TRIBOLOGICAL DESIGN OF MACHINE ELEMENTS

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TRIBOLOGICAL DESIGN OF MACHINE ELEMENTS

edited by

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Introduction

The fifteenth Leeds-Lyon Symposium on Tribology was held from 6th-9th September 1988 at Bodington Hall, The University of Leeds. On previous occasions each Symposium has focused attention on a current and significant research topic, usually reflecting the interests of the Leeds or Lyon research groups, but this time the vitally important subject of technology transfer was recognized. Delegates appeared to appreciate this rare opportunity to discuss the impact of their studies upon machine design, since some 154 of them from 21 countries attended the Symposium on the "Tribological Design of Machine Elements". We were particularly pleased to welcome a strong group of friends from INSA, Lyon, led by Professor Maurice Godet.

The Symposium was dedicated to the late Professor F T Barwell, who died on 15th January 1988 after a long illness. Freddie Barwell was a gentleman. He did much to promote a sound engineering approach to tribology and his book on "Bearing Systems - Principles and Practice" reflected his interest in applying tribological knowledge to bearing design. We know that he would have approved the topic chosen for the 15th Leeds-Lyon Symposium. He regularly attended and contributed to the Symposia and will be sorely missed.

The Symposium opened in customary style with the Keynote Address on the Tuesday evening. On this occcasion we were pleased to welcome Professor Herb Cheng of Northwestern University, Evanson, USA, to set the scene, with a wide ranging and personal account of his view of "The Tribological Design of Machine Elements". Delegates then travelled to York to enjoy the Symposium Dinner in the ancient Merchant Adventurers Hall. The Guest of Honour at the dinner was Peter Jost, who took the opportunity to comment on the coming of age of tribology. Peter himself chaired the Working Party which produced the now famous report which introduced the word tribology some twenty one years ago.

The somewhat unusual nature of the Symposium provided the organizers with an opportunity to offer a perspective on the current state of tribological design in a number of major industries. This was achieved by inviting selected authors to present Review Papers on Tribological Design in;-

- **The Aerospace Industry** — **(J A Dominy, Rolls Royce Ltd, Derby U K)**
- **The Railways** — **(C Pritchard and T G Pearce, British Rail, Derby, U K)**
- **The Automobile Industry** — **(P A Willermet, Ford Motor Company, Dearborn, U S A)**
- **The Process Industries** — **(J D Summers-Smith, Guisborough, U K)**
- **The Power Generation Industry** — **(P G Morton, GEC Stafford, U K)**
- **The Nuclear Industry** — **(T C Chivers, CEGB, Berkeley, UK)**
- **The Spacecraft Industry** — **(R A Rowntree, E W Roberts and M J Todd, National Centre of Tribology, Risley, U K)**
- **The Electronic Industry** — **(E A Muijderman, A G Tangena, F Bremer, P L Holster, A V Montfoortand, Philips, Eindhoven, The Netherlands)**

Information Storage and and Retrieval **(B Bhushan, IBM, San Jose, USA)**

We are particularly grateful to the authors of these Review Papers for preparing these commentaries on contemporary practice in tribological design.

Some sixteen working sessions were included in the Programme and this necessitated the holding of parallel sessions throughout the morning of Thursday 8th September. The fifty four papers presented nevertheless represented only about fifty percent of those offered, thus facing the organizers with the difficult task of declining many attractive offers. Two sessions were devoted to the Review Papers mentioned earlier, while others dealt with;- Seals; Cams; Belts; Gears; Rolling Element Bearings (2); Plain Bearings (2); Wear; Ceramics; Hydrostatic Bearings; Information Storage and Retrieval/Magnetic Bearings and Knowledge Based Systems. The latter sessions reflect the growing interest in the role of tribology in the computer based information society of the 1980's. We are particularly grateful to the distinguished Chairmen who presided over the Symposium Sessions and whose names are recorded in this volume.

Parallel sessions were also introduced into the Social Programme held on the afternoon of Thursday 8th September. The intricacies of the arrangements were described in a specially arranged Wednesday evening session, which is rapidly becoming an established feature of the Symposia in Leeds, by Mr Brian Jobbins. All delegates were taken to Whitby, where the famous explorer Captain Cook learned his seamanship, but this was achieved by following one of three routes. Most of the delegates travelled by coach to Pickering and then crossed the North Yorkshire Moors on the Railway operating on the line originally built by George Stephenson in 1836. Smaller groups visited Kilburn village to see the hand carving of the famous Thompson (Mouseman) oak furniture, or the Fylingdales Early Warning Radar Station. The journey home was broken for dinner at the Crown Hotel, Boroughbridge after a very full day. As far as we know all delegates returned to Bodington Hall!

Initial discussion of the arrangements for each Leeds-Lyon Symposium usually commences some eighteen months to two years before the Symposium is held. The Proceedings are then published within the following year. This rolling organizational cycle of some three years duration calls for considerable dedication and support. We are particularly grateful for the financial support for the Symposium generously provided on this occasion by:

[1] **British Petroleum Research Centre, Sunbury-on-Thames, UK**
[2] **Fiat Research Centre, Turin, Italy**
[3] **Michell Bearings plc, Newcastle-upon-Tyne, UK**
[4] **SKF Engineering and Research Centre, The Netherlands**
[5] **The US Army Research Development and Standardisation Group, UK**

The Symposium literature was once again presented to delegates in handsome wallets provided by Elsevier, Publishers of the Journal WEAR, and we are pleased to acknowledg this most welcome support.

The smooth running of the Symposium owes much to the enthusiasm and hard work contributed by colleagues in the Institute of Tribology in

The University of Leeds. We would particularly like to express our appreciation to Mrs Sheila Moore, Mrs Catharine Goulborn, Mr Stephen Burridge, Mr Ron Harding, Mr Brian Jobbins, Mr David Jones, Dr John Fisher, our technicians and our current research students and fellows. We are also most grateful to the staff of Elsevier Science Publishers BV, Amsterdam, for their professional and friendly service in producing the volumes of Proceedings of the Leeds-Lyon Symposia.

As we were undertaking the editorial work on the present volume of Proceedings, we heard with great sadness of the death on February 15th 1989 after a long illness of one of our editorial colleagues, Professor Daniel Berthe. Daniel was a great enthusiast for this Anglo-French cooperation and we know that the biennial arrangements in Lyon depended heavily upon him and Professor Godet. His contributions to tribology and his intellect were respected and appreciated by colleagues and friends in Lyon and Leeds. On behalf of all our delegates and his friends in Leeds we would like to send our deepest sympathy to Daniel's family and his colleagues in Lyon. The Leeds-Lyon Symposia on Tribology which Daniel Berthe helped to establish have covered a wide range of topics since their inception in 1974, as illustrated by the following list.

[1]	1974	(Leeds)	**Cavitation and Related Phenomena in Lubrication**
[2]	1975	(Lyon)	**Super Laminar Flow in Bearings**
[3]	1976	(Leeds)	**The Wear of Non-Metallic Materials**
[4]	1977	(Lyon)	**Surface Roughness Effects in Lubrication**
[5]	1978	(Leeds)	**Elastohydrodynamics and Related Topics**
[6]	1979	(Lyon)	**Thermal Effects in Tribology**
[7]	1980	(Leeds)	**Friction and Traction**
[8]	1981	(Lyon)	**The Running-In Process in Tribology**
[9]	1982	(Leeds)	**Tribology of Reciprocating Engines**
[10]	1983	(Lyon)	**Numerical and Experimental Methods in Tribology**
[11]	1984	(Leeds)	**Mixed Lubrication and Lubricated Wear**
[12]	1985	(Lyon)	**Mechanisms and Surface Distress**
[13]	1986	(Leeds)	**Fluid Film Lubrication - Osborne Reynolds Centenary**
[14]	1987	(Lyon)	**Interface Dynamics**
[15]	1988	(Leeds)	**The Tribological Design of Machine Elements**

The 16th Leeds-Lyon Symposium on Tribology will be held in Lyon, France, under the title "Mechanics of Coatings" from Tuesday 5th to Friday 8th September 1989. We greatly look forward to meeting our friends in Lyon once again.

Duncan Dowson
Chris Taylor

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SESSION I
KEYNOTE ADDRESS

Chairman: Professor D Dowson

Paper I(i) The Tribological Design of Machine Elements

Paper I(i)

The tribological design of machine elements

H. S. Cheng

History of tribology indicates that many important tribological concepts and theories were stimulated by the needs of new machinery developments, and the new tribological research findings, in turn, have helped to upgrade the design of more efficient and more reliable tribological elements. This mutual dependence is illustrated with some classical examples.

Discussions are then given to the tribological design process of generic elements and its application to determine tribological performance of machine elements, and to the needs in improving this design process to meet the requirements of future machineries in aerospace, automotive, and information processing industries.

1. INTRODUCTION

Good tribological design has often been recognized as the key to the success in developing new machineries. This was true in the days of Reynolds when railroads relied heavily on good journal bearing design of the axles, true in the days of Model T when automotive engines depended strongly on good design practice of main shaft and connecting rod bearings, and is still true today when computers rely critically on good tribological performance of the head/disc interface.

It is now over a century since Reynolds first established the formulation of fluid film lubrication controlling the tribological performance of sliding surfaces. In this past hundred years, a vast amount of design data has been generated in predicting the average lubricant film thickness between the sliding surfaces generated either hydrodynamically or hydrostatically. In some cases, an accurate prediction of the average lubricant film is sufficient to ensure a good tribological design of mechanical components. However, in most other cases, particularly for counterformal contacts, the lubricant film thickness is insufficient to ensure a good design because film thickness alone cannot predict the lubrication breakdown leading to sliding failures by scuffing and wear. It is the lubrication breakdown of the asperity oil film and the surface film which control the failure process. The lack of accurate predictions of the asperity oil film and the surface film breakdown is seen to be the weakest link in the tribological design process.

As new technologies emerge in aerospace, energy, manufacturing, and communication industries, machineries are required to operate under much higher temperature with higher efficiency and reliability. These requirments present new challenges in research and design of tribological elements to discover new lubrication concepts, new materials, new surface modification techniques, and new predictive theories to develop the advanced machineries.

In this paper, some significant historical developments in tribology are used to illustrate the mutual dependence of tribological design and machinery development. Weak areas in current methods of tribological design of machine elements are then indicated and discussed. Some newly emerged tribological concepts for meeting the new challenges in tribological design are also described.

1.1 Historical Developments

In tribological design, one is mostly concerned with two basic elements, the conformal sliding bearings and counterformal rolling and sliding contacts. Historically, evidences of conformal slidings can be traced back thousands years ago when iron journal bearings were used in olive crushing machines in Greece and bronze journal bearings were used as the wheel bearings in the Chinese South Point Chariot[1].

Few would disagree that the most significant historical development in design of sliding bearings is the discovery of a continuous oil film in a journal bearing by Tower[2] and the subsequent derivation of the Reynolds equation[3] which established the foundation for prediction of tribological performance for design of sliding bearings. This most important development was motivated by the problems associated with the journal boxes in railcar axles. Thus, it is a classic example in which a critical need in machinery development triggered a major breakthrough in tribological design which in turn benefits the development of many other future machineries.

Tribological design of rolling and sliding contacts has been evolved mostly from the needs of better rolling element bearings and gears in machinery development. Unlike the sliding bearings for which rational design was established around the turn of the century, there have been no rational criteria for designing the rolling and sliding contacts until almost half a century later when Blok[4] established the critical temperature concept to predict the scuffing threshold and Lundberg and Palmgren[5] established a method to predict the contact fatigue life.

The importance of lubricant film in rolling and sliding contacts was recognized, but satisfactory prediction of lubricant film thickness was not possible until the developments of elastohydrodynamic lubrication[6]. While the detailed influence of lubricant film thickness in contact failures is still being assessed by researchers, its role as an indicater of the severity of asperity contacts is firmly placed in tribological design.

Tribological design of very low speed and heavily loaded sliding contacts depends on the protection of boundary films. Historically, the ties between boundary lubrication and machinery development are less evident than hydrodynamic and elastohydrodynamic lubrication from the first publication on boundary film by Hardy[7]. Nevertheless, these films are the last line of defence against severe wear. Unless rational methods are formed in predicting the conditions of breakdown of these boundary films, the process of tribological design will always be incomplete.

2. TRIBOLOGICAL DESIGN PROCESS

There are two types of elements in tribological design. The first type is simple generic elements involving either conformal sliding surfaces or the counterformal rolling and sliding contacts, as shown in Fig. 1. The second type is tribological machine elements which include all bearings, gears, cams, and other rolling and sliding components. A brief discussion is given to the tribological design process for each of these two types of tribo-elements.

2.1 Generic Tribo-Elements

The processes for designing a generic tribo-element, as illustrated in Fig. 2, begin with the input data, which can be arranged in the following five groups:

1. Geometrical Data, such as the principal radii of the contacts,
2. Roughness Data, such as the average roughness height, average asperity radii, etc.
3. Operating Conditions, such as range of load, speed, temperature, etc.
4. Lubricant Properties, such as viscosity, shear modulus, limiting shear stress, thermal conductivity, etc.
5. Material Properties of Solids, such as elastic and shear modulus, thermal conductivities, specific heat, yielding stress, fracture toughness, hardness, etc.

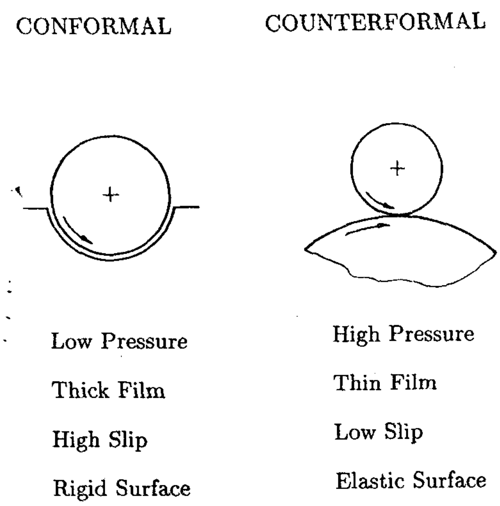


Fig. 1 Generic Tribo-Elements

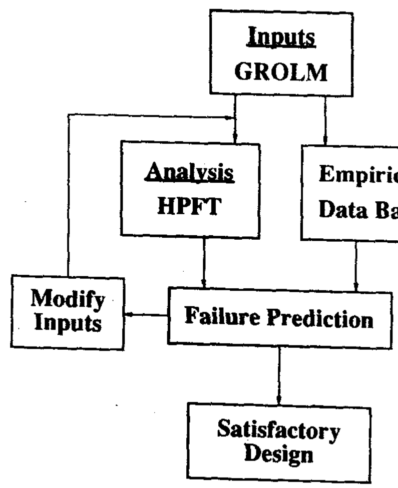


Fig. 2 Design Process for a Generic Tribo-Element

One may use a subroutine named "GROLM" to identify the five major groups of input data needed to initiate the tribological design process.

At the present time, most of these input data is entered manually in the computer programs for tribological design. This inefficient method of handling the input data can be improved if the lubricant and material database is computerized and interfaced directly with the tribo-design program.

The second step in the design process is to determine the tribological performance described by:

1. The film thickness, H
2. The contact pressure, P
3. The friction force, F
4. The contact temperature, T

A subroutine named "HPFT" may be used to identify the calculation of these tribological performance variables which are used to determine whether the element would fail under the given operating conditions.

For contacts operating in thick film lubrication, the distributions of film thickness, pressure, friction, and temperature can be predicted by theories ignoring the surface roughness effects. Indeed, the smooth-surface theories in hydrodynamic, hydrostatic, elastohydrodynamic lubrication have been fully developed in the past century to enable the prediction of these quantities to a high degree of accuracy. Reviews of these developments can be found in the Proceedings of the 1986 Leeds- Lyon Symposium on Tribology[8].

For contacts operating in thin film lubrication, the sliding asperities are no longer separated by a thick oil film. They are protected by a very thin oil film or by a surface film. Tribological performance in the thin film regime, as represented by average quantities of the film thickness, lubricant and asperity pressure, lubricant and asperity shear stress, and surface temperature, $\bar{h}$, $\bar{p}$, $\bar{p}_a$, $\bar{\tau}$, $\bar{\tau}_a$, $\bar{T}_s$, shown in Fig. 3, may not be sufficient to predict tribological failure. There is a need to extend the tribological performance to include the characteristics of these quantities at the asperity level considering each asperity as a micro-contact. These micro-quantities are labelled as asperity film thickness, contact pressure, shear stress, and surface temperature, h^*, p^*, τ^*, T_s^*, as shown in Fig. 3.

Some analyses of tribological performance in thin-film lubrication in terms of the above described quantities are available[9], but the area has not been fully developed. Moreover, the relation of these performance variables with the major tribological failures such as contact fatigue, scuffing, and wear are not fully understood. They are yet to be identified and quantified. Considerable more efforts are needed in this area.

The most important step in tribological design is failure prediction. Ideally, failure predictions should be based on analytical failure models which relate a major failure mode to certain critical tribological variables, such as the relation between contact fatigue life and a critical maximum Hertzian pressure, and the relation between scuffing and a critical total contact temperature. Unfortunately, analytical predictions of sliding failures based on calculated tribological performance variables in the thin-film regime are not always accurate and reliable. One cannot predict analytically the tribological failures as accurately as the structural failures. This most critical block is also the weakest link in the tribological design process.

In the case there are no accurate analytical failure models available, alternative methods, based entirely or partially on failure database obtained experimentally, may be used.

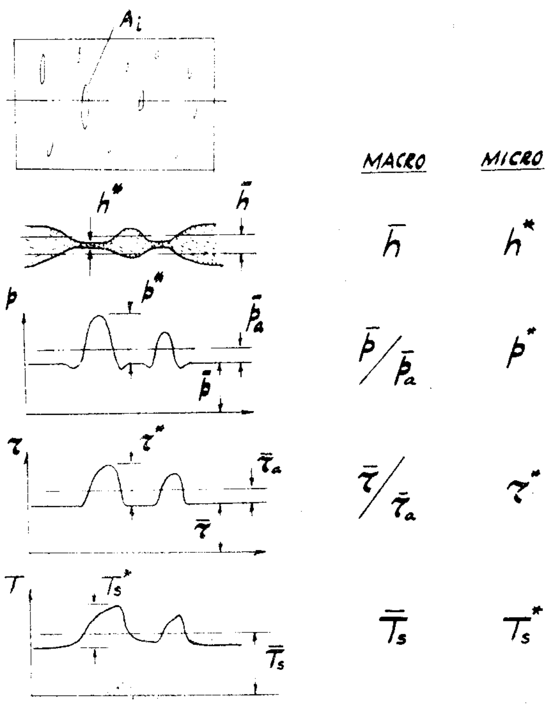


Fig. 3 Average and Micro-Contact Variables in Thin-Film Lubrication

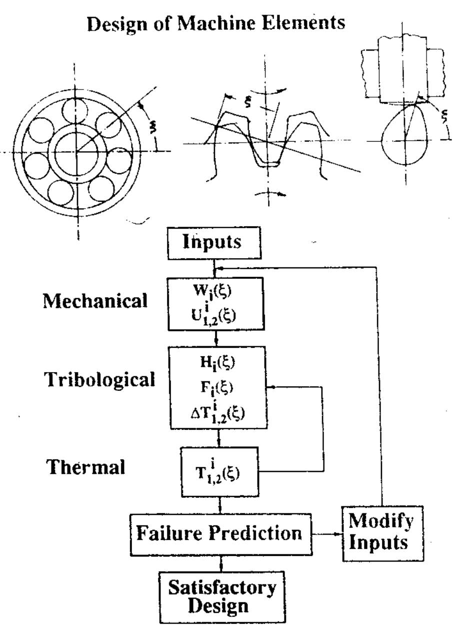


Fig. 4 Block Diagram for Design of Tribological Machine Elements

2.2 Tribological Machine Elements

Fig. 4 shows a block diagram illustrating typical steps for designing a machine element, such as rolling bearings, gears, and cams. As shown in the top figures, each of the elements may contain many generic tribo-elements like EHL contacts which may interact dynamically and thermally.

The design process begins with entering all the geometrical, roughness, operating, lubricant, and material property data, as described earlier for the generic elements. The first step is to determine the cyclic dynamic loads on all the interacting generic elements. In this block, the symbol $W_i(\xi)$ denotes the dynamic load for the ith contact; ξ denotes a space coordinate tracing the contacting path; and $U^i_{1,2}(\xi)$ denotes the surface velocities of the ith contact. The tribological performance together with the bulk surface temperatures $T^i_{1,2}(\xi)$ are determined for all the interacting elements simultaneously by an interactive process. The calculated tribological performance is used to predict failure for each element. A satisfactory design is obtained if no tribological failure is found for all tribo-elements.

The weakest link in this design process, as described earlier, is the lack of reliable analytical failure predictive tools for the tribo-elements. A second weak link in this process is the calculation of the bulk surface temperatures $T^i_{1,2}(\xi)$. The reliability of these predictions is dependent on how close one can predict the surface convective coefficients around all tribo-elements. A slight error in the bulk surface temperature predictions can lead to considerable uncertainties in tribological failure calculations.

3. CURRENT NEEDS

In reviewing the tribological design processes, one can identify a number of weak areas where continuous research is needed to improve the design process. These include:

1. Prediction of Tribological Performance in Thin Film Lubrication
2. Analyticl Models of Contact Fatigue Life
3. Modelling of Scuffing Failure
4. Wear Modelling in Lubricated Contacts
5. Prediction of Bulk Surface Temperature
6. Computerized Tribological Design System

Brief descriptions are given to each of the above listed areas.

3.1 Prediction of Tribological Performance in Thin-Film Lubrication

In thin-film lubrication, the tribological failure is likely controlled by events at the asperity contacts. The distributions of asperity lubricant film thickness, pressure, temperature, and shear stress are pertinent information for establishing failure criteria in this regime. These quantities are, of course, dependent upon the surface roughness topography represented by a set of parameters such as average height, asperity radius, skewness, etc.

At present, there have not been many efforts in studying these distributed quantities at the asperity contacts. For example, the distribution of asperity oil film or micro-EHL film can be calculated for rough EHL contacts by using the classical EHL theories for distributed asperities; however, this has not been fully treated. Computer simulation of asperity pressure and temperature have been achieved by Lai and Cheng[10] for randomly generated rough surfaces and by McCool[11] based on Greenwood and Williamson's model[12] for spherical asperities. However, the use of these methods for determining the characteristics of the asperity pressure and temperature distributions is yet to be achieved.

3.2 Analytical Modelling of Contact Fatigue

In the past three decades, life prediction in lubricated contacts has relied heavily on the Lundberg-Palmgren formula[5] derived from a set of extensive bearing life tests relating the maximum orthogonal shear stress and stress volume to life. Successes in this area include Li, Kauzlarich, and Jamison's application of L-P's relation to asperity contacts to investigate the effects of partial-EHL to contact fatigue life[13], and Harris and Iaannides' recent extension of L-P's model to include the effects of the entire stress field[14]. Fig. 5 shows the predicted trend

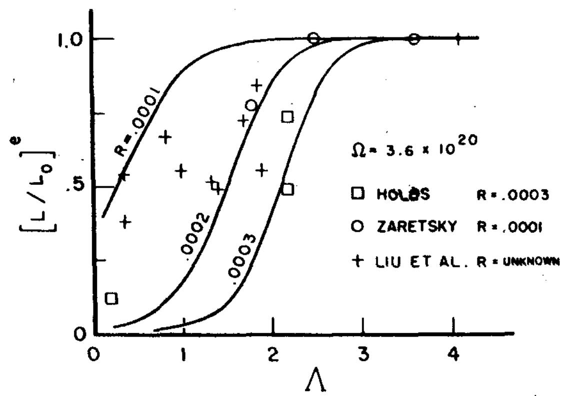


Fig. 5 Predicted Trend of Increasing Fatigue Life with Increasing Λ

of increasing fatigue life with increasing Λ, the ratio of lubricant film thickness to surface roughness. It agrees well with those observed in bearing life tests.

An alternative model to L-P's relation for contact fatigue life was developed by Tallian, Chiu, and Van Amerongen[15]. Their model is considerably more sophisticated than L-P's model, and includes not only the material strength effect and Hertzian stresses but other effects like defect shape, defect density, asperity contact stresses, and traction due to shearing the lubricant and the asperity junctions. Trends similar to that shown in Fig. 6 for the EHL effects can also be predicted from Tallian's theory.

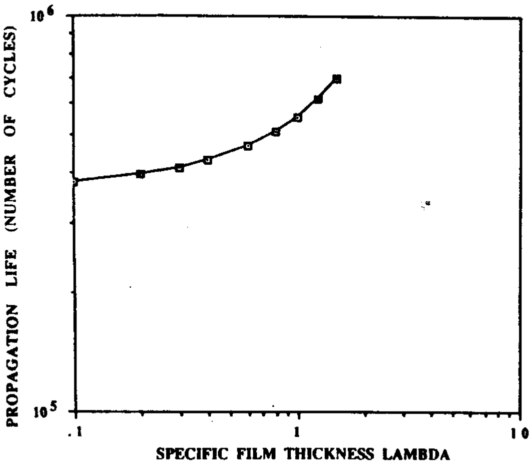


Fig. 6 Analytical Prediction of Life with Specific Film Thickness[17]

While the Lundbery-Palmgren's empirical relation has proven to be a practical tool for life prediction of rolling bearings, it is not an analytical model based on the basic principles governing the fatigue crack growth in contacting solids. Contact fatigue modelling based on basic relations controlling initiation and propagations of surface and subsurface cracks appears to be a more satisfying and rational approach, and should accommodate more easily the effects like lubrication, asperity stress, friction, and material imperfections. Analytical models based on propagation of cracks with a known initial crack length using Paris type propagation laws have been developed by Miller, et al.[16]. The extension of this approach for contacts with a known distribution of initial cracks under both asperity and Hertzian contact stresses has been carried out recently by Blake[17]. A typical life calculation shows a trend similar to earlier results on the effect of EHL film thickness in the low Λ region.

It appears that the status of contact fatigue life prediction is still heavily dependent on empirical methods originated from Lundberg and Palmgren. Analytical models based on understandings of crack propagation yield results encouraging, but are insufficient for accurate life prediction because of the exclusion of initiation life. Continous efforts are needed for developing a complete model for contact fatigue life including both initiation and propagation life.

3.3 Modelling of Scuffing Failure

For lubricated contacts operating at a high slide to roll ratio, the load can be limited by scuffing which is characterized by a sudden transition from low to very high wear rate accompanied by metal transfer. Because of a lack of adequate understanding of the lubrication breakdown mechanisms causing scuffing failure, analytical predictions of the threshold of scuffing have not been entirely satisfactory, as discussed by Dyson[18]. Future success in this area appears to depend on continuous efforts in understanding the breakdown of the following two types of films in protecting the sliding asperities.

The first line of defense is the asperity oil film, also known as the micro-EHL film. The effectiveness of this film is dependent upon the oil viscosity around the sliding asperity. If the oil is pressurized by EHL, asperity oil film would be effective due to a high oil viscosity around the asperity. Dyson[19] used the oil viscosity at the inlet half of the Hertzian conjunction as a measure of scuffing. He employed a convenient parameter, $Q = 1 - (\eta_0/\eta)$ to indicate the inlet viscosity and scuffing: when $Q \rightarrow 0$, scuffing would occur. This scuffing criterion based on the parameter Q has found a reasonable correlation with scuffing experiments[20], Fig. 7. Similar results, obtained by Snidle and Rosside[21] and recently by Lee and Cheng[22] also support the "Q" criterion (Fig. 8).

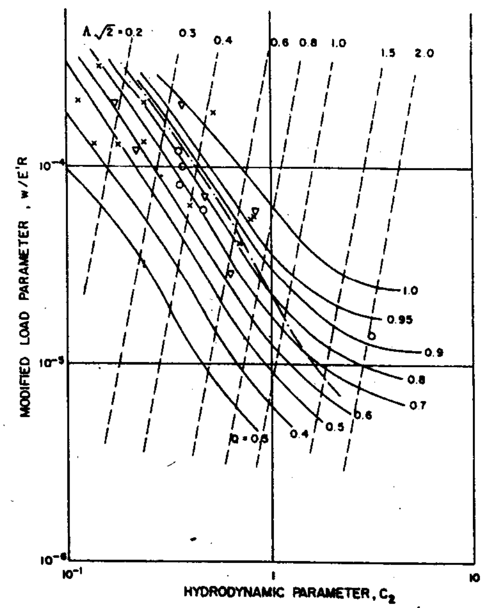


Fig. 7 Correlation of Experimental Scuffing Conditions with Some Hydrodynamic Parameters (o Straight mineral oil; ∇ Mild e.p. oil; × Straight mineral oil; —·—· Threshold value of C_2 based on $Q = 1.0$ with dry contact Geometry; $\sigma/R = 1.8\times10^{-5}$; $\alpha E' = 3333$; $p*/E' = 7.5\times10^{-5}$; $C_3 = 0$)

The second line of defense is the boundary or surface films which include the monolayer from adsorption of polar species, the chemical film from metal and additive reaction, and the polymeric film from metal/lubricant/oxygen catalytic reaction. Breakdown of these films seems to be all associated with some critical temperature at which these films cease to function.

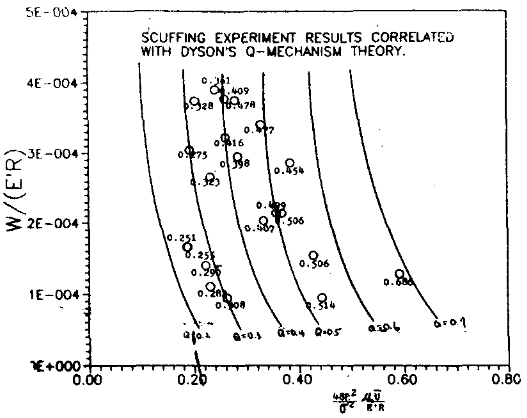


Fig. 8 Further Correlation of Experimental Scuffing Conditions with Dyson's Theory

Among these three types of surface film, only the prediction of critical temperature for the adsorbed film has had some success. Prediction for the breakdown of chemical and polymeric film are difficult because of a lack of surface tools capable of indentifying the exact chemical structure of these films. Since in most cases the threshold conditions at failure are determined by the breakdown of this last line of defense, the surface film, developments of failure criteria for surface films become an extremely important task.

3.4 Wear Modelling in Lubricated Contacts

Reliability prediction of tribo-systems would be much easier if designer can accurately predict and control wear in lubricated contacts. Unfortunately, such wear predictions are not possible at the present time because of a lack of good wear models in the lubricated regime. Tribologists must rise to meet this challenge.

Wear behavior in dry and lubricated contacts, as envisioned by participants in a recent workshop[23], is so complex that a general predictive model appears to be a nearly impossible task. However, a restrictive model, based on a limited number of simulative wear tests, may be successful in predicting wear within a range of operating conditions.

Much of the past modelling efforts in lubricated wear seems to follow this approach. For example, Rowe[24] modified the Archard's wear model for dry wear developed a relation for wear in boundary lubrication constants to be derived from wear experiments. It appears that Rowe's model may be extended to include the effect of EHL in reducing the asperity contacts and effect of micro-EHL in reducing the wear at the asperity contacts. This seemingly simple approach might be a useful first step towards developing an adequate wear model in lubricated contacts.

3.5 Prediction of Bulk Surface Temperature

The significance of bulk temperature in determining the tribological performance has been well recognized by designers. However, there still is a lack of an effective method in handling this problem other than the cumbersome and costly finite element method.

With the current trends towards developing integrated computer softwares for prediction of tribological performance, it appears to be ripe to develop a versatile computation tool to determine the bulk temperatures of a system of interactive tribo-elements using Blok's thermal network concept[25]. Of course, the thermal resistance for each tribo-element within the network can still be determined beforehand by the finite element codes and enter in the network as numeric database.

3.6 Computerized Tribology Information System

In carrying out a tribological design process as shown in Fig. 2, it is often difficult to locate the pertinent numeric database for the lubricant and material properties, the appropriate softwares for calculating the tribological performance, and the pertinent critèria for predicting the failure thresholds. Often designers find themselves spending so much time in searching such information and still end up with not quite the most desirable data or method. This problem can be alleviated if a centralized and computerized tribological information system can be developed. Such need has been pointed out by several countries,and action are now underway to develop such systems for designers. An example of this is ACTIS, known as a computerized tribological information system currently being developed by the National Institute of Standards and Technology with support from the U.S. Department of Energy[26]. As shown in Fig. 9, ACTIS will have six databases with the numeric and design databases aiming directly to tribological design and other databases to assist in production and research.

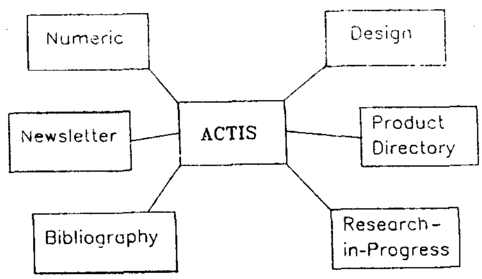


Fig. 9 The Six Database in ACTIS

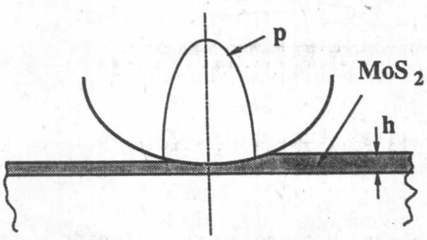
4. NEW CHALLENGES

In the past, tribological design has centered mainly around metallic pairs lubricated by mineral or synthetic fluids. This situation is changing rapidly due to needs in developing high efficiency and light weight automotive and aircraft engines, precision manufacturing machines, and information processing equipments. Numerous unconventional materials, lubricants, and lubrication concepts have emerged as promising candidates to meet these new needs. Yet, there is very little analytical understandings to guide the design of these tribological applications. There exist many golden opportunities for tribologists to break new grounds in these new areas.

4.1 Mechanics of Soft Films

The use of a low shear strength layer, such as MoS_2 or graphite on a hard substrate to reduce friction is well known. Its friction characteristic can be readily predicted based on a simple linear relation between the shear strength and the normal contacting pressure. Such relation yields a frictional coefficient which decreases asymtotically with pressure to a constant α, as shown in Fig. 10. For some low shear MoS_2 films very small, α have been measured in vacuum[27].

Friction of Doposited MoS_2



$$\tau = \alpha p + \tau_o$$

$$f = \frac{\int \tau \, dA}{W} = \alpha \int p \, dA + \tau_o A$$

$$= \alpha + \frac{\tau_o}{\bar{p}}$$

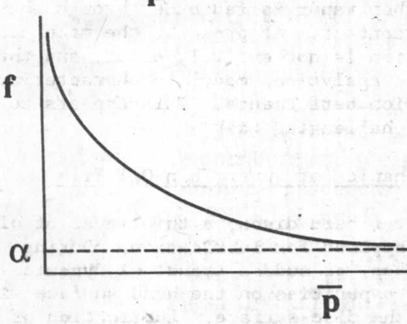


Fig. 10 Friction Characteristics of A Doposited Soft Film

Since life of these soft films is determined by wear, wear prediction becomes essential in design of such films. Wear modelling of a soft film based on analysis of a rigid indentor on a perfectly plastic thin layer may shed some light on the wear process. Coupled with a careful experiment of wear generation between a single conical sliding tip and a soft layer, a semi-empirical wear model may be realizable.

4.2 Mechanics of Hard Coatings

The use of hard coatings to reduce wear is also a well known concept. However, its growing acceptance in tribological applications is quite recent. At present, the selection of coating materials and thickness is achieved largely by trial and error. There is a need of basic understanding of the mechanics of hard coatings to guide the design. Recently, Halling and Arnell[28] introduced a useful concept of effective hardness to guide the selection of coating thickness for both hard and soft coatings. Fig. 11 shows that the effective hardness is depending on the ratio of the coating thickness t to the radius of the indenting slider β. When this ratio approaches 0.03, the coating is fully effective, and the substrate has no effect on the hardness of the layered structure.

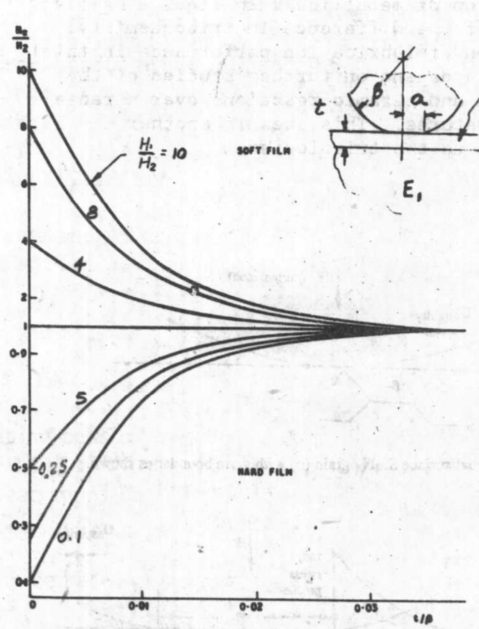


Fig. 11 Effective Hardness of A Soft or Hard Coating

Very little is available to determine quantitatively the beneficial effect of hard coatings on scuffing load, wear rate, and fatigue life. Since hard coatings are anticipated to be used more extensively for wear reduction, it is critical to determine the predominant failure modes, such as interfacial debonding, subsurface spalling, and surface cracking, and to derive some failure prediction methods based on stress field calculated for the layered structure.

4.3 Lubrication and Wear of Ceramics

Ceramics has emerged in the past decades as the most promising candidate for tribo-elements operating at elevated temperatures. In a recent report[29], issued by the U.S.