construction and use of fractal models of an entire surface texture.

1.2. Optimization

Current approach to the optimization of surface textures in lubricated contacts with aims of minimum friction or maximum load capacity is, in most cases, by ‘trial and error’, i.e. changes are introduced and their effects are studied. The optimization problem has been solved to some extent using numerical optimization methods such as conjugate sequential, sequential quadratic programming [13,14] and genetic [15,16] techniques for relatively simple surface textures and specific constraints imposed on pressure, film thickness, sliding ratio and lubricant rheology. These studies have also showed that different methods handle the same constraints differently. This leads to difficulties when the texture shape optimization is subject to new, complex and/or more realistic constraints. So far this problem has only been partially addressed by modifying the existing methods or developing a new one. Consequently, a proliferation of various ways of solving the optimization problem has been observed and there are great difficulties associated with the selection of an appropriate method. A much needed solution is the development of a general and systematic optimization approach that works within a wide range of constraints for arbitrary surface textures.

In this paper, recent developments and methods used in fractal analyses and optimization of surface textures are reviewed. Directional fractal signatures (DFS) and unified optimization approach are focal points of the review.

2. Characterization of surface textures by fractal methods

The use of fractal methods in the numerical characterization is justified by the fact that topographical features of surface textures exhibit to some extent similarities at different length scales (self-similarity, multi-scale nature). When the methods are applied into the surface image (for example, a range image containing 3D information about surface topography), a single value of FD for the entire texture is calculated. However, the single FD
provides limited information, i.e. the anisotropic nature of texture (changes in statistical characteristic with direction) is not quantified. Therefore DFS methods were developed to rectify the problem by calculating FD at individual scales and directions. Also, a partition iterated function system (PIFS) was developed to encapsulate the entire texture image data. These two approaches will be discussed.

2.1. Surface data presentation

Surface texture data is represented by a digital image of $N_x \times N_y$ pixels, where $N_x$ and $N_y$ are the number of pixels in the horizontal and vertical directions, respectively. Assuming that $L_x = \{1,2,\ldots,N_x\}$, $L_y = \{1,2,\ldots,N_y\}$ are spatial domains $X$, $Y$ and $L_z = \{1,2,\ldots,N_z\}$ is the gray-scale level domain $Z$, the image is an elevation function $z = I(x,y)$ defined on a horizontal plane $(x,y) \in L_x \times L_y$. $z \in L_z$ is the gray scale value, $x$ and $y$ are integer numbers representing coordinates of pixels in $X$ and $Y$ spatial domains and $N_z$ is the number of gray-scale level values.

2.2. Directional fractal signature methods

DFS methods have unique ability to characterize accurately surface roughness and anisotropy in all possible directions at individual scales. This is achieved by calculating a fractal signature (FS) (i.e. a set of FDs at individual scales) in different directions. They are

- FS Hurst orientation transform (FSHOT), and
- variance orientation transform (VOT).

FSHOT method: The method calculates differences in gray scale values of all pairs of pixels within a ring region. The inner and outer radii of the region are chosen by the user according to image sizes and scales of interest. Typical values for 256×256 pixel images are 4 and 16 pixels, respectively (Fig. 1a). As the region moves across the entire image, 1 pixel at a time, all the gray scale differences and the corresponding directions and distances between paired pixels are stored. The direction (a) is defined as an angle between a line running through the pair of pixels and the image horizontal axis (Fig. 1a).

FSHOT method uses the greatest absolute difference between gray-scale values of all pairs of pixels within a ring region. The absolute differences are plotted against between-pixel distances in log–log coordinates (Fig. 1b). The log–log data points are divided into overlapping 5-data point subsets with a line fitted to each subset (Fig. 1c). For a given direction there are subsets/lines in each log–log plot, and for each line the slope and the between-pixel distance corresponding to the central log–log data point are recorded. The between-pixel distance represents the individual subset scale, and the slope of the line being the Hurst coefficient (H). The FD is determined from the Hurst coefficient, that is, $FD = 3-H$.

The Hurst coefficients are plotted using polar coordinates as a function of direction at each scale (image size) and an ellipse is fitted to each plot (Fig. 1(d)). From the fitted ellipses, the following FSs and texture aspect ratio signature ($\text{StrS}$) are calculated:

- $FS_{Sta}$ is defined as the set of FDs calculated at individual surface texture image sizes
in a direction along the roughest part of the texture (i.e., the direction with the highest value FD). This direction is the angle between a line parallel to the horizontal axis of the image and the minor axis of the fitted ellipse, with $F_{S_{ta}}$ defined as $3-S_{ta}$ where $S_{ta}$ is half the minor axis length of the ellipse.

- $F_{SH}$ and $F_{SV}$ are defined as the sets of FDs calculated at individual surface texture image sizes in the horizontal or vertical directions respectively.

- $StrS$ is defined as the set of ratios of the minor axes to the major axes of the fitted ellipses. This measures the degree of surface texture anisotropy at different texture image sizes. $StrS$ values range from 0 to 1, with lower values representing higher surface texture anisotropy.

![Schematic illustration of FSHOT and VOT methods](image)

**Fig. 1.** Schematic illustration of FSHOT and VOT methods (adapted from [17]).

**Variance orientation transform method:** In VOT method, variances of the difference between gray-scale values of all pairs of pixels in each direction are calculated instead of the absolute differences. The fractal dimension is calculated as $FD = 3 - H$, where the Hurst coefficient $H$ is equal to a half of the slope of line fitted to log-log plots.

The FSHOT and VOT methods have been applied to images of artificial images of fractal surfaces and trabecular bone textures obtained from healthy and osteoarthritic knee radiographs [18,19]. The VOT method showed lesser sensitivity to measurement conditions such as image noise, blur, magnification and projection angle and higher accuracy in measuring surface roughness and anisotropy than those obtained for the other methods. Because of this the VOT method was used in subsequent studies [17].

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The VOT method has been applied to 3D real engineering surfaces, i.e. sandblasted surfaces and grounded surfaces. The surfaces can not be reliably characterized by commonly used roughness parameters. This is because values of an average roughness parameter Ra, 3D average roughness parameter (Sa) and fastest decay auto-correlation rate (Sal) obtained for the surfaces are approximately equal [17]. Using the VOT method, statistical significant differences in roughness were found between the surfaces at small and medium scales. It was also found that sandblasted surfaces have significantly lower anisotropy than grounded surfaces at all scales. The VOT method has been also applied to surface textures of adhesive wear particles. Three sets of the wear particles were generated using a pin-on-disk apparatus. The first set contains particles generated by wear occurring for “running in” (a period of high friction coefficient) conditions under low load level. The next set has the particles, but generated for high load. The third set contains particles generated by wear occurring in “steady-state” conditions (a period of low friction coefficient), and load used was low. Surface characterization methods used so far were not able to detect minute changes occurring in the particle textures. This includes highly capable methods such as a combination of Discrete Wavelet Transform and statistical co-occurrence, which had a classification error rate of 33% [20]. The VOT method was able to differentiate between surface textures of the wear particles generated at different operating conditions. Differences were detected at small scales and it was found that changes in load had the greatest effect on particle textures.

2.3. Partitioned Iterated Function (PIFS) system

Another way of surface texture characterization is through the use of fractal model which encapsulates an entire image data [21,22]. The fundamental basis of the model is the self-transformability of surface texture, meaning that one part of the texture image can be transformed into another part of the image reproducing itself almost exactly. This allows an encapsulation of the image data into a set of mathematical transformations, i.e. into a fractal model. The set contains $N$ contractive affine transformations, i.e. $PIFS = \bigcup_{j=1}^{N} f_j(DOM_j)$.

Each transformation $PIFS = \bigcup_{j=1}^{N} f_j(DOM_j)$ converts a larger part of the surface texture image (called domain) $DOM_j$ into a smaller part (called range) $RAN_j$, located elsewhere on the same image, i.e. $f_j(DOM_j) = RAN_j, j = 1, 2, \ldots, N$. An example of the transformation is shown in Fig. 2. The fractal model constructed is called a PIFS. If such a model is obtained for a surface texture image it will contain detailed information including dimple shape, dimple spacing, size and orientation [22].
Fig. 2. Surface texture with marked self-transformable parts. A larger part of the image converts to a smaller part of the image using mathematical transformations containing information about location, scale, translation, rotation, contrast and brightness. A set of these transformations gives the fractal model (PIFS) of the texture image.

When an arbitrary image is applied iteratively to the PIFS, a sequence of decoded images (called intermediate images or transition frames) converging to attractor is obtained (Fig. 3).

Fig. 3. Example of surface texture and its images obtained after decoding the PIFS. At each iteration an intermediate image or a transition frame is generated. The figure was taken from [22].

A focus of our recent work is on use of PIFS models as a surface characterization method for 3D analyses of textured surfaces in hydrodynamic bearings [23]. Decoded image obtained from PIFS model is not an exact copy of the original surface texture, resulting in loss of some texture details, i.e. the details that are not encapsulated in the transformations. Subsequently, when the surface texture encoded in PIFS model is used in analyses of hydrodynamic
bearings errors occur in calculations of load capacity and friction force. To minimize the errors the PIFS model needs to be optimized. In the recent study [23] optimal models were found through an exhaustive search for all possible combinations of parameters such as tolerance, recursive depth, scaling and offset. The models were evaluated using a hydrodynamic parallel pad bearing textured with four different configurations of 64 elliptical dimples (denoted by S1, S2, S3 and S4) having increasing complexity (Fig. 4). Surface texture S1 exhibits dimples aligned along the x-axis direction (Fig. 4a). All dimples on S1 are identical. S2 differs from S1 in that one half of the dimples is deeper than the second half (Fig. 4b). S3 is the same as S1, except that half of the dimples is aligned along the y-axis direction (Fig. 4c). S4 exhibits random dimples of different shapes, depths and orientations (Fig. 4d). For each surface texture a range-image was encoded into the optimal PIFS model and then the model was decoded. This resulted in 8 decoded range images of surface textures. Each surface was separately used in the parallel bearing and pressure distributions were calculated. Differences in pressure distribution between the original image and the corresponding decoded image were calculated and they are shown as 256 gray scale level images in Fig. 5. The black colour represents the value of 0 (no difference) and the white colour stands for the maximum absolute difference equals to 0.2. Pressure values ranged from 0 to 6.

![Fig. 4. Range images of textured surfaces (a) S1, (b) S2, (c) S3 and (d) S4 (adapted from [23]).](image)

![Fig. 5. Difference in pressure distributions between the original images and the image obtained after decoding PIFS: (a) S1, (b) S2, (c) S3 and (d) S4. Darker colour represents smaller differences.](image)

Using the pressure generated in bearings both load capacity and friction force were calculated and percentage differences between the original and decoded images were recorded. Results showed that the optimal PIFS models produced the load and friction that were slightly different (i.e. <2% and <0.04% respectively) from those calculated for the original surface images [23]. This indicates that PIFS is accurate and it would be useful for the characterization of textured surfaces. Further studies are needed to confirm the performance
of PIFS in bearings textured with other patterns and worked under other lubrication regimes (e.g. elastohydrodynamic).

3. Optimization of surface textures

Geometry of surface textures in bearings and seals has been optimized with aims of low friction, minimal wear and high load capacity. Finding the optimal textures is complex and highly nonlinear problem and there is no guarantee that the solution is a global optimum. This is because dynamic behaviours of bearings and seal-like structures are nonlinear and Reynolds or Navier-Stokes equations are required, and also effects of temperature, cavitation, turbulent flow and lubricant properties need to be accounted for and geometry of surface textures can be of any possible shape.

Generally, the optimization of surface textures can be stated as a constrained optimization problem, i.e.: Find the surface texture \( h(x,z) \) that minimizes (or maximizes)

\[
g(h)
\]

subject to the following constraints: the governing nonlinear partial differential equation (PDE), i.e. the Navier-Stokes equations \( NSE(q,h) = 0 \), boundary conditions that pressure vanishes at bearing edges and initial conditions of lubricant velocities, where:

- \( g \) is the objective functional, e.g. objectives commonly used in the bearing design such as a coefficient of friction \( \mu = \iint_{\Omega} \tau dx dz / \iint_{\Omega} pdx dz \), a friction force \( F = \iint_{\Omega} pdx dz \) or a load \( W = \iint_{\Omega} pdx dz \),
- \( q = [u \ v \ w \ p]^T \) is the state vector that contains the lubricant velocity field and the pressure field, respectively,
- \( x, z \) are the planar coordinates,
- \( h(x,z) \) is the texture shape, also called the film thickness,
- \( \tau \) is the sheer stress field, and
- \( \Omega \) is a planar region that represents the bearing surface.

The \( y \) direction is defined through the film thickness \( (y = h(x,z)) \).

Common approaches are based on “trial and error” methods, intuition and observations from nature and they provide solutions using results obtained from experiments, design charts [e.g. 3,24], numerical simulations of system dynamics conducted for various sets of parameters and conditions [e.g. 25-27] and heuristic optimization algorithms [e.g. 15,16]. Although these approaches can produce improved results there is no theoretical guarantee that the results are optimal or even feasible. Also, the approaches require long computational time since the governing equations are solved at each step of optimization.

Another approach is analytical solutions. Necessary and sufficient conditions for the optimality are calculated using the calculus of variation and optimal surface textures are found. However, due to the fact that the dynamic equations are complex the approach
produced optimal textures for only simple cases such as step bearings and specific constraints imposed on pressure, film thickness, sliding ratio and lubricant rheology [28,29].

In light of the above difficulties computational mathematical approaches appear as the method of choice for texture optimization. Recent studies base on a sequential quadratic programming (SQP) method. In the method a quasi-Newton algorithm is employed to solve the first-order necessary conditions (a gradient of the Lagrangian function vanishes to zero). Optimal solution is found by solving a sequence of subproblems. Each subproblem is the minimization of a quadratic approximation of the Lagrangian function subject to a linear approximation of the constraints. The method was used to optimize the groove geometry of thrust air bearings for various objective functions such as bearing flying height, surface friction torque, dynamic stiffness, a product of the height and stiffness, and a ratio of the torque and stiffness [30]. The groove was described by a third degree of spline function and the film pressure was obtained from Reynolds equation. In other study, a 2D slider bearing the film thickness was represented by a polynomial [31]. Parameters (called design variables) of the polynomial were optimized with the aim of minimizing the coefficient of friction (at given minimum film thickness) subject to pressure, shape, load and center of pressure constraints. Dynamics of the bearing was governed by Reynolds equation coupled with a stress field. For SQP the user is required to supply values of the objective function and constraints, as well as their gradients. In all previous studies, the gradients were not provided explicitly to the SQP solver, instead finite-difference estimates were used. However, a choice of the perturbation vector that gives accurate estimates is highly non trivial [32]. And also, if several and more optimal parameters of surface textures need to be found the optimization task becomes increasingly time-consuming and error-prone. One exception could be the study conducted by Ostayen [33], who calculated analytically the gradients. However, it was not shown/discussed whether the gradients were used in actual optimization.

The above studies have showed that different methods handle the same constraints differently. This leads to difficulties when the surface texture optimization is subject to new, complex and/or more realistic constraints. So far this problem has only been partially addressed by modifying the existing methods or developing a new one. This approach results in a proliferation of various ways of solving the optimization problem and great difficulties associated with the selection of an appropriate method. Therefore, a much needed solution is the development of a general and systematic optimization approach that works within a wide range of constraints for arbitrary surface textures.

A first step in the development of such general approach is a unified computational approach based on optimal control [34]. The underlying idea is to formulate the surface texture optimization problem as a combined optimal control and optimal parameter selection problem. For 1D cases the combined problem is:

Maximize (or minimize) an objective functional

\[
G_0(u,z) = \phi_0(x(t_f),z) + \int_{t_i}^{t_f} g_0(t,x(t),u(t),z)dt
\]

with respect to the control function \( u(t) \in R \) and the system parameters \( z = [z_1,\ldots,z_n] \in R^n \), subject to

- the system dynamics
\[ \dot{x}(t) = f(t, x(t), u(t), z) \]

- the initial conditions
\[ x(t_s) = x^0(z) \]

- all-time control constraints
\[ \alpha_k u(t) + \beta_k \geq 0, k = 1, \ldots, n_g \]

- the canonical form constraints
\[ G_k(u, z) = \phi_k(x(\tau_k), z) + \int_{\tau_k}^{t} g_k(t, x(t), u(t), z) dt \geq 0, k = 1, \ldots, n_{gc}, \text{ and} \]

- the system parameter only constraints
\[ G_k(z) \geq 0, k = 1, \ldots, n_{gz} \]

where \([t_s, t_f]\) is the time interval, \(x(t) = [x_1(t), \ldots, x_n(t)]^T \in R^n\) is the state function vector, \(f = [f_1, f_1, \ldots, f_n]^T\) is a vector of functions, \(\tau_k \in (0, t_f]\) is the characteristic time associated with the constraint \(G_k\), \(\alpha_k\) and \(\beta_k\) are scalar parameters, \(g_0, g_k, \phi_0\) and \(\phi_k\) are scalar functions, \(n_z, n_r, n_{gc}, n_{gl}\) and \(n_{gz}\) are number of system parameters, state variables, canonical constraints, all-time control constraints and system parameter only constraints, respectively.

In the new unified approach proposed the correspondences between the shape optimization and the combined problem are employed by replacing:

- spacial variable with time \((x = t)\),
- film thickness with control signal \((h(x) = u(t))\),
- integration constants and film shape parameters with system parameters,
- load or friction with objective functional,
- Reynolds equation with system dynamics, and
- pressure boundary conditions with all-time control and canonical form constraints.

Let the control function be parametrized by a weighted sum of basis functions with parameters \(\{\sigma_j, j = 1, 2, \ldots, k\}\)
\[ u(t) = \sum_{j=1}^{k} \sigma_j B_j(t) \]

where \(B_j(t) = \begin{cases} 1, & t_{j-1} \leq t \leq t_j \\ 0, & \text{otherwise} \end{cases} \) is a piecewise constant function defined over a set of knots \(\{t_s = t_0, t_1, \ldots, t_k = t_f\}\). Once the control signal is parametrized (called control parametrization) the objective functional and all the constraint functionals become functions of the parameter vector \(\Theta = [\sigma_1, \ldots, \sigma_k, z_1, \ldots, z_n]^T \in R^{n_p}\) and the combined problem becomes a nonlinear mathematical programming problem (NLMP), i.e.
\[ \min_{\Theta} \bar{G}_0(\Theta) \]
subject to the constraints

\[ \bar{G}_i(\Theta) = 0, \quad i = 1, 2, \ldots, n_c \]
\[ \bar{G}_i(\Theta) \geq 0, \quad i = n_c + 1, \ldots, n_{gc} \]
\[ u_j^L < \sigma_j < u_j^U, \quad j = 1, \ldots, k \]
\[ z_k^L < z_k < z_k^U, \quad k = 1, \ldots, n_z \]

where \( u_j^U, z_k^L \) and \( u_j^L, z_k^U \) are the lower and upper limits of the control signal and the system parameters, respectively. The NLMP can be efficiently solved using existing software packages.

Our initial study showed that the above approach/method works for 1D cases for which the Reynolds equation can be transformed into a set of ordinary differential equations (ODEs) [38]. In this study, a partially textured infinitely long parallel bearing was optimized for the maximum load capacity. As an example, the optimal solution of a partially textured parallel bearing is shown in Fig. 6. The film thickness of the bearing is given by

\[ h(x, \xi, \varepsilon) = \begin{cases} \xi & l_0 < x < l_1 \quad \text{and} \quad \text{mod}(x - l_0, \varepsilon D/((l + \varepsilon)m - 1)) \\ 1.0 & \text{otherwise} \end{cases} \]

where \( \xi = h/h_m \) is the dimple height ratio, \( \varepsilon = l_D/l \) is the dimple length ratio, \( m \) is number of dimples and \( D \) is the length of textured portion.

Fig. 6. Example of the geometry of the partially textured parallel bearing optimized for the maximum load carrying capacity. The optimal ratios of \( \varepsilon_{\text{opt}} = 1.52 \) and \( \xi_{\text{opt}} = 0.15 \) were calculated for the bearing textured with two dimples \( m = 2 \) and the untextured portions \( l_0 = 0.5 \) and \( l_1 = 0.125 \). This result agrees with data published in [28].

Further development of the approach includes the optimization of surface textures in hydrodynamic contacts governed by 1D and 2D Navier-Stokes equations. The ultimate goal is a the development of unified computational approach for the optimization of 3D hydrodynamic and elastohydrodynamic lubrication contacts for any geometry of texture shapes and any lubricant rheology.
Reference


Chapter 8

Conclusions

This report summarizes the work conducted in Annex IV describing the progress made in the Annex. While the work is continuing for the next two years, the essential basic research has been completed. Now the task moves on to resolve the translational challenges of moving the technology from laboratory to practice.

The report integrates modeling, design, representation providing a perspective that is both wide and deep, characteristic of a multidiscipline research effort. In this case, international cooperation proves essential in garnering the talents of leading scientists in four continents to reach this stage.