THE UNIVERSITY OF HULL

Evaluation of Rolling Contact Fatigue Resistance for Coated Components

Being a Thesis Submitted for Degree of Doctor of Philosophy in the University of Hull

by

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ABSTRACT

The thesis reviews and studies current evaluation mechanisms, techniques and machines for testing rolling contact fatigue failure resistance and load capacity of coated components. The thesis investigates both normal and accelerated rolling contact fatigue evaluation test mechanisms and their models, and evaluation test technique principles suitable to the appraisal of coated bearing components. A major contribution of the thesis is the design and development of a new rolling contact fatigue evaluation test machine for coated components. Tests of the rolling contact fatigue of coated bearing raceways under the oil lubricant, grease lubricant and no lubricant conditions, applying the new rolling contact fatigue evaluation mechanisms, evaluation technique principles and the new test machine, have been performed. The accelerated rolling contact fatigue tests of the coated bearing raceways use SiC powder in the oil lubricant.

The new rolling contact fatigue test machine has been found suitable for evaluating the rolling contact fatigue resistance of components with superhard coatings. The accelerated rolling contact fatigue test method has been shown to give comparable rolling contact fatigue test results to those obtained in a normal rolling contact fatigue test, while being much faster. In the fatigue test, the cyclic maximum shear stress produces an initial fatigue crack near the substrate surface of the test bearing raceways. The observed phenomena are consistent with theory, although the location of the initial crack is much closer to the surface than would be predicted by a 'static' Hertzian analysis. Insufficient traction forces on the contact surface between the rolling elements of a test coated bearing makes gross skidding occur, leading to rapid wear, over-heating and final failure of the test coated bearing. The L50 fatigue life of the test coated bearing raceway tends to decrease with increase of the coating thickness and coating hardness of the test coated bearing raceway.
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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
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<tbody>
<tr>
<td>$a$</td>
<td>Semimajor axis of the projected contact ellipse</td>
<td>mm</td>
</tr>
<tr>
<td>$a^*$</td>
<td>Dimensionless semimajor axis of the projected contact ellipse</td>
<td></td>
</tr>
<tr>
<td>$b$</td>
<td>Semimajor axis of the projected contact ellipse</td>
<td>mm</td>
</tr>
<tr>
<td>$b^*$</td>
<td>Dimensionless semimajor axis of the projected contact ellipse</td>
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<tr>
<td>$E$</td>
<td>Modulus of elasticity</td>
<td>N/mm²</td>
</tr>
<tr>
<td>$\varrho$</td>
<td>Complete elliptic integral of the second kind</td>
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<tr>
<td>$\psi$</td>
<td>Complete elliptic integral of the second kind</td>
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<tr>
<td>$F$</td>
<td>Force</td>
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<tr>
<td>$G$</td>
<td>Shear modulus of elasticity</td>
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</tr>
<tr>
<td>$l$</td>
<td>Roller effective length</td>
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<td>$Q$</td>
<td>Normal force between rolling element and raceway</td>
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</tr>
<tr>
<td>$r$</td>
<td>Radius of curvature</td>
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<td>Principal stress</td>
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<tr>
<td>$u$</td>
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<td>$U$</td>
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<td>z₁</td>
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<td>Z</td>
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<tr>
<td>γ</td>
<td>Shear strain</td>
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<tr>
<td>δ</td>
<td>Deformation</td>
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<tr>
<td>δ*</td>
<td>Dimensionless contact deformation</td>
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<tr>
<td>ε</td>
<td>Linear strain</td>
<td></td>
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<tr>
<td>ζ</td>
<td>z/b, roller titling angle</td>
<td>°, rad</td>
</tr>
<tr>
<td>θ</td>
<td>Angle</td>
<td>rad</td>
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<tr>
<td>κ</td>
<td>a/b</td>
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</tr>
<tr>
<td>λ</td>
<td>Parameter</td>
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</tr>
<tr>
<td>ξ</td>
<td>Poisson's ratio</td>
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</tr>
<tr>
<td>σ</td>
<td>Normal stress</td>
<td>N/mm²</td>
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<tr>
<td>τ</td>
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<tr>
<td>v</td>
<td>Auxiliary angle</td>
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<tr>
<td>i</td>
<td>Refers to inner raceway</td>
<td></td>
</tr>
<tr>
<td>o</td>
<td>Refers to outer raceway</td>
<td></td>
</tr>
<tr>
<td>r</td>
<td>Refers to radial direction</td>
<td></td>
</tr>
<tr>
<td>x</td>
<td>Refers to x direction</td>
<td></td>
</tr>
<tr>
<td>y</td>
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<tr>
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<td>yz</td>
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<td>xz</td>
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<tr>
<td>I</td>
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<tr>
<td>II</td>
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CHAPTER 1 Introduction

1.1 Background of the thesis research

In the development of modern materials, hard and superhard coatings have been successfully used to improve the surface material functionality of components. Many classes of composites exist, most of which have improved mechanical properties such as stiffness, strength, toughness and resistance to fatigue. Coating composites (surface engineered materials) are designed to specifically improve functions, and these may be, for example, tribological, electrical, optical, electronic, chemical and magnetic. It is thus natural to select the bulk of a component to meet the demands for stiffness, strength, formability, cost, etc. and then modify or add another material as a thin surface layer. This surface layer or coating is the carrier of virtually all other functional properties. The application of coatings on cutting tools, bearings, gears, camshafts, and special key components is, therefore, a very efficient way of improving their mechanical and tribological properties (Choy, K. L. 2003; Hogmark, S., 2000; Holmberg, K., 2000).

Evaluation technologies are crucial to improve the coated component design and to design new coating materials for meeting application demands. However, a general theory covering all relevant properties and parameters involved in the design and application of tribological coating composites is very far from being realised. Such a theory would have to treat the long chain of relationships ranging from the coating deposition parameters to the tribological response of the coated component. Generally, the end-users of coated components are recommended to make the final evaluation of the tribological response in field tests or in component tests, i.e. tests where the actual component is evaluated under realistic conditions. Simplified laboratory tests often deviate from the actual situation as to
nominal and real contact pressure, sliding speed, heat conductivity and capacity, ambient cooling, etc., which makes correlation to the real case problematic (PalDey, S. and Deevi, S. C., 2003; Hogmark, S., 2000; Stewart, S. and Ahmed, R. 2002).

The current fatigue failure evaluation techniques and methods e.g. the indirect evaluation methods, the impact methods and the sliding methods often generate premature cracks occur before the initial fatigue cracks when superhard coated specimens are tested. Also four ball and modified four ball simplified laboratory test methods often deviate from the actual situation as to nominal and real contact pressure, sliding speed, heat conductivity and capacity, ambient cooling, etc., which again means that the results may not correlate with real in-service performance (Huq, M. Z. and Celis, J. P., 2002; Wang Dong F. and Koji Kato, 2001; Dommarco, R. C., 2002).

1.2 Thesis motivation

The broad aims of the thesis research work were to:

1. Seek a new evaluation technique method for the evaluation of fatigue failure resistance and load capacity of coated components, i.e. rolling bearings, gears and camshafts, to fulfil the needs of their applications.

2. Seek an improved understanding of coating contact fatigue test mechanisms to support new methods for the evaluation of fatigue failure resistance and load capacity of coated components.

3. Design and manufacture a new rolling contact fatigue test machine, according to the new findings on coating contact fatigue test mechanisms and new evaluation technique principles, for the evaluation of rolling contact fatigue failure resistance and load capacity of coated components, i.e. rolling bearings, gears and camshafts, to fulfil the
needs of their applications.

1.3 Thesis research line

The chronology of the work undertaken was as follows:

1. Review and study current evaluation mechanisms, evaluation technique principles and evaluation test machines of fatigue failure resistance and load capacity of coated components.

2. Study rolling contact fatigue evaluation mechanisms, evaluation technique principles and evaluation test machines, which should be suitable to the appraisal of coated components, e.g., rolling bearings, gears, camshafts, etc.

3. Study high-speed rolling contact fatigue evaluation test mechanisms, and evaluation test technique principles which should be suitable to the appraisal of coated components.

4. Design and manufacture a new rolling contact fatigue evaluation test machine for coated components.

5. Carry out evaluation experiments of rolling contact fatigue of coated bearings applying the new rolling contact fatigue evaluation mechanisms, evaluation technique principles and evaluation test machine.

6. Summarise and present findings on rolling contact fatigue evaluation mechanisms, evaluation technique principles and evaluation test machines.
CHAPTER 2 Coated component evaluation review

2.1 Introduction

This chapter reviews the general technical application situation of coating materials, hard coatings, superhard coatings, and coated components. It systemically analyses and discusses coated component performances and classifies failure mechanisms. It also discusses contact fatigue failure mechanisms of coated components, including the microstructural deformation, fatigue crack accumulation, fatigue initial crack depth and fatigue crack growth.

It surveys current evaluation technology methods for coated component damage. The evaluation system of coated component properties and evaluation parameters are discussed.

The chapter reports on the evaluation methods of contact fatigue failure resistance for coated components. It systematically analyses and discusses coated component contact fatigue failure mechanisms, test evaluation methods, test evaluation principles and test machine structure.

The chapter, according to modern coating technology development requirements, presents research comments on requirements for further study and improvement in evaluation technology and methods for contact fatigue resistance of coated components.

2.2 Coatings, superhard coatings, and coated components

In the development of modern materials, hard coatings and superhard coatings have been successfully used to improve the surface material functionality of components. Many classes of composites exist, most of which improve mechanical properties such as stiffness, strength, toughness and resistance to fatigue. Coating composites (surface engineered
materials) are designed to specifically improve functions, and these may be, for example, tribological, electrical, optical, electronic, chemical and magnetic. It is thus natural to select the bulk of a component to meet the demands for stiffness, strength, formability, cost, etc. and then modify or add another material as a thin surface layer. This surface layer or coating is the carrier of virtually all other functional properties. The application of coatings on cutting tools, bearings, gears, camshafts, and special key components is, therefore, a very efficient way of improving their mechanical properties and tribological properties (Choy, K. L. 2003; Hogmark, S., 2000; Holmberg, K., 2000).

As to the hardness, the coatings are usually divided into two groups: (1) hard coatings having hardness <40 GPa. (2) superhard coatings having hardness >40 GPa, i.e. cubic boron nitride (c-BN), amorphous diamond-like carbon (DLC), amorphous carbon nitride (a-CN_x) and polycrystalline diamond. Both the technological process of their production and their properties, i.e. hardness, wear and oxidation resistance, however, are continuously being improved (Musil, J., 2000).

Surface coatings for tribological applications are associated with deposition temperatures ranging from room temperature to over 1000°C. The coating thicknesses range from the sub-micron level to several millimetres. Typically, the atomistic methods produce nanocrystalline structure thin coatings which are explored for tribological applications (Musil, J., 2002; Hogmark, S., 2000).

2.3 Coated component failure performances

Coating material damage failures can be divided into main two groups by their damage mechanisms and their service life: premature damage failures and fatigue damage failures.
2.3.1 Coated component premature damage failures

Coated material damage mechanisms are different to those for homogeneous material. The properties of substrate and interface play a significant role in the damage performance of the coating materials. The coating design should be selected to match the tribological situation, the life-limiting and surface damage mechanism of the intended application. Tribological coatings can fail prematurely due to detaching, cracking and pitting. According to premature damage performance, coating material damage failures can be classed three kinds (Hogmark, S., 2000; Ahmed, R. and Hadfield M., 1998):

1. Damage without exchange of material

Typical damage without exchange of material is displayed as scratches and cracks. This category basically involves permanent changes in component geometry and/or in surface topography. Decisive parameters for a change in geometry are Young's modulus, hardness of coating and substrate, and coating toughness. Tiny scratches or cracks may disqualify a forming tool used, e.g. to press compact discs. Coating hardness is the crucial parameter for scratch resistance; and coating toughness or fracture resistance are crucial for the resistance to surface cracking.

2. Wear with loss of material

Generally, tribological applications put higher demands on the coating adhesion than any other area of application, although the demands may differ substantially from one situation to the other. It is instructive to distinguish between the actual adhesive forces (the strength of the physical adhesion or atomic bonding which acts between coating and substrate) and the practical adhesion. The practical adhesion is the ability of the coating composite to resist interfacial failure in its practical application.
The wear resistance of a coated component is mainly determined by the coating as long as it covers the contact area. As soon as the coating is partly worn through, or the substrate is exposed due to adhesive failure or cracking and spalling, the wear resistance of the substrate material becomes important. However, two main categories can be distinguished: wear dominated by coating detachment and wear caused by gradual removal of coating material. The latter often involves mild wear due to abrasion, erosion, chemical dissolution, etc., and does not deviate from the mechanisms causing wear of homogeneous materials.

3. Damage with material pick up

Work material locally adhered (e.g. to the surface of a sheet forming tool used in the automotive industry) will inevitably produce indentations or scratches in the surface of the product. Material transfer between the contact surfaces of sliding machine elements is a similar problem often named galling, scuffing or even seizure.

Material pick up from the counter surface is, again, not unique to coating composites. It is generally reduced or avoided by giving the surface a smooth topography and making sure that the chemical affinity to the counter surface is low. This is often accomplished by applying an appropriate tribological coating.

Tribochemical layers may form, e.g. when machining certain work materials at high cutting speeds. They are the result of mechanical smearing or chemical reactions with constituents in the work material, and may have the positive effect to protect the coating from excessive damage (Hogmark, S., 2000).
2.3.2 Coated component fatigue damage failures

Since a small permanent deformation of the substrate, or interface, or coating material is accumulated in each contact event (ratcheting), the fracture limit of the coating will eventually be reached and fatigue failure will occur.

Fatigue damage failures are very typical damage failures of coating materials, and fatigue failure resistance is an extremely important criterion to assess the capacity of coating materials. The improvement of the evaluation methods for coating material fatigue failure should be of benefit to both new applications and new designs of coating materials. Further study of the damage failure mechanisms, improved evaluated technologies, and the design of practical evaluated methods for new coating materials are necessary research projects in the surface engineering field.

2.4 Current evaluation technologies and methods for coated components

Coated component evaluation technology is used to check and measure the chemical, thermal and mechanical properties of the coating, interface and substrate of coated components, and to identify the capacity of coated components for designers and users. Evaluation technologies have been taking an important role in the improvement and progress of both new coating materials design and new coating applications. People involved in coatings development and production usually assess some of the relevant coating characteristics and basic properties, whereas end-users should focus on the tribological response of the coated component under the actual application (Hogmark, S., 2000; Stewart, S., 2002).
2.4.1 Coated component evaluation system

Coating component evaluation technologies are the passport for coated components to hold promise for industrial application. Also, coated component evaluation results are the main reference factors for designers to improve coating design. The science and technology development and progress have provided the new technological conditions and new theoretical basis for modern coating material design and new coating component applications. So the evaluation technologies have been taking an important role in the improvement and progress of both new coating material design and applications. However, a general theory covering all relevant properties and parameters involved in the design and application of coated components is very far from being realised (Hogmark, S., 2000). Further study on coating evaluation technologies is necessary, seeking new evaluation methods and meeting the needs of the new applications. A proposed coated component evaluation system is shown in Figure 2.1. The idea of this is to show a need for a progressive sequence of design and evaluation in order to fulfil the operating requirements of the coated component.
Figure 2.1 Coated component evaluation system
2.4.2 Coated component properties and parameters

The coating component properties, in fact, are an integration of properties from coating, interface and substrate, and will decide the coated component responses from applications. The common coated component properties are determined by the intrinsic mechanical property parameters and material property parameters of coated components.

2.4.2.1 Substrate evaluation parameters

1. Intrinsic mechanical property parameters:
   Young's modulus, residual stress, hardness and toughness or fracture resistance,

2. Material property parameters
   Material composition, microstructure, topography, substrate adhesion

2.4.2.2 Coating film property parameters

1. Intrinsic mechanical property parameters:
   Young's modulus, residual stress, hardness and toughness or fracture resistance,

2. Coating characteristic parameters:
   Thickness, chemical composition, microstructure, morphology and topography, etc

3. The deposition parameters:
   Substrate temperature, plasma characteristics, etching time, substrate bias, etc

2.4.2.3 Interface

The adhesion at the interface between coating and substrate is a crucial property of most applications of coated components. The damage failures of coated component usually occur at or near the interface of a coated component.
2.4.3 Technical methods of coated component evaluation

2.4.3.1 Evaluation parameters of coated components

Evaluation parameters of coated components are usually used to estimate the tribological response of a coated component in operation to cope with the actual tribological situation, i.e. conditions such as geometry, contact pressure, sliding velocity, temperature, lubrication, etc. Evaluation parameters of coated components involve the premature damage failure resistance and the fatigue failure resistance.

2.4.3.2 Intrinsic mechanical properties

Some intrinsic mechanical coating properties of particular interest, the Young's modulus, residual stress, hardness and toughness or fracture resistance, are presented below:

1. Young's modulus

The Young's modulus of the coating \((E_c)\) is a useful parameter, e.g. for measurements and calculations of the stress state and the cracking and delamination behaviour of coating composites. It is possible to obtain \(E_c\) through a number of techniques, where the uniaxial tensile test is the most straightforward (Hollman, P. et al., 1997). The intrinsic elastic modulus of thin coatings can also be obtained by nanoindentation, vibrating reed tests, bulge tests, beam bending tests, ultrasonic wave propagation, etc. (Hogmark, S., 2000).

2. Residual stresses

a. Residual stress mechanism of coatings

Superhard PVD and CVD coatings usually display high residual stresses \((\sigma_{\text{res}})\). Structural misfits in epitaxial nucleation and growth, and ion bombardment during growth are two stress origins of intrinsic nature. The stresses induced during cooling from the deposition temperature due to mismatch in thermal expansion between coating and
substrate materials, and possible phase transformations occurring during cooling are two sources of the external origin. The final stress state is a combination of these components (Windischmann, H., 1992).

The actual stress during application ($\sigma$) is given by

$$\sigma = \sigma_{\text{res}} + \sigma_{\text{app}}$$  \hspace{1cm} (2.1)

where $\sigma_{\text{app}}$ denotes the stress field induced by the application, including external forces and thermal mismatch stresses due to frictional heating.

Too high compressive stresses may result in spontaneous coating detachment, e.g. during cooling from the process temperature. In less severe cases, the coating may detach when the coated component becomes loaded externally. The risk for detachment is closely related to the geometry and topography of the coating/substrate interface, the smoother the interface the less is the risk. On an uneven surface, the interfacial normal or shear stress generated by the residual coating stresses can exceed 50% of the residual stress level (Wiklund, U. et al., 1999).

Coating film residual stress is a crucial effect parameter on the premature failure resistance and fatigue failure resistance.

b. Residual stress evaluation of coatings

The popular techniques used for residual stress measurements are based either on measurements of the elastic strains in the film using X-ray diffraction, or on the deflection of thin coated substrates. X-ray techniques can yield information of all strain components in the coating, and also give information about the strain distribution through the thickness of the coating. To obtain the residual stress using X-rays, the elastic constants of the coating
must be known. In the substrate deflection techniques, the coating residual stress is
determined through measuring the deflection it causes to the substrate (Venkatraman, R.,

3. Hardness

Coating developers often use hardness measurement to assess coating quality and to
predict the coating performance in various applications. However, the importance of a high
intrinsic coating hardness should not be exaggerated. Generally, in pure two-body abrasive
wear, the wear resistance is very closely coupled to the hardness, as long as the abrasive
particles (or abrasive surface) are harder than the wearing surface. Most counter surfaces
expected for tribological applications of coated components are softer than 20 GPa, a value
exceeded by many of today's PVD and CVD coatings (Oliver, W.C., 1992).

4. Coating hardness measurement

Intrinsic hardness values of thin hard coatings can be directly measured by
conventional microhardness testing if the indentation depth does not exceed some 10% of
the coating thickness. Consequently, direct measurements using Vickers indentation are
restricted to coatings thicker than about 5 \( \mu \)m. It is possible to interpret microhardness
values obtained from thinner coatings by using models which consider the substrate
deformation (Jönsson B. and Hogmark, S., 1984).

During the last decade, nanoindentation has become the predominant technique to
obtain intrinsic mechanical properties of thin coatings. In nanoindentation, the applied load
is typically 0.01–5 g as compared to 5–1000 g for microhardness testing. In
nanoindentation, the tip displacement, load and time are continuously recorded. The
hardness and Young's modulus are obtained from the load/displacement curves using
different theoretical approaches, e.g. as proposed by Oliver and Pharr (Oliver, W.C., 1992).
5. Toughness

Coating cracking or fracture often precedes damage of PVD and CVD coatings. Toughness is an intrinsic property of coating materials, which defines the ability of the coating composite to accommodate deformation in tension or compression without crack nucleation and propagation.

Several investigators have used beam bending to assess the deformability of coatings and to obtain numerical estimates of their toughness (or fracture resistance) (Wiklund, U. et al., 1997). In the device, the bending load is continuously increased and the critical strain to initiation of the first crack is recorded acoustically or in the SEM. It has been observed that multilayered coatings generally show higher critical strains to fracture than do homogeneous coatings.

Since cracking is usually initiated by tensile stresses, any compressive residual stress has first to be relaxed. Consequently, if the coating initially has a high compressive residual stress, the coated component can take more tensile strain before the coating will fracture. The critical component strain is thus a more important parameter than the critical intrinsic tensile strain of the coating.

It has been reported that the true fracture strain of many PVD coatings is very low compared to that of corresponding homogeneous bulk ceramics, which indicates that there is a huge potential for improving their toughness (Hogmark, S., 2000).

2.4.3.3 Evaluation methods for coated component tribological properties

1. Scratch resistance

a. Evaluation test principle
Scratch testing has, together with hardness measurements, become the most common way to assess the mechanical quality of coating composites. Usually, the scratch test utilises a spherical diamond tip of Rockwell C geometry (200μm tip radius). During testing the tip load is continuously increased and a critical load for coating failure is detected. The failure criterion may be occurrence of the first crack or first induced cohesive or interfacial fracture. The failure may be determined by friction and acoustic emission (AE) recording. Optical microscopy or SEM should confirm the results.

b. Scratch resistance evaluation test

Scratch testing can also give detailed information about different modes of coating failures, and much knowledge of a coating composite can be gained by studying the scratched sample, e.g. in the SEM. A general rule of thumb says that a critical load of 30 N in scratch testing with a Rockwell C diamond tip is sufficient for sliding contact applications. Critical loads of 60–70 N are frequently recorded for PVD coatings on hardened HSS (Larsson, M., 1996).

2. Resistance to abrasive wear

In situations of mild abrasion, the coating material may determine the wear resistance of a coating composite. Standard abrasive wear tests are usually too coarse to be useful for generating the intrinsic wear resistance of thin coatings.

However, by the micro abrasion test originally proposed by Kassman et al. (Kassman, Å. et al., 1991) and further developed by Rutherford and Hutchings (Rutherford, K.L. et al., 1996, and Rutherford, K.L. and Hutchings, I.M., 1997), and Gåhlin et al (Gåhlin, R. et al., 1997), it is possible to distinguish the abrasive wear resistance of a thin coating material from that of the substrate, also in situations where the coating is worn through.
A small grinding wheel (dimple grinding) or ball (ball cratering) is used to produce a spherical crater in the surface of the coating composite. The contact area is surrounded by an abrasive medium. The test is interrupted at regular intervals and the crater volume is estimated either from measuring the diameter using an optical microscope, or directly by using 3D surface profilometry. By assuming Archard's law to be valid for coating and substrate individually, one arrives at a simple rule of mixture

\[
SL = \frac{V_c}{K_c} + \frac{V_s}{K_s}
\]  

(2.2)

where \( S \) is the sliding distance, \( L \) the applied load, \( V_c \) and \( V_s \) the wear volumes of coating and substrate, respectively and \( K_c \) and \( K_s \) are the specific wear rates of the coating and substrate. The intrinsic wear resistance of some PVD coatings is obtained as \( 1/K_c \).

In addition to the coating and substrate wear resistances, by subsequent inspection of the test craters in the SEM, the micro abrasion test reveals any content of coating defects such as pores and cracks. A poor adhesion is often detected from the presence of spalling in the coating substrate interface.

Microabrasion test results must be handled with caution. Two-body abrasive wear of the coated surface prevails if the abrasives adhere to the surface of the grinding wheel or ball. This is likely to be the case if the surface of the wheel or ball is softer than the tested material. If the tested material is softer, the abrasives will become pressed into its surface, resulting in abrasive wear of the rotating ball or wheel. In situations where both surfaces are of equal hardness or the particles form thick layers, the particles may roll rather than slide, and the wear will be dominated by three-body abrasion (Axén, N. et al, 1994). Other
parameters which must be kept under control are size distribution and volume fraction of the abrasive particles and viscosity and wetting angle of the liquid medium.

3. Resistance to particle erosion

Surface damage caused by impinging hard particles is usually referred to as particle erosion. Generally, resistance to particle erosion requires a combination of hardness and toughness, with the toughness being the dominant parameter. For a thin coating to be effective in erosion protection, individual impacts must not plastically deform the substrate material. The extension of plastic strain is controlled by the particle size, velocity and angle of impact. In mild situations, where only the coating is permanently deformed by the impacts, particle erosion can be used to evaluate intrinsic erosion properties of the coating, or as a micro scale toughness test (Hogmark, S., 2000).

4. Resistance to sliding wear

a. Sliding wear

Sliding wear is here referred to as wear in a tribological system where the coated component slides against a relatively smooth counter surface, free from hard particles or hard asperities. Naturally, sliding wear involves a very large group of tribological situations, and the wear may range from very mild chemical wear to severe adhesive wear and coating detachment.

b. Sliding wear evaluation

After a sliding wear test, the mass loss of typical PVD or CVD coatings is too small to be resolved by weighing. However, the situation has recently been improved by the introduction of accurate surface profilometers and by atomic force microscopes. These techniques allow very small wear volumes to be mapped and measured (Gåhlín R. and
Jacobson, S., 1998). Apart from the small volumes involved, there is no principal difference in evaluating the intrinsic sliding wear resistance of thin coatings and bulk materials. In tooling applications, tribochemical mechanisms often dominate over mechanical wear. Thus, the selection of counter material as well as reproduction of representative temperature and atmosphere is crucial for components in simulative tests.

A new test for evaluation of friction and load carrying capacity of coating composites has recently been suggested (Hogmark, S. et al, 1998). Two elongated specimens are slid against each other in a way similar to that of the contact between the edges of a pair of scissors. If the load is gradually increased, as in the scratch test, each contact spot along the wear track will experience a unique load. Unidirectional as well as multipass sliding can be applied, and critical loads for coating failure can be obtained as for the scratch test.

The advantage over the scratch test is, however, that the contact situation is very much closer to practical applications of sliding contact. In addition, the counter material as well as the contact geometry (radius of contacting rods) can be selected to represent the intended application.

2.4.3.4 Evaluation of contact fatigue resistance of coated components

The contact fatigue resistance of a coated component is explained as a capacity for the coated component to resist contact fatigue failure under special test conditions. The effects of factors to result in contact fatigue failures are extremely complicated; they are related to coating material properties, substrate properties, coating technical methods, and applied conditions, etc.

The contact fatigue evaluation takes very important role in the procedure in coated component design and its application. The coated component evaluation can be divided into
two steps: one, coated component design evaluation; two, coated component application evaluation.

First, the role of the evaluation in the coated component design step is to provide some essential indicators for the design of coating composition and deposition technique to meet the need of coated component application properties. In this step, the objects to be evaluated are the intrinsic property parameters of coating and substrate.

In the step involving coated component applications, the role of the evaluation of coated components is to provide some necessary indicators for increasing the capacities of coated components to resist damage failures and to accommodate the loads in practical application conditions. Before putting coated component into use, the expectant criterion of the ability to resist damage failures and to absorb loads under practical applications must have been provided to the end-users.

As modern new materials extend rapidly into new applications, the material fatigue resistance evaluation technologies appear more and more important. For example, the fatigue evaluation theories and technical methods of homogeneous materials have been improving continuously. The fatigue failure evaluation methods of coating materials, superhard coating materials, nanostructured superhard coating materials and coated components have become an urgent problem requiring solution. The modern coating technological progress mostly depends on solution to this problem. Therefore, the practicability and reliability of the evaluation methods for coated components will be vitally important to the applications of coated components. (Tushsky, L., 2002).
2.5 Review evaluation methods of contact fatigue failure resistance of coated component

General evaluation methods for fatigue failure resistance of coated components can be divided into two classes by the fatigue mechanisms of coated components. One is the indirect method, indirectly estimating from intrinsic mechanical properties of the coating. Another is the direct experimental method, and it again is classed into two main sorts by the cyclic stress situation on the sample. They are the pulse stress experimental method and the rolling contact stress experimental method.

2.5.1 Indirectly estimated method for fatigue resistance of coated components

According to intrinsic mechanical properties (Yuan's model, hardness, toughness residual stress, etc.), the fatigue failure resistance is estimated and calculated (Hogmark, S., 2000). The estimated value can represents the general tendency of fatigue failure resistance of coated components. The common indirectly estimated methods are the static indentation test method and the micro and nanoindentation test method. It is becoming routine in many industrial and academic research laboratories to evaluate the wear-resistance of advanced coatings by depth sensing indentation based on methods such as nanoindentation and scratch testing.

Although indirect estimation methods are often very useful, the results often do not directly correlate with actual coating performance when coatings and hard thin films are subjected to erosive wear or dynamic loading during service and fail by a fatigue process. Also, conventional diamond indenters are difficult to prepare accurately and easily become damaged when used on superhard material coatings like cubic boron nitride, silicon carbide and diamond itself (Zhang, L.Y., 1997).
2.5.2 Experimental methods for contact fatigue resistance of coated components

Conventional experimental methods for evaluating the contact fatigue failure resistance of coated components have two groups, which are pulse impacted methods and rolling contact fatigue methods. The principle of pulse impacted fatigue methods is explained as a cyclic pulse impacted force acts on the coating specimen to get the expected information of coating specimen fatigue duration properties. The rolling contact fatigue experimental evaluation method utilizes the contact cyclic stresses acting on rolling contact elements to provide the expected information on coating specimen fatigue duration properties, and then it is possible to utilize the information of coated specimen fatigue duration properties to evaluate coated component fatigue failure resistance.

2.5.2.1 Pulse impacted fatigue test methods

Pulse impacted fatigue experimental testers permit a controllable alternating impacted force. Hence diagrams of the contact load that leads to coating fatigue fracture versus the corresponding number of impacts can be obtained.

The coating impact test, in combination with its finite elements method (FEM) simulation, has been successfully used to characterize the fatigue performance of thin hard monolayer coatings, as well as of multilayer ones (Bouzakis, K. D. et al., 1999). The test is based on successive impacts of a cemented carbide ball onto a plane-coated specimen, which induce contact loads and strain. The fatigue failure mode of each specimen is classified by means of SEM observations, EDX microanalyses and profilometry. FEM simulating models of the impact tests are used to determine the critical stress components, which introduce coating fatigue failure. Critical values for stress components, responsible for distinctive fatigue failure modes of the coating substrate compounds are obtained and
the fatigue limits of various coatings are illustrated in a way which is analogous with Smith and Woehler diagrams. To further improve this method, an advanced impact tester supported by appropriate software facilities, able to evaluate the fatigue strength of hard coatings, was developed (Bouzakis, K. D. et al., 2001). In this enhanced tester the contact loads as well as the number of impacts can be readily varied so that the fatigue failure for coatings with different technological specifications and material properties can be obtained. Moreover, the continuous data acquisition as well as the real time monitoring and evaluation of the test bench are enabled. The test results are recorded in diagrams containing the impact load versus the number of successive impacts that a coating substrate compound can withstand. Thus, through appropriate computer software, the fatigue strength of thin hard coatings can be automatically determined and expressed in the form of Smith and Woehler diagrams. The code is supported by an extended database, implemented into an analytical procedure based on pre-conducted FEM calculations, covering a wide range of coating substrate compounds, considering also a variety of technological specifications and material properties.

The pulse impacted fatigue test is introduced as a convenient method to determine quantitatively the fatigue behaviours of coating-substrate compounds in a form of generally applicable lifetime diagrams (Bouzakis, K. -D., et al., 1999). The successful operation of such lifetime diagrams in various fatigue related coating applications has led to the concept of a redesigned and optimized impact tester, as well as to the facilitation of the test procedure and to the establishment of coating fatigue data (Bouzakis, K. D. et al., 2001).

The pulse impacted fatigue experimental principle is based on an indentation mechanism between indenter and coating spacers. The indenters of the impact tester usually are made from superhard materials such as diamond. So when the superhard thin
coatings, e.g. diamond, diamond-like carbide are evaluated with a diamond indenter, the
difficulty existing here is that brittle fracture may occur in both indentor and coating. The
conical and spherical shaped indenters develop circumferential tensile stresses, around the
region of contact, which cause brittle fracture and therefore prevent accurate measurement
of the impact indentation. Fracture often then occurs due to the high localized tensile
stresses which are developed at the tip of the indenter (Zhang, L.Y., 1997).

2.5.2.2 Pulse impacted fatigue test method using a soft impressor

Zhang has further studied the pulse impacted fatigue test method using a soft impressor
technique (Zhang, L.Y., 1997). The soft impressor method has been used to investigate the
room temperature deformation of magnesium oxide, silicon and diamond, under conditions
of cyclic loading, and to extend the method to a quantitative measurement of the
mechanical integrity of diamond coated materials.

The essential principle of the soft impressor technique is that it encourages plastic
deformation rather than fracture in hard, brittle ceramic materials. This effect can be
achieved by using an impressor in the form of a cone, made from a softer material than the
specimen, under a known normal load. As the load is applied, the initial sharp conical tip of
the softer impressor deforms plastically, the cone become blunter until its flattened tip is
large enough to support the applied load elastically.

The pulse impacted fatigue experimental method using the soft impressor technique
theoretically presents a method to evaluate superhard thin coatings' fatigue failure
resistance. However, the difficulty existing here is that the method is unsuitable to consider
the practical application conditions and the evaluation integrity of fatigue failure resistance
of coated components. Also, it is impossible to prepare for full scale tests under practical
application conditions.
2.5.2.3 Evaluation methods of rolling contact fatigue failure

Rolling contact fatigue is responsible for the failure of coated components, e.g. rolling element bearings, gears, camshafts etc. Rolling contact fatigue can have different appearances, depending on the initial causes. It ranges from surface pitting, peeling, spalling, flaking, and to section cracking. Whatever the initial causes, cyclic stress is the common factor. The requirements for improved life, reliability and load capacity are increasing as the coated component applications spread (Stewart, S. and Ahmed, R. 2002).

1. Four ball-rolling contact fatigue test

Rolling contact fatigue testing machines utilising four balls and a modified four balls configuration currently are the most widely used machines to evaluate the rolling contact fatigue performance of coated specimens. The four ball rolling contact fatigue testing machine (Wang, Y. and Hadfield, M., 1999) is shown in Figure 2.2.

It consists of an assembly that simulates an angular contact rolling element bearing. The stationary steel cup represents a bearing outer-race, three lower balls represent the rolling elements within a bearing-race and the upper ball represents the inner-race. The assembly was loaded via a piston below the steel cup, from a lever-arm load. The upper-ball was assembled to a drive shaft via a collet and contacts three lower-balls when the machine is stationary. The contacting positions between the upper ball and lower balls were immersed in lubricating oil. The tests are terminated automatically at a set number of drive-shaft revolutions measured by a tachometer. A vibration sensor automatically stops tests at a predetermined potentiometer sensitivity.
Figure 2.2 Four ball rolling contact fatigue test machine

(Wang, Y. and Hadfield, M., 1999)
2. Modified three ball-cone rolling contact fatigue test

A modified four-ball machine shown in Figure 2.3 was used to conduct the rolling contact fatigue tests to comprehend the performance of coated element cones (Ahmed, R. and Hadfield M., 1999; Ahmed, R., 2002).

![Figure 2.3 Modified three ball-cone rolling contact fatigue test machine](image)

1, Belt drive; 2, speed sensor; 3, Spindle driving motor; 4, coated cone and collet; 5, cup assembly; 6, Thermocouple; 7, thrust bearing; 8, bellows; 9, Pressure Gauge
This modification not only allowed the rotation of the planetary-balls, but also correctly modelled the kinematics of deep groove rolling element bearing, and precisely defined the contact load. The coated rolling element cone replaced the upper drive-ball, which represented the inner race of the rolling element ball bearing, and rolling contact fatigue tests were conducted. As the planetary balls were free to rotate, there was no gross sliding in the modified three-ball assembly. Rolling contact fatigue tests were conducted under immersed lubrication conditions. Failure was defined as the increase in vibration amplitude above a pre-set level.

The frictional torque measured represents the sum of the frictional torque in the three-ball cup assembly and the frictional torque due to the rolling element thrust bearing, through which load is applied to the cup assembly. The frictional torque values were recorded for the entire test duration (Ahmed, R., 2002).

3. Five ball-rod rolling contact fatigue test

Figure 2.4 shows the modified ball-rod rolling contact fatigue test machine (Dommarco, R. C., 2002). The modification was introduced by replacing three balls and retainer by five balls without retainer, sample, balls, upper and lower cups and testing load. The rotation of the sample exerts a rotational motion superimposed on a translational motion of the balls. The balls translate in the same direction as the sample rotates; therefore the number of stress cycles per sample revolution is lower than the number of balls.
Figure 2.4 Five ball-rod rolling contact fatigue test machine
4. Ball-on-cylinder rolling contact fatigue test

![Diagram of ball-on-cylinder test machine]

Figure 2.5 Rolling ball-on-cylinder test machine

A ball-on-cylinder rolling contact fatigue test machine is shown schematically in Figure 2.5. The ball rotates in a bronze socket along a fixed axis parallel to the axis of rotation of the ring, resulting in a pure rolling motion of the ball. During the test, the ring rotates against the ball (Hogmark, S., 1998). A vibration detector is mounted on the lever arm detecting the surface deterioration as a result of surface fatigue. The test stops when the vibration level increases by 15%.
5. One ball rolling contact fatigue test

![Diagram of a one ball rolling contact fatigue test machine]

Figure 2.6 One ball rolling contact fatigue test machine

A modified reciprocating sliding machine Figure 2.6 was used to conduct rolling tests on coated specimens, where a standard steel ball (Ø 12.7 mm) was loaded against a coated plate. The ball is forced to roll between the loading unit and the lower flat specimen as the loading unit moves back and forth (Podgornik, B. and Vizintin, J., 2002).

2.6 Challenge for evaluation technology of coated components

As mentioned above, evaluation technology is crucial to the design of new coating materials and the improvement of the coated component properties for meeting the demands of applications coated components. However, a general theory covering all relevant properties and parameters involved in the design and application of tribological coating composites is very far from being realised. Such a theory would have to treat the long chain of relationships ranging from the coating deposition parameters to the tribological response of the coated component. Generally, the end-users of coated
components are recommended to make the final evaluation of the tribological response in field tests or in component tests, i.e. tests where the actual component is evaluated under realistic conditions. Simplified laboratory tests often deviate from the actual situation as to nominal and real contact pressure, sliding speed, heat conductivity and capacity, ambient cooling, etc., which makes correlation to the real case problematic (Hogmark, S., 2000).

Fatigue damage is the very typical damage causing failure of coated components, and fatigue failure resistance is an extremely important evaluative criterion to the capacity of coated components. Further studying the fatigue failure resistance of coated component and improving the evaluative technologies for coated component fatigue resistance must benefit both new applications and new designs of coated components. Summaries of current fatigue failure evaluation technologies and methods are as follows:

1. Indirect fatigue method.

When coated components are subjected to dynamic loading during service and fail by a fatigue process, the indentation hardness evaluation method is difficult to directly correlate with actual coated components fatigue failure performance. Also, conventional diamond indenters are difficult to prepare accurately and easily become damaged when used on superhard material coatings like cubic boron nitride, silicon carbide and diamond itself (Zhang, L.Y., 1997).

2. Pulsed impact evaluation method using a hard impressor.

Fracture often occurs due to the high localized tensile stresses which are developed at the tip of the indenter when the pulse impact evaluation method uses hard impressors which are made form superhard materials such as diamond. The difficulty existing here is that conical and spherical shaped indenters develop circumferential tensile stresses, around the
region of contact, which cause brittle fracture and therefore prevent accurate assessment of the impact resistance under an elastic stress state.

3. Pulsed impact evaluation method using a soft impressor

The soft impressor impact technique theoretically presents a method to evaluate superhard thin coatings' fatigue failure resistance. However, the difficulty existing here is that the method is unsuitable for evaluation of the fatigue resistance in practical application conditions of coated components. Also, it is not possible to prepare this test for full-scale conditions.

2.7 Route to improve contact fatigue evaluation technology

1. Test mechanism

Obviously, a good adhesion between the substrate and coating film is crucial to the contact fatigue resistance of coated components. Any adhesion test must superimpose an external stress field over the interface between the coating and substrate to cause a measurable adhesive failure. Since this stress field will depend on the geometry and type of loading (indentation, scratching, sliding, abrasion, impact, etc.) as well as on the elastic and plastic parameters of the coating and substrate, so the evaluation results often deviate from the coated component actual fatigue situation. The rolling contact method is able to superimpose an external stress field over the interface between the coating and substrate to cause a measurable adhesive failure. However, the rolling contact evaluation method is one of several suitable evaluation methods for contact fatigue resistance of coated components. Further studies to seek rolling contact fatigue failure mechanisms and develop the evaluation method of rolling contact fatigue resistance of coated components are very necessary.
2. Test machine

There is a need to develop current test capabilities and design a new test machine which is able to evaluate real coated components, to simulate realistic conditions as to nominal and real contact pressure, sliding speed, heat conductivity and capacity, ambient lubrication and cooling, etc., and to detect and diagnose precisely the initial fatigue cracks for meeting the need of the applications of coated components.

3. Design a new test machine

As mentioned above, the rolling contact fatigue test machines, the four ball fatigue test machine and the modified four ball fatigue test machines are widely used to evaluate the rolling contact fatigue resistance of thick overlay coatings. The fatigue failure of a coated component sample is defined as the increase in the vibration amplitude which follows the formation of spalling, delamination or pitting. This increase in vibration is measured by sensors and then sent to a computer which is used to control the test machine.

Current techniques of detection and diagnosis for rolling contact fatigue initial cracks are commonly used to measure the vibration amplitude of the coated component sample using an acceleration sensor to detect the initial cracks. This method is only able to detect and diagnose the peeling of large fatigue pieces, and it is difficult to detect the micro-intermeshing cracks seen in this work. There is a need to develop current test capabilities and design a new test machine which is able to evaluate real coated components, to simulate realistic conditions as to nominal and real contact pressure, sliding speed, heat conductivity and capacity, ambient lubrication and cooling, etc., and to detect and diagnose precisely the initial fatigue cracks for meeting the need of the applications of components with thin coatings.
CHAPTER 3 Theories of rolling contact stress and mechanism of rolling contact fatigue

3.1 Theory of contact stress of components in rolling contact

3.1.1 Theory of elasticity of components in rolling contact

3.1.1.1 Characteristics of the contact problem for components in rolling contact

Usually, when a load acts on the contact zone between components in rolling contact, only small area of contact is developed between the mating members. Consequently, although loading may only be moderate, stresses induced on the surfaces of the rolling elements are usually large. Since the effective area over which load is supported rapidly increases with depth below a rolling surface, the high compressive stress occurring at the surface does not permeate the entire rolling member. Therefore, bulk failure of a rolling member is generally not a significant factor in rolling contact component design; however, destruction of the rolling surface is (Harris TA. 2001).

3.1.1.2 Elastic contact theory of rolling contact components

The loading acting between the components of rolling contact is a typical elastic contact problem. The classical solution for the local stress and deformation of two elastic bodies apparently contacting at a single point was established by Hertz (Hertz, H., 1896). Today, contact stresses are frequently called Hertzian or Hertz stresses in recognition of his accomplishment.

In that light consider an infinitesimal cube of an isotropic homogeneous elastic material subjected to the stresses shown in Figure 3.1.
Figure 3.1 Stresses acting on an infinitesimal cube of material under load
(Hertz, H., 1896)
Considering the stresses acting in the \( x \) direction and in the absence of body forces, static equilibrium requires that

\[
\sigma_x \, dz \, dy + \tau_{xy} \, dx \, dz + \tau_{xz} \, dx \, dy - \left( \sigma_x + \frac{\partial \sigma_x}{\partial x} \right) dz \, dy
\]

\[- \left( \tau_{xy} + \frac{\partial \tau_{xy}}{\partial y} \right) dx \, dz - \left( \tau_{xz} + \frac{\partial \tau_{xz}}{\partial z} \right) dx \, dy = 0 \tag{3.1}
\]

Therefore,

\[
\frac{\partial \sigma_x}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} = 0 \tag{3.2}
\]

Similarly, for the \( y \) and \( z \) directions, respectively,

\[
\frac{\partial \sigma_y}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yz}}{\partial z} = 0 \tag{3.3}
\]

\[
\frac{\partial \sigma_z}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} = 0 \tag{3.4}
\]

Equations (3.2)–(3.4) are the equations of equilibrium in Cartesian coordinates.

Hooke's law for an elastic material states that, within the proportional limit:

\[
\varepsilon = \frac{\sigma}{E} \tag{3.5}
\]
In which $\varepsilon$ is strain and $E$ is the modulus of the strained material. If $u$, $v$, and $w$ are the deflections in $x$, $y$, and $z$ directions, then

$$\varepsilon_x = \frac{\partial u}{\partial x}$$

$$\varepsilon_y = \frac{\partial u}{\partial y}$$

$$\varepsilon_z = \frac{\partial w}{\partial z}$$

(3.6)

If instead of an elongation or compression the sides of the cube undergo relative rotation such that the sides in the deformed conditions are no longer mutually perpendicular, then rotational strains are given as

$$\gamma_{xy} = \frac{\partial u}{\partial y} + \frac{\partial u}{\partial x}$$

$$\gamma_{xz} = \frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}$$

$$\gamma_{yz} = \frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}$$

(3.7)
When a tensile stress $\sigma_x$ is applied to two faces of a cube, then in addition to an extension in the $x$ direction, contractions are produced in the $y$ and $z$ directions as follows:

\[
\varepsilon_x = \frac{\sigma_x}{E}
\]

\[
\varepsilon_y = -\frac{\xi \sigma_x}{E}
\]  \hspace{1cm} (3.8)

\[
\varepsilon_z = -\frac{\xi \sigma_x}{E}
\]

Equations (3.9) were obtained by the method of superposition.

\[
\varepsilon_x = \frac{\sigma_x}{E} \left[ \sigma_y - \xi (\sigma_x + \sigma_z) \right] \cdot
\]

\[
\varepsilon_y = -\frac{\xi \sigma_x}{E} \left[ \sigma_x - \xi (\sigma_x + \sigma_y) \right]
\]  \hspace{1cm} (3.9)

\[
\varepsilon_z = -\frac{\xi \sigma_x}{E} \left[ \sigma_x - \xi (\sigma_x + \sigma_y) \right]
\]
In accordance with Hooke’s law, it can further be demonstrated that shear stress is related to shear strain as follow:

\[
\gamma_{xy} = \frac{\tau_{xy}}{G}
\]

\[
\gamma_{xz} = \frac{\tau_{xz}}{G}
\]

\[
\gamma_{yz} = \frac{\tau_{yz}}{G}
\]

In which \(G\) is the modulus of elasticity in shear, defined as

\[
G = \frac{E}{2(1 + \xi)}
\]

One can further define the volume expansion of the cube as follows:

\[
\xi = \xi_x + \xi_y + \xi_z
\]

Combining equations (3.9), (3.11) and (3.12) one obtains for normal stresses
\[ \sigma_x = 2G \left( \frac{\partial \sigma}{\partial x} + \frac{\xi}{1 - 2\xi} \epsilon \right) \]

\[ \sigma_y = 2G \left( \frac{\partial \sigma}{\partial y} + \frac{\xi}{1 - 2\xi} \epsilon \right) \]  

Equation (3.13)

\[ \sigma_z = 2G \left( \frac{\partial \sigma}{\partial z} + \frac{\xi}{1 - 2\xi} \epsilon \right) \]


3.1.2 Surface stresses analysis of rolling contact components

Using polar coordinates rather than Cartesian, Boussinesq (Boussinesq, J., 1892) solved the simple radial distribution of stress within a semi-infinite solid as shown in Figure 3.2. With the boundary condition of a surface free of shear stress, the following solution was obtained for radial stress:
\( \sigma_r = \frac{2F \cos \theta}{\pi r} \)  \hspace{1cm} (3.14)

Where \( F \) is a contributed force, \( r \) and \( \theta \) are the polar coordinates of a point.
It is apparent from equation (3.14) that as $r$ approaches 0, $\sigma_r$ becomes infinitely large. Hertz reasoned that instead of a point or line contact, a small contact area must form, causing the load to be distributed over a surface, and thus alleviating the condition of infinite stress. In performing his analysis, he made the following assumptions:

- The elastic limit of the material is not exceeded; that is, all deformation occurs in the elastic range.
- Loading is perpendicular to the surface; that is, the effect of surface shear stresses is neglected.
- The contact area dimensions are small compared to the radii of curvature of the bodies under load.
- The radii of curvature of the contact areas are very large compared to the dimensions of these areas.

For stress distribution in a semi-infinite elastic solid, Hertz introduced the assumptions:

\[ X = \frac{x}{b} \]

\[ Y = \frac{y}{b} \] \hspace{1cm} (3.15)

\[ Z = \frac{z}{b} \]
In which $b$ is an arbitrary fixed length and hence, $X$, $Y$, and $Z$ are dimensionless parameters. Also,

$$\frac{u}{c} = \frac{\partial U}{\partial X} - Z \frac{\partial V}{\partial X}$$

$$\frac{u}{c} = \frac{\partial U}{\partial Y} - Z \frac{\partial V}{\partial Y}$$

(3.16)

$$\frac{u}{c} = \frac{\partial U}{\partial Z} - Z \frac{\partial V}{\partial Z} + V$$

in which $c$ is an arbitrary length such that deformations $u/c$, $v/c$ and $w/c$ are dimensionless. $U$ and $V$ are arbitrary functions of $X$ and $Y$ only such that

$$\nabla^2 U = 0$$

(3.17)

$$\nabla^2 U = 0$$

Furthermore, $b$ and $c$ are related to $U$ as follows
\[ \frac{b}{c} = -2 \frac{\partial^3 U}{\partial Z^2} \]  

(3.18)

The foregoing assumptions, which are partly intuitive and partly based on experience, when combined with elasticity relationships (3.7), (3.10), (3.12), (3.13), (3.14), yield the following expressions:

\[ \frac{\sigma_x}{\sigma_0} = Z \frac{\partial^3 V}{\partial X^2} - \frac{\partial^3 U}{\partial X^2} - 2 \frac{\partial}{\partial Z} \frac{\partial V}{\partial X} \]

\[ \frac{\sigma_y}{\sigma_0} = Z \frac{\partial^3 V}{\partial Y^2} - \frac{\partial^3 U}{\partial Y^2} - 2 \frac{\partial}{\partial Z} \frac{\partial V}{\partial Y} \]

\[ \frac{\sigma_z}{\sigma_0} = Z \frac{\partial^2 V}{\partial Z^2} - \frac{\partial V}{\partial Z} \]  

(3.19)

\[ \frac{\tau_{xy}}{\sigma_0} = Z \frac{\partial^2 V}{\partial X \partial Y} - \frac{\partial^2 U}{\partial X \partial Y} \]

\[ \frac{\tau_{xz}}{\sigma_0} = Z \frac{\partial^2 V}{\partial X \partial Z} \]
\[
\frac{\tau_{xy}}{\sigma_0} = \frac{\partial^2 V}{\partial Y \partial Z}
\]

in which

\[
\sigma_0 = (-2Gc)/b \quad \text{and} \quad U = (1 - 2\xi) \int \sqrt{V(X,Y,\zeta)} d\zeta
\]

From the preceding formulas the stresses and deformations may be determined for a semi-infinite body limited by the \(xy\) plane on which \(\tau_{xy} = \tau_{yz} = 0\) and \(\sigma_z\) is finite on the surface, that is, at \(z = 0\).

Hertz's last assumption was that the shape of the deformed surface was that of an ellipsoid of revolution. The function \(V\) was expressed as follows:

\[
V = \frac{1}{2} \int \frac{X^2}{k^2 + S^2} - \frac{Y^2}{1 + S^2} - \frac{Z^2}{S^2} \frac{k dS}{\sqrt{(k^2 + S^2)(1 + S^2)}} \quad (3.20)
\]

in which \(S_0\) is the largest positive root of the equation

\[
\frac{X^2}{k^2 + S_0^2} - \frac{Y^2}{1 + S_0^2} - \frac{Z^2}{S_0^2} = 1 \quad (3.21)
\]
and

\[ k = \frac{a}{b} \quad (3.22) \]

Here, \( a \) and \( b \) are the semimajor and semiminor axes of projected elliptical area of contact. For an elliptical contact area, the stress at the geometrical centre is

\[ \sigma_0 = -\frac{3Q}{2\pi ab} \quad (3.23) \]

the arbitrary length \( c \) is defined by

\[ c = \frac{3Q}{4\pi Ga} \quad (3.24) \]

For the special case \( k = \infty \), then

\[ \sigma_0 = -\frac{2Q}{\pi b} \quad (3.25) \]
Since the contact surface is assumed to be relatively small compared to the dimensions of the bodies, the distance between the bodies may be expressed as

\[ z = \frac{x^2}{2r_x} + \frac{y^2}{2r_y} \]  

(3.27)

in which \( r_x \) and \( r_y \) are the principal radii of curvature.

Introducing the auxiliary quantity \( F(\rho) \) as determined by equation, this is found to be a function of the elliptical parameters \( a \) and \( b \) as follows:

\[ F(\rho) = \frac{(k^2 + 1) \mathcal{G} - 2\psi}{(k^2 + 1) \mathcal{G}} \]  

(3.28)

in which \( \mathcal{G} \) and \( \psi \) are the complete elliptic integrals of the first and second kind, respectively.

\[ \psi = \int_0^{\pi/2} \left[ 1 - \left(1 - \frac{1}{k^2} \right) \sin^2 \phi \right]^{-1/2} d\phi \]  

(3.29)
\[ \mathcal{J} = \int_{0}^{\pi/2} \left[ 1 - \left( 1 - \frac{1}{k^2} \right) \sin^2 \phi \right]^{1/2} \, d\phi \]  

(3.30)

By assuming values of the elliptical eccentricity parameter \( k \), it is possible to calculate corresponding values of \( F(\rho) \) and thus create a table of \( k \) vs \( F(\rho) \).

Brewe and Hamrock (Brewe, D. and Hamrock, B. 1977), a least squares method of linear regression, obtained simplified approximations for \( k \), \( \psi \), and, \( \mathcal{J} \). These equations are:

\[ k \approx 1.0339 \left( \frac{\mathcal{R}_x}{\mathcal{R}_y} \right)^{0.636} \]  

(3.31)

\[ \mathcal{J} \approx 1.0003 + \frac{0.5958}{\left( \frac{\mathcal{R}_x}{\mathcal{R}_y} \right)} \]  

(3.32)

\[ \psi \approx 1.5277 + 0.6023 \ln \left( \frac{\mathcal{R}_x}{\mathcal{R}_y} \right) \]  

(3.33)

The directional equivalent radii \( \mathcal{R} \) are defined by
\[ R_x^{-1} = \rho_{xI} + \rho_{xII} \]  

(3.34)

\[ R_y^{-1} = \rho_{yI} + \rho_{yII} \]  

(3.35)

where subscript \( x \) refers to the direction of the major axis of the contact and \( y \) refers to the minor axis direction.

Recall that \( F(\rho) \) is a function of curvature of contacting bodies.

\[ F(\rho) = \frac{(\rho_{III} - \rho_{I2}) + (\rho_{II} - \rho_{II2})}{\sum \rho} \]  

(3.36)

It was further determined that

\[ a = a^* \left[ \frac{3Q}{2 \sum \rho} \left( \frac{1 - \xi_i^2}{E_i} + \frac{1 - \xi_{II}^2}{E_{II}} \right) \right]^{1/3} = 0.236a^* \left( \frac{Q}{\sum \rho} \right)^{1/3} \]  

(3.37)

(for steel bodies)
\[ b = b^* \left[ \frac{3Q}{2\sum \rho} \left( \frac{(1 - \xi_1^2)}{E_1} + \frac{(1 - \xi_{II}^2)}{E_{II}} \right) \right]^{1/3} = 0.236b^* \left( \frac{Q}{\sum \rho} \right)^{1/3} \] (3.38)

(for steel bodies)

\[ \delta = \delta^* \left[ \frac{3Q}{2\sum \rho} \left( \frac{(1 - \xi_1^2)}{E_1} + \frac{(1 - \xi_{II}^2)}{E_{II}} \right) \right]^{2/3} \sum \frac{\rho}{2} \]

\[ = 2.79 \times 10^{-4} \delta^* Q^{2/3} \sum \rho^{1/3} \] (3.40)

(for steel bodies)

in which \( \delta \) is the relative approach of remote points in the contacting bodies and

\[ a^* = \left( \frac{2k^2 \vartheta}{\pi} \right)^{1/3} \] (3.41)

\[ b^* = \left( \frac{2\vartheta}{k\pi} \right)^{1/3} \] (3.42)
\[
\delta^* = \frac{2\psi}{\pi} \left( \frac{\pi}{2k^3 \varrho} \right)^{1/3}
\]  

(3.43)

values of the dimensionless quantities \(a^*, b^*,\) and \(\delta^*\) as functions of \(F(\rho)\) are given (Brewe, D. and Hamrock, B. 1977). For an elliptical contact area, the maximum compressive stress occurs at the geometrical centre. The magnitude of this stress is

\[
\sigma_{\text{max}} = \frac{3Q}{2nab}
\]

(3.44)

The normal stress at other points within the contact area is given by equation (3.45)

\[
\sigma = \frac{3Q}{2nab} \left[ 1 - \left( \frac{x}{a} \right)^2 - \left( \frac{y}{b} \right)^2 \right]^{1/2}
\]

(3.45)

3.1.3 Subsurface stresses of rolling contact components

Hertz's analysis applied only to surface stresses caused by a concentrated force applied perpendicular to the surface. Experimental evidence indicated that failure of rolling contact components in surface fatigue caused by this load emanates from points below the stressed surface (Harris TA. 2001). Therefore, it is of interest to determine the magnitude of the subsurface stresses. Since the fatigue failure of the surfaces on contact is a statistical
phenomenon dependent on the volume of material stressed, the depths at which significant stresses occur below the surface are also of interest.

3.1.3.1 Principal stresses along $Z$ axis below the contact surface

Jones (Jones, A. 1946) after Thomas and Hoersch (Thomas, H. and Hoersch, V. 1930) gives the equations by which to calculate the principal stresses $S_x$, $S_y$, and $S_z$ occurring along the $Z$ axis at any depth below the contact surface. Since the surface stress is a maximum at the $Z$ axis, therefore the principal stresses must attain maximum values there. The analysis model of principal stresses occurring along the $Z$ axis below the contact surface is shown in Figure 3.3. The surface stresses are the maximum at the $Z$ axis, and the principal stresses $S_x$, $S_y$, and $S_z$ are decreasing with the depth increasing along the $Z$ axis.

3.1.3.2 Maximum shear stresses on the $z$ axis below the contact surface

Since each of the maximum principal stresses can be determined, it is further possible to evaluate the maximum shear stresses on the $z$ axis below the contact surface. By Mohr's circle (Timoshenko, S. and Goodier, J. 1970), the maximum shear stress is found to be

$$\tau_{yz} = \frac{1}{2} (S_z - S_y)$$

(3.46)

The maximum shear stress occurs at various depths $z$; it is shown in Figure 3.4 and Figure 3.5 below the surface, being at 0.467$b$ for simple point contact and 0.786$b$ for line contact.
Figure 3.3 Principle stresses along Z axis below contact surface
Figure 3.4 Maximum shear stresses on the z axis below the contact surface

(Timoshenko, S. and Goodier, J. 1970)
Figure 3.5 Shear stresses at depths beneath the contact surface (x=y=0) (Johnson, K. 1954)
3.1.3.3 Shear stress variation amplitude below the contact surface

During the passage of a load rolling element over a point on the component surface, the maximum shear stress on the z axis varies between 0 and $\tau_{\text{max}}$. If the element rolls in the direction of the y axis, then the value from negative to positive for values of y less than and greater than zero, respectively. Thus, the maximum variation of shear stress in the yz plane at any point for a given depth is $2\tau_{\text{yx}}$. The shear stress variation amplitude below the contact surface is shown in Figure 3.6. The resulting distribution of shear stress at depth $z_0$ in the direction of rolling for $b/a = 0$ (Palmgren, A. and Lundberg, G., 1947).

3.1.4 Effect of surface shear stress

Herrtz reasoned that instead of a point or line contact, a small contact area must form, causing the load to be distributed over a surface, and thus alleviating the condition of infinite stress. In performing his analysis, he made assumptions in which the effect of surface shear stresses is neglected. Since in most rolling component applications, lubrication is at least adequate, the sliding friction between the rolling components is negligible. This means that the shear stresses acting on the rolling components in contact, that is, the elliptical areas of contact, are negligible compared to normal stresses. However, in the determination of the rolling component endurance with regard to fatigue of the contacting rolling surface, the surface shear stress cannot be neglected and in many cases is the most significant factor in determining endurance of a rolling component in a given application. The means for determining the effect on the subsurface stresses of the combination of normal and tangential (traction) stresses applied at the surface are extremely complex with regard to fatigue failure.
Figure 3.6 Shear stress variation amplitude below contact surface
(Palmgren, A. and Lundberg, G., 1947)
Zwirlein and Schlicht (Zwirlein, Q. and Schlicht, H., 1980) have calculated subsurface stress fields based upon assumed ratios of the surface shear stress to applied normal stress. Their analysis shows that as the ratio of the surface shear to normal stress increases, the maximum stress moves closer to the surface. At a ratio of \( \frac{\tau}{\sigma} = 0.3 \), the friction coefficient is 0.3, and the maximum shear stress occurs at the near surface. If a shear stress is applied at the contact surface in addition to the normal stress, the maximum shear stress tends to increase and it is located closer to the surface.

3.2 Contact fatigue mechanism

3.2.1 Progress of the contact fatigue failure

Forsyth (Forsyth, P.J., 1969), using electron microscopy, showed that slip band intrusions and extrusions occurred on the surface of metals when they were subjected to cyclic loading. The slip lines appear as parallel lines or bands within the grains. Slip band intrusions form stress concentrations which may be sites for crack development. This is primarily controlled by shear stresses rather than normal stresses.

Failure occurs without visible signs of plastic deformation and at stresses not only lower than the ultimate strength and yield point, but even below the limit of elasticity as well. Fatigue is preceded by the appearance and development of typical microcracks caused by structural features in the substrate or coating.

First, the fatigue crack develops along sliding systems located in the zone of maximum tangential stresses. The length and growth rate of the crack are small, and there no furrows at fracture. Second, the crack grows normal to applied stresses. This is a period of stationary crack growth, and its rate is proportional to the amplitude of the stress intensity coefficient. Finally, the fatigue crack is catastrophically developed, which ends with failure.
3.2.2 Material microstructure effect on the contact fatigue

Metals are crystalline in nature indicating that the atoms are arranged in an ordered manner. Most structural metals are polycrystalline and, thus, consist of a large number of individual ordered crystals or grains. Each grain has its own particular number of individual properties, ordering directional properties. Some grains are oriented such that planes of easy slip or glide are in the direction of the maximum applied shear stress. Slip occurs in ductile metals within individual gains by dislocations moving along crystallographic plans. It also occurs under both monotonic and cyclic loading (Fuchs, H.O. and Stephens, R.I., 1980).

Fatigue cracks initiate in local slip bands and initially tend to grow in a plane of maximum shear stress. This growth is quite small, usually of the order of several grains. As cycling continues, the fatigue cracks tend to coalesce and grow along planes of maximum tensile stress. The two stages of fatigue crack growth are called stage I and stage II (Fuchs, H.O. and Stephens, R.I., 1980). A fatigue crack initiates and grows across several grains controlled primarily by shear stresses, and grows in a zigzag manner essentially perpendicular to, and controlled primarily by, the maximum tensile stress. Most fatigue mechanism in high strength or brittle metals may not contain slip bands. Microcracks are often formed directly at discontinuities, such as inclusions or voids, and then grow along planes of maximum tensile stresses.

3.2.3 Position of which contact fatigue failure occurs

As explained earlier, the stresses distributions in contacts between smooth surfaces can be estimated according to Hertz (Hertz, H. 1896). Since the maximum shear stress is generated below the contact surface (and the depth of the maximum shear stress form the
surface will change with the friction force acting on the contact surface) the depth of the initial fatigue crack also changes with the depth of the maximum shear stress to the surface.

Lundberg et al. (Lundberg, G. and Palmgren, A. 1947) postulated that it is the maximum orthogonal shear stress $r_o$ that initiates the crack and that this shear stress occurs at a related depth below the surface. The point which a fatigue crack commences to propagate at is named a weak point of fatigue failure.

Fatigue is interpreted as gradual accumulation of damage leading to a change in properties, formation and development of cracks and failure. Under the repeated action of varying loads applied to locations of violent increase in stresses due to holes, turning, inclusions, and cracks, etc., sudden fractures of fatigue failure occur. A place of fatigue fracture usually coincides with the zone of stress concentration caused by a major change in a cross-section (Tushsky, L., 2002).
CHAPTER 4 Mechanisms and models of rolling contact fatigue of coated components

4.1 Introduction

4.1.1 Evaluation technologies for rolling contact fatigue resistance of coated components

Evaluation technologies are crucial for the design of new coating materials and the improvement of coated components. However, a general theory covering all relevant properties and parameters involved in the design and application of tribological coating composites is very far from being realised. Such a theory would have to treat the long chain of relations ranging from the coating deposition parameters to the tribological response of coated components. Generally, the end-users of coated components are recommended to make the final evaluation of the tribological response in field tests or in component tests, i.e. tests where the actual component is evaluated under realistic conditions to fulfil the needs of applications of coated components. Thus, versatile and reliable techniques for evaluation of coated components will continue to be important for new coating composite designs and their applications. However, the rolling contact evaluation method is one of the most suitable evaluation methods for contact fatigue resistance of coated components. Further studies characterize and categorize rolling contact fatigue failure mechanisms and develop the evaluation method of rolling contact fatigue resistance of coated components are very necessary.
4.1.2 Two important evaluation parameters of rolling contact fatigue resistance of coated components

1. Adhesion of the interface of coated components

Obviously, a good adhesion to the substrate is a crucial property of coated component applications. Any adhesion test must superimpose an external stress field over the interface between the coating and substrate to cause a measurable adhesive failure. Currently, evaluation methods of adhesion will only be representative of the particular test from which it has been obtained, and then it often deviates from the coated component’s actual situation. A preferred method is needed urgently to evaluate the practical adhesion of coated components.

2. Integrity of the coating of coated components

The mechanical integrity is an important evaluation parameter to evaluate the overall mechanical properties of the whole working surface of a coated component. The quality of superhard coating of a coated component can be assessed in many ways: i.e. composition; bonding; structure; surface topography; defect density and mechanical strength. However, whilst it is relatively easy to deposit hard and superhard materials from the vapour phase, good adhesion has been difficult to achieve, and therefore the mechanical integrity of the coating substrate interface is probably the most important single criterion (Zhang, L.Y., 1997).

Comparing with the bulk components, the adhesion of the interface of coated components and the integrity of the coating of coated components are extremely important property criteria for coated components
4.1.3 Routes to improve fatigue evaluation technologies of coated components

Fatigue damage failure is the very typical failure mode of coated components, and the load capacity of coated components usually is limited by the fatigue life of coated components. Further study and improvement of the evaluative technologies for contact fatigue resistance of coated component must benefit both the design and application of coated components. The routes to improve the contact fatigue evaluation technologies of coated component are suggested as follows:

The rolling contact mechanism is close to the application situation of actual components. The fatigue failure mechanism in rolling contact of coated components should be further studied to simulate actual practical application conditions of coated components. By rolling contact fatigue failure mechanisms, to develop current test machines and design the new test machine which is able to evaluate real coated components, to simulate realistic conditions as to nominal and real contact pressure, sliding speed, heat conductivity and capacity, ambient lubrication and cooling, etc., and to detect and diagnose precisely the initial fatigue cracks to meet the needs of the applications for coated components.

4.2 Mechanism and model of rolling contact fatigue of coated components

4.2.1 Mechanism model of rolling contact fatigue

1. Model

The mechanism model of a coated component rolling contact is shown in Figure 4.1. The radial load N and tangential load T act on the rolling element 1, and the rolling element 1 rolls on the surface of the coated component at the angular speed \( \omega \). As a rolling element under compressive load travels over the surface of the coated component, the material in
the forward portion of the contact surface that is in the direction of rolling will undergo compression while the material in the rear of the contact is being relieved of stress. Coated material in the contact zone is squeezed up to form a bulge in the forward portion of the

Figure 4.1 Coated Component rolling contact mechanism model

1-Roller, 2-Coating, 3- Coated component substrate
contact. Subsequent depression is formed in the rear of the contact area. Such an additional
tangential force or an additional distribution of shear stress \( \tau_f \) is required to overcome the
resisting force of the bulge.

2. Mechanism

a. Rolling contact stresses

i. Surface contact pressure

The surface contact pressure of a test coated component is shown in Figure 4.2. Normal
stress distributes at different levels below the contact surface. The discrete localised elastic
and plastic normal load distributions associated to asperity contacts are gradually
smoothened below the surface. The pressure peaks at the coating interface may lead to local
plastic deformation of the substrate, despite the average stress level being rather low.

ii. Maximum normal stress \( \sigma_{\text{max}} \)

Maximum normal stress occurs at the geometrical centre of the rolling contact surface
of a coated component; see Chapter 3, equation (3.44). The magnitude of this stress is

\[
\sigma_{\text{max}} = \frac{3Q}{2\pi ab}
\]  

(4.1)

iii. Maximum shear stress \( \tau_0 \)

In Chapter 3, by Mohr's circle (Timoshenko, S. and Goodier, J. 1970), the maximum
shear stress on \( z \) axis at \( z \) point is found to be

\[
\tau_{zx} = \frac{1}{2} (S_z - S_y)
\]  

(4.2)
iv. Maximum shear stress depth $z_0$

As shown in Chapter 3, the maximum shear stress $\tau_0$ occurs at depth $z_0$, below the surface, being at $0.467b$ for simple point contact and $0.786b$ for line contact.

v. Maximum shear stress amplitude
As stated in Chapter 3, during the passage of a loaded rolling element over a point on the component surface, the maximum shear stress on the z axis varies between 0 and $\tau_{\text{max}}$. If the element rolls in the direction of the y axis, then the value ranges from negative to positive for values of y less than and greater than zero, respectively. Thus, the maximum variation of the shear stress in the yz plane at any point for a given depth is $2\tau_{yx}$, and the maximum variation of the shear stress in the yz plane at the $z_0$ point is $2\tau_0$.

vi. Additional shear stress $\tau_f$

An additional distribution of shear stress $\tau_f$ is produced to overcome the resistance of the bulge. The value of shear stress $\tau_f$ changes with the conditions of the lubrication and load, and the tribological properties of the surface materials.

b. Rolling contact fatigue mechanism of coated components

The mechanism model of a coated component rolling contact is shown in Figure 4.1 and the pressure distribution of the coated component contact surface is shown in Figure 4.2. The radial load N and tangential load T act on the rolling element 1, and the rolling element 1 rolls on the surface of the coated component at the angular speed $\omega$. The normal stress is distributed at different levels below the contact surface. The weakest point in the contact location of the coated component acted on by the maximum variation of shear stress $\tau_0$ produces the initial fatigue crack, and the initial crack gradually accumulates leading to a change in properties, formation and development of the crack and failure. Under the repeated action of varying loads applied at locations of a drastic increase in stresses due to holes, turning, inclusions, and cracks. etc., sudden fractures due to fatigue failure occur.

4.2.2 Characteristics of the mechanism model of rolling contact fatigue

The characteristics of the mechanism model of rolling contact fatigue were as follows:
1. Suitability to the realistic application conditions of coated components

End-users of coated components are recommended to make the final evaluation of the tribological response in field tests or in actual component tests, i.e. tests where the actual component is evaluated under realistic conditions. This mechanism model of the coated component fatigue resistance is close to the realistic application conditions of coated components. Further, the real coated bearings can directly be evaluated by this method.

2. Practicability for the simulation of actual load situation

As seen in Figure 4.1, the radial test load $N$ is easily designed and controlled according to the application conditions. The tangential friction $T$ can be changed by controlling the lubrication situation of the contact surface. Therefore it is convenient to simulate the real load conditions of a coated component.

3. Avoid premature cracking of hard and superhard coatings

First, the materials of the mating rolling elements using the evaluation test can be chosen freely to meet the demands of the applications. Second, the variations of stresses acting in the contact zone are smooth when the roller rolls into and out of the contact range. The range of maximum shear stress is from $-\tau_0$ to $\tau_0$, smoothly, so the production of the initial fatigue crack is controllable.

4. Improvement of interface adhesion evaluation

The maximum shear stress occurs at various depths $z$ below the surface, as shown in Chapter 3, Figure 3.4 and Figure 3.5, being at $0.467b$ for a simple point contact and $0.786b$ for a line contact. The maximum shear stress is close to the interface between the coating and substrate, therefore, it is easier to detect the defects at the interface.

5. Improvement of the coating integrity evaluation
According to the mechanism model of the rolling contact fatigue, the evaluation method of rolling contact fatigue is able to carry out the integrity evaluation to the full-scale contact surface of a coated component. However, the pulse impacted method and indentation method are both "point" evaluation methods, they are therefore unsuitable to obtain the evaluation result of coating integrity for coated components.

4.3 Mechanism model of accelerated rolling contact fatigue

Rolling contact fatigue tests are historically very time consuming. Researchers have been seeking consistent methods to accelerate rolling contact fatigue tests in a manner representative of the true environment of coated components, efficient methods to accelerate rolling contact fatigue test have very significant benefit (Wang, L., 2000; David J. Mitchell, 2000). One of the accelerated fatigue test methods is the increase of the test speed, and another is the increase of Hertzian contact stress. Both have to be limited by the capacity of the test machine. In the following section, a new test method for accelerated rolling contact fatigue is presented according to the mechanism of rolling contact fatigue using a lubricant containing micro-particulates added.

4.3.1 Rolling contact failure analysis under micro-particulates added

1. Analysis of micro-particulates contacts

The micro-particulates contacts, a small contact area between an abrasive grain and the contact zone, can cause considerable stresses. When the produced stresses exceed the elastic limit but still remain lower than the yield limit, fatigue failure take place; if the stresses are above the yield point, the wear is accompanied by plastic deformation of microvolumes (microcutting) and post deformation failure occurs (Tokarev, AO., 2000).
Therefore, the characteristic of the micro-particulates contacts provides a route for seeking the accelerated fatigue test method.

2. Real contact area with micro grains present

The mean area of the real contact points is affected by the existence of the embedded abrasive grains and the surface topographical features of the surface layer which is harder than the bulk material of the coated component. The mean cross-sectional area of the grooves on the contact surface of a coated component is calculated from the profile curve measured perpendicular to the sliding direction. The mean area of the real contact points is a factor that has a direct effect upon the contact stresses of the coated component and the failure performances of the coated components (Hisakado, T. et al., 1993).

3. Squeezing mechanism of the micro grain abrasive

Between rolling element contact surfaces hard micro grains are squeezed into the metal surface of rolling element and accompanied by deformation which results in failure of metal microvolumes and formation of wear particles. Being embedded into the wear surface, the solid grain tends to shift the metal of the contact point by repeated deformation or brittle chipping depending on the hardness. In these conditions of solid grains-wear surface interaction shearing becomes the leading process in the formation of wear debris, and resistance to shearing or breaking off becomes the main criterion for wear resistance (Tokarev, A.O., 2000).

4.3.2 Effects of micro-particulates on the microstructural defects

As intrinsic properties of coated component materials, the microstructural defects of coated components take a very important role in the coated component fatigue resistance. The microstructural defects of coated components involve micro-crack defects, coating film
integrity defects and interface defects. Micro-cracks exist on the surface, subsurface, interface and substrate of a coated component. When a cyclic load acts on the micro-crack, it grows and spreads; finally, the coated component fatigue occurs. Residual stresses can result in the micro-crack defect. Interface integrity defects can be caused by the discontinuous compound between coating film and substrate. The interstices or the weak bonds exist on the interface of coated components. Coating film and substrate include impurity and interstice to reduce the capacity of rolling contact fatigue resistance of coated components. The microstructural defects of coated components are normally at the micrometre and nanometre scale, so they are not sensitive for common fatigue evaluation methods, i.e. the impact method, sliding method and rolling contact method. The main reason is that the real contact area is far more than the size of the microstructural defects. However, the effects of micro-particulates on the microstructure defects of coated components are very sensitive.

4.3.3 Effects of micro-particulates on rolling contact failure

Contamination of the lubricant by particulate debris is one of the most common failure initiators in operating bearing systems (David J. 2000; Williams, J.A., 1992; Sayles, R.S., 1988; Mitchell, D.J., 1999). Particulate contaminated lubricants enable accelerated wear testing without increasing the Hertzian contact stress to a level which may change the wear and failure mechanisms relative to the stress environments experienced in operation (Mitchell, D.J., 1997). Coating material properties, lubricant type, particulate contaminant composition, size distribution, hardness, elastic modulus and morphology are interrelated factors which determine the combination of wear mechanisms which will result (Salehizadeh, H. 1992; Wedeven, L.D. 1995; Xuan, J.L., 1989). The particulate
contaminants were added to the lubricant both to evaluate their capability of reducing the test time and to study the wear mechanisms they induce.

4.3.4 Mechanism model of accelerated rolling contact fatigue of coated components

1. Mechanism model of accelerated rolling contact fatigue

The mechanism model of accelerated rolling contact fatigue of coated components is shown in Figure 4.3. The radial load \( N \) and tangential load \( T \) act on the rolling element 1, and the rolling element 1 rolls on the surface of the coated component at the angular speed \( \omega \). The lubricant containing micro-particulates added is provided into the contact zone between the rolling element and test coated component.
Figure 4.3 Mechanism model of accelerated rolling contact fatigue
1-Roller, 2-Coating, 3-Coated component substrate
2. Mechanism of accelerated rolling contact fatigue

The mechanism model of rolling contact fatigue of coated components is shown in Figure 4.3. The lubricant containing added micro-particulates is provided into the contact zone between the rolling element and the test coated component. The contact area of the real contact is the sum of contact area of all real contact points, which is affected by the existence of the embedded abrasive grains and the surface topographical features of the surface layer which is harder than the bulk material of the coated component. Normal stress distributes at different levels below the contact surface. The contact points of micro grains contact zone of the coated component are acted on by the maximum shear stress. The defect points mating to the contact points are acted on by the variation of the maximum contact stresses, and produce the initial fatigue cracks, and the initial cracks are gradually accumulated leading to a change in properties, formation and development of the failure crack. Under the repeated action of cyclic loads applied to locations of drastic increase in stresses due to defects e.g. holes, turning, inclusions, and cracks. etc., sudden of fatigue fracture failures occur.

The initial fatigue crack first occurs in the microstructural defect point. Micro-grains in the contact zone act directly on the defect points of the contact zone of the coated component, the concentration stresses of defect points occur and cause the initial defects of the coated component to be spread.

The micro-grains between the roller and the test coated component are squeezed into metal of the contact surface of the test coated component, and the contact points deform, which results in failure of the coated component. Being embedded into the contact surface of the coated component, the solid grains tend to shift the metal of the contact points by repeated deformation, or brittle chipping, depending on the hardness. The micro-grains
between contact surfaces will change the contact state of coated components. The contact area and contact stress contribution will be related to grain number, grain material properties and grain geometry. Under the condition of micro-grain media contact, a small contact area between an abrasive grain and contact surface of coated component will cause considerable stresses which results in initial fatigue cracks of the coated component. Two situations are possible: when the produced stresses exceed the elastic limit but still keep lower than the yield limit, fatigue failure takes place; if the stress is above the yield point, the wear is accompanied by plastic deformation of microvolumes (micrcutting) and post deformation failure occurs.

4.3.5 Characteristics of mechanism model of accelerated rolling contact fatigue

Characteristics of the mechanism model of accelerated rolling contact fatigue are as follows:

1. The lubricant containing added micro-particulates is provided into the contact zone between the roller and coated component. Added micro-particulates take an important role of transmission and intensification of the load and stresses. The initial fatigue cracks are produced in a short time.

2. The hard grains act on the contact surfaces of the coated component and make the defects of the weak points of the coated component spread and result in the dislocation deformation or initial microcracks.

3. By selecting the mechanical property parameters and geometry parameters of the micro-particulates, the different typical failures of a coated component will be gained.
CHAPTER 5 Design of rolling contact fatigue test machine

5.1 Introduction

As stated in Chapter 2, current evaluation methods of the contact fatigue failure, e.g. the indirect evaluation methods, the pulsed impact methods and the sliding methods cannot avoid that premature cracks occur before the initial fatigue cracks when superhard coated specimens are tested. Four ball and modified four ball methods have to be simplified laboratory tests, and the evaluation results often deviate from the actual situation as to nominal and real contact pressure, sliding speed, heat conductivity and capacity, ambient cooling, etc., which makes correlation to the real case problematic. Disappointing evaluation results due to errors in detection and diagnosis often occur in the contact fatigue testing of coated components.

Current techniques of detection and diagnosis using in rolling contact fatigue test machines for rolling contact fatigue initial cracks are commonly used to measure the vibration amplitude of the coated component sample using an acceleration sensor to detect the initial cracks. This method is only able to detect and diagnose the peeling of large fatigue pieces, and it is difficult to detect the micro-intermeshing cracks.

This Chapter reports the design and development of a new test machine for rolling contact fatigue, for the realization of the characteristic functions of the mechanism models of the rolling contact fatigue and accelerated rolling contact fatigue, for the evaluation of real coated components, for the simulation of the realistic conditions as to nominal and real contact pressure, sliding speed, heat conductivity and capacity, ambient lubrication and cooling, etc., and for the precise detection and diagnosis of the initial fatigue cracks. Therefore, this permits further study of the test mechanisms of rolling contact fatigue.
resistance of coated components using this newly design test machine of rolling contact fatigue resistance as urgently demanded for the applications of coated components.

5.2 Design of rolling contact fatigue test machine

The 51306 coated bearing is chosen as a typical evaluation component for the design of the test machine. The main design parameters of the rolling contact fatigue test machine will be obtained according to the characteristic parameters of the 51306 coated bearing. The rolling contact fatigue test machine consists of the mechanical transmission system, control and measurement system, lubricant and cooling system and computer data treatment system.

5.2.1 Mechanical transmission system

The diagram of the mechanical transmission system is shown in Figure 5.1 (a) and the photograph is shown in Figure 5.1 (b). The mechanical transmission system comprises the load unit, transmission unit and quick-stop unit.

1. Load unit

The test load $N$ is put on the specimen 2 and 6 by the weight $Q$ with a 1:6 load lever. The range of weight $Q$ is decided by the limit capability of the 51306 coated bearing.

2. Transmission unit

The main motor 7 transfers the power and motion to the test specimen 2, test rollers 5 and test specimen 6.

3. Quick-stop unit

The quick-stop unit comprises the quick-stop cam 1, quick-stop contact switch 3 and friction clutch 4. When the friction torque between the specimen 2 and rollers 5 is over the limit of the friction torque $T_f$ which acted on specimen 2 by the friction clutch 4, the quick-
stop cam 3 begins turning with the specimen 2, and the quick-stop contact switch 3 will be turned on, and then the main motor is braked to quick-stop.

Figure 5.1 (a) Diagram of mechanical transmission system

1-quick-stop cam, 2-specimen, 3-quick-stop contact switch, 4-friction clutch, 5- roller, 6-specimen 7-main motor, 8-weight, 9-load lever
Figure 5.1 (b) Photograph of mechanical transmission system
5.2.2 Control and measurement system

The diagram of the control and measurement system is shown in Figure 5.2 (a) and the photograph of the control and measurement system is shown in Figure 5.2 (b). The control and measurement system comprises the control unit and the measurement unit.

5.2.2.1 Measurement system

The main measurement parameters of the test machine of rolling contact fatigue include the temperature, friction torque, vibration, and speed of the test specimen. The signals of these parameters are captured individually by temperature sensor, friction torque sensor, vibration sensor and speed sensor and transferred to the computer through PicoScope data conversion. The data of temperature, friction torque, vibration, and speed of the test specimen are measured, treated and saved on-line by computer using the PicoScope software manufactured by Pico Technology Ltd. The charts of on-line test data records of the temperature, friction torque, vibration, and speed of the test specimen are shown in Figure 5.3.

5.2.2.2 Control system

The control system comprises the computer, main motor, motor speed controller, main magnetic switch and quick-stop contact switch. The control system can realize the computer program control and adjust.

1. Control and adjustment of the motor speed

The main motor working speed can be controlled and adjusted by the motor speed controller. The adjustable range of the main motor speed is in the range 0 to 3000 revs/min, providing the test specimen with an adjustable speed in the 0 to 6000 range revs/min, by the one degree belt transfer.
Figure 5.2 (a) Diagram of the control and measurement systems
Figure 5.2 (b) Photograph of the control and measurement systems
Figure 5.3 Test record charts on line of temperature, friction torque, vibration and turning speed of the test specimen
2. Automatic quick-stop of the test machine

The control system can realize the function of automatic quick-stop of the test machine, as soon as the one of the test parameters of temperature, friction torque, vibration and turning speed of the test specimen is over the limit value. First, the limit values of the temperature, friction torque, vibration and turning speed of the test specimen are input to the computer. Then, the sensor's transfer, on-line, the measuring data to the computer through the PicoScope data conversion. As soon as the one of the measured data parameters is over its limit value given, the computer will carry out the quick-stop program, and the main magnetic switch turns off, and then the main motor is braked to quick-stop, whilst the alarm turns on.

3. The diagnosis and detection of the initial fatigue crack of the test specimen

a. Vibration method

The vibration sensor is fixed on the cell of the test specimen to measure, on-line, the vibration value of the specimen. When the initial fatigue crack appears on the surface of the test specimen, the roughness of the specimen surface becomes poor and it results in the increase of the vibration of the test specimen. As soon as the increased magnitude of the vibration of the test specimen gets over the limit value given, the computer starts the quick-stop program and then brakes the main motor to quick-stop it.

b. Method of the friction torque

The friction torque sensor is fixed on the friction clutch which mates with the test specimen, and the friction torque which is transferred from the test specimen to the friction clutch is measured on-line. The limiting magnitude given of the friction torque of the test specimen is input into the computer. When the initial fatigue pitting crack appears on the
surface of the test specimen, the roughness of the contact surface of the test specimen becomes poor and results in the increase of the friction torque between the test specimen and the test rollers. As soon as the increase magnitude of the friction torque of the test specimen gets over the limit value, the computer starts the quick-stop program and then brakes the main motor to quick-stop it. Whilst, as soon as the increased magnitude of the friction torque of the test specimen gets over the limit value given, the cam starts turning with the specimen relatively to the friction clutch and then the contact switch turns off the main switch to brake the main motor and to quick-stop it.

5.2.3 Lubricant and cooling system

5.2.3.1 Lubrication system

The diagram of lubrication and cooling system is shown in Figure 5.4 (a) and the photograph is shown in Figure 5.4 (b). The lubricants are provided into the cell of the test specimen by two lines. One provides the pure lubricant and another provides the lubricant containing micro-particulates. The lubrication drops free into the cell of the test coated component. The lubrication which drops into the cell of the test specimen is acted on by the centrifugal force into the contact zone between the test specimen and the test rollers; excess lubricant is recirculated from the contact zone back to the lubricant container.

1. Pure lubrication

The pure lubrication line is used in the normal rolling contact fatigue resistance test. The speed of lubricant drops is adjusted by the valve 14.

2. Lubrication containing added micro-particulates

The lubrication line containing added micro-particulates is used in the accelerated rolling contact fatigue test. The speed of lubricant drops is adjusted by the valve 17.
Figure 5.4 (a) Diagram of lubrication and cooling systems

1-quick-stop cam, 2-specimen, 3-quick-stop touch switch, 4-friction clutch, 5-roller, 6-specimen 7-main motor, 8-weight, 9-load lever, 10-13-14-16-valve, 11-17-grain lubricant container, 12-15-pure lubricant container, 18-cooling machine
Figure 5.4 (b) Photograph of lubrication systems
Figure 5.4 (c) Photograph of the cooling systems
5.2.3.2 Cooling system

The diagram of lubrication and cooling system is shown in Figure 5.4 (a) and the photograph of the cooling system is shown in Figure 5.4 (c). The cooling machine 18 provides the cool air into the cell of the test specimen for keeping the temperature of the test specimen in the permitted range.

5.3 Characteristics of the test machine

1. Suitability to evaluate the superhard material coatings

First, on the basis of the mechanism of the rolling contact, the rolling contact fatigue resistance test can avoid the production of the shock stress resulting in the premature fragile cracks. Second, for the lubricant media used in the rolling contact fatigue resistance test, the oil film provides a cushion to prevent premature fragile cracks during the variety of contact stresses acting on the test coated component. So the rolling contact fatigue test machine is suitable to evaluate the superhard coated components.

2. Suitability to evaluate the interface adhesion

The interface adhesion property is a crucial property of coated component applications. Currently, the evaluation techniques for the interface adhesion of the coated components are not enough realised (Wang Dong F., 2001; Hogmark, S., 2000). There is some improvement to use the rolling contact fatigue resistance test machine to evaluate the interface adhesion property of the coated components. On the basis of the mechanism of the rolling contact, the maximum shear stress occurs under the contact surface of the coated component. The depth from the surface of the maximum shear stress decreases as the surface friction increases. For the rolling contact fatigue resistance test machine can use lubricants of varying viscosity to adjust the surface friction and to make the maximum
shear stress near the interface, the rolling contact fatigue resistance test method should be suitable to evaluate the interface adhesion property of the coated components.

3. Simulation of the actual practical application conditions

Simplified laboratory tests often deviate from the actual situation as to nominal and real contact pressure, sliding speed, heat conductivity and capacity, ambient cooling, etc., which makes correlation to the real case problematic. The end-users of coated components are recommended to make the final evaluation of the tribological response in field tests or in component tests, i.e. tests where the actual component is evaluated under realistic conditions. Since the rolling contact fatigue resistance test machine is designed on the basis of actual application conditions of the 52306 coated bearing, the rolling contact fatigue resistance test machine can fulfil the simulation of load, speed and lubrication conditions of actual practical application conditions in the rolling contact fatigue resistance test of coated components.

4. Properties of the detection and diagnosis of the initial fatigue pitting cracks

The precision and sensitivity to detect and diagnose the initial fatigue pitting cracks of the coated component are improved since the rolling contact fatigue resistance test machine not only applies the diagnosis technique of the initial fatigue crack according to variation of the vibration signal, but also applies the diagnosis technique of the initial fatigue crack according to variation of the friction torque signal. Practically, the vibration signal is not sensitive to the micro pitting cracks on the contact surface of the coated component, but the variation of the friction torque signal is. Furthermore, the rolling contact fatigue resistance test machine applies the technique of the quick-stop, so it is possible to preserve the state of the initial fatigue crack. Therefore the rolling contact fatigue resistance test machine can precisely detect and diagnose the initial fatigue pitting cracks of the test coated components.
5. Advantages of the accelerated rolling contact fatigue test

The accelerated rolling contact fatigue test techniques possess very significant benefit, for the techniques can save some time and cost spent on the fatigue test of the coated components. The rolling contact fatigue resistance test machine is designed including two lubrication lines. The accelerated rolling contact fatigue resistance test can be divided into two steps: i) on the basis of the mechanism of micro-particulates accelerated rolling contact fatigue, the lubricant containing micro-particulates added is used in the test and the initial micro cracks are quickly produced on the contact surface of the test coated component; ii) instead of the lubricant containing added micro-particulates, the pure lubricant is used in the test again, and the initial micro cracks are spread as gradual accumulation, and then the fatigue cracks occurs. For the accelerated rolling contact fatigue test can make the initial micro cracks to be quickly produced, consequently, the rolling contact fatigue resistance test machine provides a new route for accelerated fatigue testing.

6. Advantages of the on-line measurement and remote control of the computer

The rolling contact fatigue resistance test machine is designed to possess the properties of remote control and data collection, therefore the rolling contact fatigue resistance test machine has the advantages of the convenience and practicality.
CHAPTER 6 Experiments of rolling contact failure behaviour of coated bearings

6.1 Introductions

6.1.1 Applications of rolling bearings

In this modern age of deep-space exploration and cyberspace, many different kinds of bearings have come into use, such as gas film bearings, foil bearings, magnetic bearings, and externally pressurized (hydrostatic) bearings (Harris TA. 2001). They are used in particularly demanding applications such as aircraft gas turbine main shaft ball bearings, cruise missile turbine shaft bearings, target drone engine bearings, and rocket propellant pump bearings. (Wang, L., Snidle R. W. and Gu, L., 2000).

Rolling bearings are typical tribological components. They utilize rolling contacts between the rolling elements and raceways to support load while permitting constrained motion of one body relative to another. Due to their wide availability and versatility, rolling bearings are, perhaps, the most widely used bearing type.

Rolling bearings are divided into ball bearings and roller bearings, also they are divided into radial bearings and thrust bearings respectively. Thrust rolling bearings are designed for thrust loads. The standard configuration of the thrust rolling bearing comprises upper and lower rings, and a retainer or cage to maintain a proper annular spacing between the rolling elements, as shown in Figure 6.1 and Figure 6.2. Due to the difference in the contact area, the thrust roller bearing can undergo larger loads than the thrust ball bearing, but its structure is more complex and not suitable for higher speed.
6.1.2 Thrust bearings and its failure performance

Rolling contact fatigue is a phenomenon of great practical significance to many engineering applications involving concentrated contacts. In particular, the useful life of rolling bearings is often limited by rolling contact fatigue failure. Pit formation and surface peeling resulting from rolling contact fatigue are often serious problems in bearing applications. It is widely recognized that rolling contact fatigue is one of the most
predominant failure modes in rolling element bearings. Rolling contact fatigue is associated with the formation of a spall or pit on the rolling surfaces during cyclic loading. Studies indicate that fatigue spalling can initiate either beneath the contacting surface, where the maximum shear stress occurs, or at the surface, where defects such as grinding marks, corrosion pits, etc. exist.

The application of hard thin coatings to bearing surfaces is an approach which can potentially improve rolling contact fatigue life and wear resistance of bearings. Consequently, surface modification processes and hard coating materials such as TiN, TiC, CrN, BN, etc. have been the subject of study for many researchers over the past decade (Averbach, B.L. et al. 1990).

In this Chapter, the 51306 standard thrust ball bearings and 51306 coated thrust ball bearings (with only the raceways coated) were used in the test. The reasons why the 51306 coated thrust ball bearings (with only the raceways coated) were chosen for the test bearings are as follows:

1. The 51306 thrust ball bearing is a typical flat rolling contact component. Rolling contact fatigue test results from 51306 thrust ball bearings have a representative value for other rolling contact components, e.g. gears, cam, and camshafts.

2. The bearing raceway is a weak element for rolling contact fatigue resistance of a ball bearing. Currently, the balls can be made from ceramics such as SiC for increased fatigue life and improved chemical properties. Therefore, the improvement of rolling contact fatigue behaviour of the track becomes very important.

3. The 51306 thrust ball bearing raceway has a suitable size and is easy to coat.
6.1.3 \( L_{10} \) fatigue-life and \( L_{50} \) fatigue-life

1. S-N curve

Commonly, steels have an endurance limit \( \sigma_N \), as shown in Figure 6.3.

![S-N curve diagram](image)

Figure 6.3 S-N (i.e. stress versus log of number of cycles) curves for mild steel

The endurance limit is a level of cyclically applied, reversing stress, which, if not exceeded, the structure will accommodate without fatigue failure.

2. Rolling bearing fatigue life distribution

However, the bearings appear to have dispersed characteristics in their fatigue lives. Even if a population of apparently identical rolling bearings is subjected to identical load, speed, lubrication, and environmental conditions, all the bearings will not exhibit the same life in fatigue. Instead the bearings fail according to a dispersion such as that presented in Figure 6.4. It indicates that the number of revolutions a bearing may be accomplished with
100% probability of survival, probability of survival is 1, if fatigue failure, it is zero. Alternatively, the probability of any bearing in the population having infinite endurance is zero. For this model, fatigue is assumed to occur when the first crack or spall is observed on a load-carrying surface. Since such a life dispersion exists, bearing users have chosen to use two points on the curve to describe bearing endurance (Harris TA. 2001). These are:

a. $L_{10}$ fatigue-life, that is, the life that 90% of the bearing population will endure.

b. $L_{50}$ fatigue-life, that is, the life that 50% of the bearing population will endure.

---

**Figure 6.4 Rolling bearing fatigue life distributions**
6.2 Experiment preparative

6.2.1 Test machine

The new rolling contact fatigue test machine was designed and manufactured on the basis of the mechanism models of rolling contact fatigue and accelerated rolling contact fatigue as shown in Chapter 4 and Chapter 5. The mechanical transmission unit used was a modified Unisteel rolling fatigue testing machine made by Distington Engineering Co. Ltd. The control system, measurement system and lubrication system were manufactured in the workshop of the University of Hull according to a design developed by myself. A photograph of the rolling contact fatigue test machine is shown in Figure 6.5.

Figure 6.5 Rolling contact fatigue test machine
6.2.2 Test coated bearings

The 51306 coated bearings and 51306 standard bearings were tested in the rolling contact fatigue experiments. The coatings of the coated bearings are composite materials of different kinds consisting of nanometer-sized metal carbide crystallites and an amorphous carbon-based matrix. These coatings were obtained using a closed field, unbalanced magnetron deposition system sputtering metal, boron carbines and carbon targets using Ar and acetylene gases. The characteristics of the coatings that were applied to the test bearings are shown in Table 6.1.

Table 6.1 Properties and parameters of the test bearings

<table>
<thead>
<tr>
<th>Name</th>
<th>Indentation Modulus</th>
<th>Hardness (GPa)</th>
<th>Friction Coefficient</th>
<th>Coated Method</th>
<th>Coating Thickness (μm)</th>
<th>Coating Composition</th>
</tr>
</thead>
<tbody>
<tr>
<td>CT1</td>
<td>100</td>
<td>~10</td>
<td>0.08</td>
<td>PVD</td>
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<td>nm-TiC/a-C</td>
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<tr>
<td>CT2</td>
<td>120</td>
<td>~12</td>
<td>0.12</td>
<td>PVD/CVD</td>
<td>~2.5</td>
<td>nm-TiC/a-C</td>
</tr>
<tr>
<td>CT3</td>
<td>145</td>
<td>~14</td>
<td>0.16</td>
<td>PVD/CVD</td>
<td>~3</td>
<td>nm-WC/a-CH</td>
</tr>
<tr>
<td>CT4</td>
<td>250</td>
<td>~25</td>
<td>0.25</td>
<td>PVD</td>
<td>~2</td>
<td>B4C</td>
</tr>
<tr>
<td>Substrate</td>
<td>200</td>
<td>~7</td>
<td></td>
<td>Standard 51306 Bearing Raceway</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

6.2.3 Calculation of the internal loads and predicted fatigue lives of the test bearings

1. Calculation diagram for the internal loads and predicted fatigue lives

The calculation diagram for the internal loads of the test bearings is shown in Figure 6.6. A standard 51306 rolling bearing has 12 balls. Nine balls were removed from the cage of the test bearing, and three balls were left in it. Therefore the load on one ball of the test bearing was four times that experienced by one ball of a standard bearing under the same loading conditions.
2. The calculated results

The calculated results are given in Table 6.2. \( s \) is the ratio between the ball number of a standard bearing and the ball number of the test bearing. \( Q \) is the weight on the test machine. \( N \) is the load on the test bearing. \( N_i \) is the load on one ball of the test bearing. The \( \sigma_{\text{max}} \) is the maximum normal stress in the contact zone between the ball and the raceway of the test bearing. \( P \) is an equivalent dynamic load on the test bearing. In Table 6.2, the values of the maximum normal stress \( \sigma_{\text{max}} \) were calculated according to Chapter 3, equation (3.44). The values of the predicted fatigue life of the \( L_{10} \) were calculated according to the equation, \( [L_{10} = (40300/P)^3] \), of the uncoated 51306 bearing. The predicted \( L_{10} \) fatigue lives of the uncoated 51306 bearing will provide the useful reference for the rolling contact fatigue test of the coated bearings.

![Figure 6.6 Calculation diagram of the internal loads](image_url)

Figure 6.6 Calculation diagram of the internal loads
Table 6.2 Calculated results of the internal loads and predicted fatigue lives

<table>
<thead>
<tr>
<th>Q, (N)</th>
<th>N (N)</th>
<th>Ni (N)</th>
<th>σ_max (N/mm²)</th>
<th>P (N)</th>
<th>Predicted Life L_10 = (40300/P)^3 x 10^6</th>
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<td>2898</td>
<td>38880</td>
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</tr>
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<td>9900</td>
<td>3300</td>
<td>2916</td>
<td>39600</td>
<td>1.05397</td>
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6.3 Experiments on rolling contact failure behaviour of coated bearings

6.3.1 Experiments on the rolling contact fatigue failure behaviour under oil lubricant condition

6.3.1.1 Experiment purposes

The experiment purposes are for the investigation of the fatigue failure performances of coated bearings under oil lubricated conditions, for further studies of the effect of factors on the fatigue-life of coated components, for the improved understand of the rolling contact fatigue mechanism, for the improvement of the contact fatigue resistance of coated components and for a new coating design. Also, the experiment purposes are for the improved understand of the accelerated rolling contact fatigue mechanism and for the improvements of the test method of the accelerated rolling contact fatigue of coated components.

6.3.1.2 Experiment procedure

The two groups of the 52306 coated bearings, and each group includes 6 test bearing raceways, were investigated for their fatigue performances under the oil lubrication condition. The nine balls were moved away from the cage of each test bearing, and the three balls were left in it. The test bearing parameters are shown in the table 6.1. The internal loads and predicted fatigue lives of the test bearings are shown in Table 6.2. The test parameters under oil lubricant condition are shown in Table 6.3. The details of the test machine used in this experiment are given in the Chapter 5. The type of the oil lubricant was the Shell Tonner 72 oil lubricant. The test weight $Q$ was increased from 0 to 2344 N step by step (in approximately 100 N increments) after the test machine was started each time. The quick-stop parameters of the friction torque and the acceleration of the control
system respectively are 2850 N.mm and 25m/s². When any one of these two parameters measured under the test is over its parameter limit, the control system quick-stopped the test, and then the test bearing was checked.

The three groups of the 52306 coated bearings (each group including 6 test bearing raceways) were investigated for their fatigue performances under an oil lubricant containing micro-particulates. The nine balls were removed from the cage of each test bearing, and the three balls were left in it. The test bearing parameters are shown in Table 6.1. The internal loads and the predicted fatigue lives of the test bearings are shown in Table 6.2. The test parameters under oil lubricant conditions are shown in Table 6.3. The type of the oil lubricant is again the Shell Tonner 72 oil, containing F120 Silicon Carbide Powder, (1L oil/5g Powder). The test weight Q was increased from 0 to 2344 N step by step (in approximately 100 N increments) after the test machine was started each time. First, the accelerated rolling contact fatigue test was operated under the oil lubrication condition containing added micro-particulates for 30 minutes; and then using the pure oil instead of the particulate-added oil, as shown in Figure 5.4. The test was operated continually until the initial fatigue cracks of the coated test bearing occur.
Table 6.3 Test parameters of the oil lubrication experiment

<table>
<thead>
<tr>
<th>Lubrication Type</th>
<th>Bearing Type</th>
<th>Balls Number</th>
<th>Bearing raceways Number</th>
<th>Load Q (N)</th>
<th>Maximum Normal Stress σmax (N/mm²)</th>
<th>Speed (r.p.m)</th>
<th>Torque Limit (N.mm)</th>
<th>Temp Limit (°C)</th>
<th>Vibration Limit (m/s²)</th>
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<tbody>
<tr>
<td>Pure Oil Shell Tonner Oil 72</td>
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<td>6</td>
<td>2344</td>
<td>3275</td>
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<tr>
<td>Shell Tonner Oil 72 with micro 120 Carbide Powder, 1L oil/5g P</td>
<td>Uncoated</td>
<td>3</td>
<td>6</td>
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<tr>
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6.3.1.3 Experiment results of the fatigue failures under oil lubrication condition

1. The experiment record curves

Figure 6.7 Experiment record curves of last 1000-500 seconds of CT1-O4 raceway

Figure 6.8 Experiment record curves of last 500 seconds-0 of CT1-O4 raceway
2. Fatigue performances of the test raceways

Typical photographs of failed test raceways are shown in Figure 6.9.

Figure 6.9 Fatigue failure appearances

The fatigue failure appearances of five groups of the test bearing raceways were much the same.
3. Fatigue life records curves

Figure 6.10 Fatigue life records curve of the test bearing raceways

Uncoated-Oil: means uncoated bearing raceway with clean oil

Uncoated-Grains: means uncoated bearing raceway with oil added macro particulates

CT1-Oil: means CT1 coated bearing raceway with clean oil

CT1-Grains: means CT1 coated bearing raceway with oil added macro particulates

CT4-Grains: means CT4 coated bearing raceway with oil added macro particulates
4. Fatigue life distribution curves

Figure 6.11 Fatigue life distribution curves of the test bearings
5. Calculation of the fatigue life of the test coated bearings (Oil)

The calculation of the fatigue life of the test coated bearings under oil lubricant condition is shown in Table 6.4, the L_{50}-fatigue life curves are shown in Figure 6.10.

Table 6.4 Weibull analysis results of the fatigue lives (Oil)

<table>
<thead>
<tr>
<th>Fatigue Life</th>
<th>Normal Test (x10^6 cycles)</th>
<th>Accelerated test (x10^6 cycles)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>UNC</td>
<td>CT1</td>
</tr>
<tr>
<td>Samples</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Slope:</td>
<td>1.134</td>
<td>1.615</td>
</tr>
<tr>
<td>L10:</td>
<td>2.548</td>
<td>2.657</td>
</tr>
<tr>
<td>L50:</td>
<td>13.424</td>
<td>8.530</td>
</tr>
<tr>
<td>UCL65:</td>
<td>20.671</td>
<td>11.549</td>
</tr>
<tr>
<td>LCL65:</td>
<td>8.096</td>
<td>5.981</td>
</tr>
</tbody>
</table>

Figure 6.12 L_{50}-fatigue life curves under oil lubricant condition
6. Proportion coefficient $\delta^*$ of the accelerated fatigue test

The proportion coefficients $\delta^*$ of the normal fatigue test lives $L_{50}$ to the accelerated fatigue test lives $L_{50}$ of the test bearing raceways are shown in Table 6.5.

Table 6.5 Fatigue lives $L_{50}$ of the test bearing raceways

<table>
<thead>
<tr>
<th>Samples</th>
<th>Experiment</th>
<th>$L_{50}$</th>
<th>Conditions Compared</th>
<th>Proportion $\delta^*$</th>
</tr>
</thead>
<tbody>
<tr>
<td>U-N</td>
<td>Normal</td>
<td>13.424</td>
<td>U-N/ U-N</td>
<td>$\delta^*_{U-N}$</td>
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<tr>
<td>CT1-N</td>
<td></td>
<td>8.530</td>
<td>U-N/ CT1-N</td>
<td>$\delta^*_{CT1-N}$</td>
</tr>
<tr>
<td>U-A</td>
<td>Accelerated Test</td>
<td>5.531</td>
<td>U-N/ U-A</td>
<td>$\delta^*_{U-A}$</td>
</tr>
<tr>
<td>CT1-A</td>
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<td>2.386</td>
<td>CT1-N/ CT1-A</td>
<td>$\delta^*_{CT1-A}$</td>
</tr>
<tr>
<td>CT4-A</td>
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<td>1.246</td>
<td>CT1-N/ CT4-A</td>
<td>$\delta^*_{CT4-A}$</td>
</tr>
</tbody>
</table>

$\delta^*$ is a proportion coefficient of the normal fatigue test life to the accelerated fatigue test life. U-N and CT1-N means respectively uncoated bearing raceway and CT1 coated bearing raceway in normal test. U-A, CT1-A and CT4-A means respectively uncoated bearing raceway, CT1 and CT4 coated bearing raceways in accelerated test.

6.3.1.4 Observations under the scanning electron and optical microscopes

The following were observed using microscopy:

1. Uncoated test bearing raceways

   a. Fatigue cracks spreading on the and contact surface (plan view)
b. The position of the initial fatigue cracks of the uncoated bearing raceways (Cross-sectional view)

(After $10^6$ cycles)

Figure 6.13 Optical view of fatigue cracks spreading on the uncoated contact surface

(After $2 \times 10^6$ cycles)

Figure 6.14 Optical view of fatigue cracks spreading on the uncoated contact surface
Figure 6.15 Optical view of the position of the initial fatigue cracks

2. Coated test bearing raceways (Pure oil)
   
a. Fatigue crack spreading on the coated contact surface
   
   b. The position of the initial fatigue crack of the coated bearing raceways

Figure 6.16 SEM view of fatigue cracks spreading on the contact surface

112
Figure 6.17 SEM view of fatigue cracks spreading on the contact surface

Figure 6.18 Optical view of fatigue cracks spreading on the coated contact surface
Figure 6.19 Optical view of fatigue cracks spreading on the coated contact surface

Figure 6.20 SEM view of the position of the initial fatigue cracks
Figure 6.21 Optical view of the position of the initial fatigue cracks of a typical coated bearing raceway

Figure 6.22 Optical view of the position of the initial fatigue crack of a typical coated bearing raceway
3. Coated test bearing raceways (Contaminated Pure oil)

a. Fatigue crack spreading on the contact surface

b. The position of the initial fatigue crack of the coated bearing raceways

Figure 6.23 SEM view of fatigue cracks spreading on the contact surface

Figure 6.24 SEM view of fatigue cracks spreading on the contact surface
Figure 6.25 Optical view of fatigue cracks spreading on the coated contact surface

Figure 6.26 Optical view of fatigue cracks spreading on the coated contact surface
Figure 6.27 Cross-sectional SEM view of the position of the initial fatigue cracks

Figure 6.28 Cross-sectional optical view of the position of the initial fatigue crack
6.3.1.5 Discussion of the test results of the rolling contact fatigue failures under oil lubricant conditions

1. Failure performances during rolling contact fatigue of the test bearing raceways
   a. Surface failure performances of the test bearing raceways
      As seen in Figure 6.9, the fatigue failure performances of the test bearing raceways of the five groups are very similar. As seen in Figure 6.13 to Figure 6.28, firstly, a small fatigue crack occurs near the surface of the raceways. Then, the intermeshing fatigue cracks are developed on a small zone about 2-3mm² in area. Finally, a large piece falls off the surface of the test bearing raceway. A concave hole appears on the surface.

b. Position of the initial fatigue crack of the test bearing raceways
   As shown in Figure 6.15, Figure 6.20, Figure 6.21, Figure 6.22, Figure 6.27 and Figure 6.28, an initial fatigue crack raceway occurs near the substrate subsurface. Then, the initial fatigue crack spreads to the surface of the test bearing raceway, whilst intermeshing cracks appear on the surface. Finally, a large piece falls off the surface.

2. Analysis of the failure mechanisms during rolling contact fatigue of the test bearing raceways
   The failure mechanisms in rolling contact fatigue of the test bearing raceways are influenced by many contributing factors; for example, the substrate properties, the coating properties, the load condition, operation environment, etc. However, the distribution of the maximum shear stress is usually considered to be the main factor dictating the failure mechanisms in the rolling contact fatigue of the test bearing raceways. The cyclic maximum shear stress produces an initial fatigue crack near the substrate surface of the test bearing raceways.
The observed phenomena are consistent with theory, although the location of the initial crack is much closer to the surface than would be predicted by a 'static' Hertzian analysis. The influences of dynamic surface friction force and surface intermeshing fatigue cracks will result in the variation of the position of the failure initiation so that it approaches the surface of test bearing raceways.

3. Comparison of the rolling contact fatigue lives of the test bearing raceways

The rolling contact fatigue records and fatigue curves of the five groups of the test bearings are shown in Figure 6.11 and Figure 6.12. Weibull analysis results of the fatigue life are shown in Table 6.4. The \(L_{50}\) fatigue lives of the five groups of the test bearings are shown in Figure 6.12. The \(L_{50}\) fatigue life of the uncoated bearing raceways is longer than that of the coated raceways under the normal fatigue test, and also under the accelerated fatigue test. The reasons causing the reduction of the fatigue life of a coated bearing raceway could be as follows:

a. Effect of the fatigue resistance of the substrate subsurface

Although super-hard coating can improve the surface properties of the test bearing raceways, the fatigue resistance of the substrate subsurface of the coated bearing raceway could not be improved, even it could be decreased. As shown in Figure 4.1 (Chapter 4), the maximum shear stress occurs just near the substrate subsurface of the coated bearing raceway. Therefore, the fatigue resistance of the substrate surface of a coated bearing raceway plays a very important role in the improvement of rolling contact fatigue resistance of the coated bearing.

b. Effect of the traction force being insufficient
When the traction force at the surfaces between the rolling elements is insufficient to drive the rolling process, gross skidding occurs, which leads to rapid wear, over-heating and final failure. Bearing raceways with coatings CT1 ~ CT4 have a very low dynamic friction coefficient against steel. The lubricant-coating interaction functions reduce the traction forces of the coated bearings. When micro-sliding starts to occur, the von Mises stresses increase and fatigue life decreases. This could be one of the reasons that the fatigue life of the coated bearing raceways decreases versus the uncoated bearing raceways.

4. Accelerated fatigue test

As shown in Figure 6.9 and Figure 6.23 ~ Figure 6.28 the fatigue failure performances of the test bearing raceways are much the same between the normal fatigue test and the accelerated fatigue test. It demonstrates that the accelerated fatigue test method can obtain the same fatigue test results as the normal fatigue test method. As seen in Table 6.5, $\delta^*$ is a proportion coefficient of normal fatigue test life to the accelerated fatigue test life. Comparing L50 fatigue lives of uncoated bearing raceways between the normal and the accelerated fatigue test, the $\delta^*_{U-A}$ is equal to 2.427 and to the CT1 coated bearing raceways, the $\delta^*_{CT1-A}$ is equal to 3.575. It appears the accelerated fatigue test method is very efficient to save the time consumed in the fatigue test of the coated components.

5. Functions of the detection and diagnosis of the test machine

Two main kinds of the fatigue failure performances appear on the surface of bearing raceways in the fatigue experiments. One is that a large fatigue piece peels off the surface of a test raceway and forms a concave impression on the surface of the raceway. Another is that intermeshing cracks occur on the surface of a raceway. The intermeshing fatigue cracks usually occur before the big fatigue piece falls off.
The detection and diagnosis of the initial fatigue crack by vibration is possible only by peeling of a big fatigue piece, and it is difficult to find the micro intermeshing fatigue cracks. However, the method to detect and diagnose the initial fatigue crack by the friction force variation can precisely find the initial intermeshing fatigue cracks which occur on the surface of the test bearing raceway. Combining with the quick-stop device, the rolling contact fatigue test machine can accurately detect and diagnose the initial fatigue cracks and quickly stop the operation of the test machine to preserve the initial state of the initial fatigue cracks.

6.3.1.6 Experimental conclusions

1. Theory suggests that the distribution of the maximum shear stress is the main factor influencing the failure mechanisms of rolling contact fatigue of the test bearing raceways. The maximum shear stress produces an initial fatigue crack below the surface of the test bearing raceways. In these experiments, the location of the initial fatigue crack was not as deep as ‘static’ model predicts. The observed phenomena are consistent with theory, although the location of the initial crack is much closer to the surface than would be predicted by a ‘static’ Hertzian analysis. The influences of dynamic surface friction force and surface intermeshing fatigue cracks will result in the variation of the position of the failure initiation so that it approaches the surface of test bearing raceways.

2. The fatigue resistance of the substrate subsurface of the coated bearing raceway plays a very important role in the improvement of rolling contact fatigue resistance of the coated bearing.
3. Insufficient tractional forces on the contact surface between the rolling elements of a test coated bearing causes gross skidding of the rolling element to occur, which leads to rapid wear, over-heating and final failure of the test coated bearing.

4. The accelerated rolling contact fatigue test can be used to obtain the same fatigue test result ranking as with the normal rolling contact fatigue tests. The accelerated rolling contact fatigue test is very efficient to save the time consumed in rolling contact fatigue testing of the coated components.

5. The fatigue test machine can accurately detect and diagnose the initial fatigue crack and quick-stop the operation of the test machine to preserve the initial state of the initial fatigue crack for the coated components.

6.3.2 Experiments on the rolling contact fatigue failure behaviour under grease lubricant conditions

6.3.2.1 Experiment purposes

The experiment purposes of this part of the work were to investigate the fatigue failure performance of coated bearings under the grease lubricant condition, and to carry out further studies of the effects of the variation of the lubrication situation, of the surface friction force, of the coating thickness, and of the coating hardness on the fatigue-life of coated components. Also, the experiment purposes are to achieve improved understanding of the rolling contact fatigue mechanism and to find ways to improve the contact fatigue resistance of coated components through new coating designs.

6.3.2.2 Experiment procedure

The four groups of the 52306 coated bearings (each group including 6 test bearing raceways), were investigated for their fatigue performance under grease lubrication
conditions. Again, nine balls were removed from the cage of each test bearing, and the three balls were left in it. The test bearing parameters are shown in the table 6.1. The internal loads and the predicted fatigue lives of the test bearing are shown in Table 6.2. The test parameters are shown in Table 6.6. The detail of the test machine used in this experiment is as shown in the Chapter 5. The type of the grease lubricant was the ALVANIA R3 grease. The operating speed of the test bearing was the 2050 r.p.m. The test weight $Q$ was increased from 0 to 1980 N step by step (in approximately 100 N increment) after the test machine was started each time. The quick-stop parameters of the friction torque and the acceleration of the control system respectively are 2850 N.mm and 25m$^2$/s. When any one of these two parameters measured under the test is over its quick-stop parameter limit, the control system quick-stopped the test, and then the test bearing was checked. Usually, the lack of the grease lubricant, due to the operational consumption, will cause an increase of the test temperature of the test bearing. The highest temperature limit of the test bearing is controlled at 140 °C. When the temperature of the test bearing is over 140 °C, the control system automatically stops the test, and then the test bearing is checked and the grease is added again.

Table 6.6 Test parameters of the grease lubrication experiment

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<thead>
<tr>
<th>Bearing Type</th>
<th>Bearing raceways Number</th>
<th>Balls Number</th>
<th>Grease Type</th>
<th>Load $Q$ (N)</th>
<th>Speed (r.p.m)</th>
<th>Torque Limit (N.mm)</th>
<th>Temperature Limit (°C)</th>
<th>Vibration Limit (m$^2$/s)</th>
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<td>1980</td>
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<td>2850</td>
<td>140</td>
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<td></td>
</tr>
</tbody>
</table>

6.3.2.3 Experimental results of the rolling contact fatigue failures under grease lubrication condition
1. The experiment record curves

The experiment record curves are shown in Figure 6.29.

![Figure 6.29 Experimental record curves](image)

a) Experiment record curves of last 1000-500 seconds of CT1-G2 raceway

![Figure 6.29 Experimental record curves](image)

b) Experimental record curves of last 500-0 seconds of CT1-G2 raceway
2. Fatigue life record

Figure 6.30 Fatigue life records curves of the test bearing raceways
3. Fatigue life distributions

Figure 6.31 Fatigue life distribution curves of the test bearing raceways
c. Calculation of the fatigue life of the test coated bearings (Grease)

The calculation of the fatigue life of the test bearings under grease lubricant condition is shown in Table 6.7, and the test curves of $L_{50}$-fatigue life are shown in Figure 6.8.

Table 6.7 Weibull analysis results of the fatigue lives (grease); all x10E6 cycles

<table>
<thead>
<tr>
<th>SAMPLE</th>
<th>L10</th>
<th>L50</th>
<th>65% UCL</th>
<th>65% LCL</th>
<th>SLOPE</th>
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</thead>
<tbody>
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<td>BASE</td>
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<td>1.713</td>
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<td>1.888</td>
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<td>1.733</td>
<td>2.089</td>
<td>1.393</td>
<td>2.624</td>
</tr>
<tr>
<td>CT3</td>
<td>0.967</td>
<td>1.333</td>
<td>1.449</td>
<td>1.209</td>
<td>5.872</td>
</tr>
</tbody>
</table>

Figure 6.32 $L_{50}$-fatigue life test curves under grease lubricant condition
4. Correlation between the L50 fatigue life and the coating thickness

Figure 6.33 Test curves of the L50 life and the coating thickness (Grease)

Figure 6.34 Correlation curves between the L50 life and the coating thickness
5. Correlation between the L50 life and the coating hardness

![Graph showing correlation between L50 life and coating hardness](image)

Figure 6.35 Test curves of the L50 life and the coating hardness (Grease)

6. Optical observations of fatigue cracks spreading on the surfaces

a. Uncoated bearing raceway

![Micrograph of uncoated raceway](image)

Figure 6.36 Shows of fatigue cracks of uncoated raceway (a)
Figure 6.37 Shows of fatigue cracks of uncoated raceway (b)

Figure 6.38 Shows of fatigue cracks of uncoated raceway (c)

Figure 6.39 Shows of fatigue cracks of uncoated raceway (d)
b. Coated bearing raceway

Figure 6.40 Shows of fatigue cracks of coated raceway (a)

Figure 6.41 Shows of fatigue cracks of coated raceway (b)
Figure 6.42 Shows of fatigue cracks of coated raceway (c)

Figure 6.43 Shows of fatigue cracks of coated raceway (d)
7. Optical observations of fatigue cracks of the coated raceways in polished cross-section

Figure 6.44 Appearance of the raceway failure in cross-section

CT1-Grease (a)

10 μm

CT1-Grease (b)

10 μm
6.3.2.4 Discussion of the fatigue test results under grease lubricant conditions

1. Failure performances of the rolling contact fatigue of the test bearing raceways

By the optical observation of fatigue cracks spreading on the surfaces, seen in Figure 6.36 ~ Figure 43, the fatigue failure performance of the uncoated and coated bearing raceways looks the same. The roughness of the fatigue failure surface of the coated bearing raceways is smoother and the cracks are more regular than that of the uncoated bearing raceways. As seen in Figure 6.44, the positions of the fatigue cracks are at the interface or near surface of the coated bearing raceway. Possibly, the initial fatigue cracks occur near the interface where the maximum shear stress occurs here. Therefore, the interfacial adhesion between the coating and substrate strongly influences the fatigue resistance of a coated component.

2. Comparison of the rolling contact fatigue lives of the test bearing raceways

The rolling contact fatigue records and fatigue curves of the four groups of the test bearings are shown in Figure 6.29 and Figure 6.30. Weibull analyses of the fatigue life are shown in Table 6.7. The L50 fatigue lives of the four groups of test bearings are shown in Figure 6.32. The L50 fatigue life of the uncoated bearing raceways is higher than that of the coated raceways under the normal fatigue test. The reasons causing the reduction of the fatigue life of a coated bearing raceway could be similar to the test under oil lubricant conditions.

3. Effect of the coating thickness on the L50 fatigue life

As seen in Figure 6.33 and Figure 6.34, the L50 fatigue life of a test coated bearing raceway is in inverse proportion to its coating thickness. As the coating thickness of the test
coated bearing raceway increases, the L50 fatigue life of the test coated bearing raceway tends to decrease.

4. Effect of the coating hardness on the L50 fatigue life

As seen in Figure 6.35, the L50 fatigue life of the test coated bearing raceway is in inverse proportion to its coating hardness. As the coating hardness of the test coated bearing raceway increase, the L50 fatigue life of the test coated bearing raceway tends to decrease.

5. Some coatings wear away

Some coatings wear away from the test bearing raceways under the optical observation. Sliding must exist in the contact zone between the ball and the coated bearing raceway. The sliding results in an increase of the temperature of the raceway. The increase in temperature leads to graphitization in the carbon-based coating, and formation of boric acid in the boron carbide based coatings. When a little coating is left in the raceway, the friction coefficient in the contact zone increases, and the temperature further increasing, and then the fatigue life $L_{10}$ decreases. Although it is not entirely clear when and how the coating of the test bearing raceway is worn away, it is evidence that wear does occur, and this wear is commensurate with the increase in sliding of the balls, compared to an uncoated bearing. It may also be the case that ball lateral rotation takes place (i.e. a gyroscopic effect) further increasing wear.

6.3.2.5 Experimental conclusions

1. The maximum shear stress is usually regarded as the main effect factor on the fatigue resistance of the coated bearing raceway. The fatigue resistance of the substrate
subsurface of a coated bearing raceway will decide the fatigue life of the coated bearing raceway.

2. The lubrication condition plays a very important role in the rolling contact fatigue test of coated bearings. The rolling fatigue life of a coated bearing is lower under the grease lubrication condition than that of under oil lubrication condition.

3. The L50 fatigue life of the test coated bearing raceway decreases with the increase of the coating thickness, when the coating thickness of the test coated bearing raceway is in the range of 1~3 microns.

4. The L50 fatigue life of the test coated bearing raceway decreases as the coating hardness of the test coated bearing raceway increases, when the coating thickness of the test coated bearing raceway is in the range of 7-25 GPa.
6.3.3 Experiment of the coated bearing failures under no lubricant condition

6.3.3.1 Experiment background

Most bearing failures are more likely to occur due to inadequate lubrication or interruption of lubricant delivery than from any other causes. Some terrible disasters of aircrafts occur due to the failure of the main bearings of the aircrafts (Berkovits, A., 1995; Tauqir, A., 2000). For example, a fighter plane crashed in 1996. The on site investigations pointed out that the terrible disaster was caused due to the damage of a main bearing of the aero-engine of the fighter plane (Salam, I., 1998).

The capability for bearings to respond to oil interruption under the full-load operation has been demonstrated to be a vital reliability performance to the vehicles of the aviation and navigate. Aircraft gas turbine engine mainshaft bearings, for example, are typically required to survive for 30 s after oil shut-off at the full power takeoff condition (Obara, S. and Suzuki, M., 1997; Hadfield, M., 1998). A relatively stable and low friction coefficient is an intrinsic advantage maintaining the capability of failure resistance of bearings during the lubricant interruption. Hard and superhard coatings have been successfully used to improve the surface material (surface engineered materials) tribological functions.

For the investigation of the failure performances of coated bearings under no lubricant condition, the experiment of failure performances no lubricant were carried out using the new rolling contact fatigue test machine. 51306 coated bearings were investigated in the experiment. The mechanism model of the coated bearing failure under no lubricant is presented, and the test results are discussed.
6.3.3.2 Experiment procedure

The coated bearing raceways were investigated for their fatigue performance under no lubrication condition. Also, nine balls were removed from the cage of each test bearing, and the three balls were left in it. The parameters of the bearings tested are as shown in the Table 6.1. The internal loads and the predicted fatigue lives of the bearings tested are shown in Table 6.2. The test parameters are shown in Table 6.8. The operating speed of the test bearing was 2050 r.p.m. The packing oil of the test coated bearings was cleaned off before the test. The test weight $Q$ was increased from 0 to 200 N step by step (in approximately 20 N increments) after the test machine was started each time. The quick-stop parameters of the friction torque and the acceleration of the control system respectively are 4000 N.mm and 25$m^2$/s. When any one of these two parameters measured under the test is over its quick-stop parameter limit, the control system quick-stopped the test, and then the test bearing was checked.

Table 6.8 Test coated bearing and test parameters

<table>
<thead>
<tr>
<th>Coated Bearing</th>
<th>Test Bearings Amount</th>
<th>Load $Q$ (N)</th>
<th>Lubricant</th>
<th>Speed (r.p.m)</th>
<th>Torque Limit (N.mm)</th>
<th>Balls</th>
<th>Average Life-Time (Min)</th>
<th>Average Life-Revolutions $10^6$</th>
</tr>
</thead>
<tbody>
<tr>
<td>CT1</td>
<td>5</td>
<td>200</td>
<td>No</td>
<td>2050</td>
<td>4000</td>
<td>3</td>
<td>97</td>
<td>0.193941</td>
</tr>
</tbody>
</table>

6.3.3.3 Experiment results

The five 51306 coated bearings were operated under no-lubricant condition. The damage performances of the five test bearings were almost the same as each other.
1. Experiment record curves

The test record curves are shown in Figure 6.45.

![Test record curves of coated bearings (no lubricant)](image)

Figure 6.45 Test record curves of coated bearings (no lubricant)

2. Damage performance of the test coated bearings under no-lubricant condition

After checking the tested bearings, the main failure observed was damage of the bearing cage. The upper surface of the damaged cage was bulged by the balls of the bearing and worn by the edges of upper-raceway. The damaged cages experienced severe deformation and their failure performances are shown in Figure 6.46. The worn marks of the raceways are shown in Figure 6.47 and Figure 6.48.
Figure 6.46 Failure performance of the damaged cage under no-lubricant condition
Figure 6.47 Appearance of upper-raceway wear

Figure 6.48 Appearance of lower-raceway wear
6.3.3.4 Analysis and discussion of the test results

1. Failure mechanism model of the damaged cage under no lubricant condition

The failure mechanism model of the damaged cage under no lubricant condition is shown in Figure 6.49. The load \( N \) acts on the upper-raceway of a test bearing and whilst the test bearing turns. The middle circle of the test bearing cage turns at the circle speed \( V_c \). The centre speeds of the ball 1 and ball 2 are individually \( V_1 \) and \( V_2 \). The \( F_{nl} \) and \( F_{n2} \) are individually the normal forces of the contact points between the balls and the cage. \( F_{f1} \) and \( F_{f2} \) are individually the friction forces of the contact points between the balls and the cage. \( F_{ol1} \) and \( F_{ol2} \) are individually the sum forces of the normal force and friction of the contact points between the balls and the cage. \( F_{tn1} \) and \( F_{tn2} \) are individually the perpendicular component forces of the contact points between the balls and the cage. \( F_{tl1} \) and \( F_{tl2} \) are individually the level component forces of the contact points between the balls and the cage. \( \Theta_1 \) is the angle between the sum forces \( F_{ol1} \) and the level component forces \( F_{tl1} \). \( \Theta_2 \) is the friction angle between the sum forces \( F_{ol1} \) and the normal forces \( F_{nl1} \).

2. Analysis of the failure mechanism of the test bearing cage

Since the error from the manufacture of the bearings exists, \( V_1 \) and \( V_2 \) are theoretically not equal each other, and then \( F_{tl1} \) and \( F_{tl2} \) are produced by restriction. Static equilibrium requires that

\[
F_{tl1} + F_{tl2} = 0 \quad (6.1)
\]

Also, the values of the level component forces, \( F_{tl1} \) and \( F_{tl2} \), are related to the friction coefficients between the ball and the cage, and to the load \( N \) of the test bearing and the rotation speed \( n \) of the test bearing. As seen in Figure 6.49, the perpendicular component forces \( F_{tn1} \) is:

\[
F_{tn1} = F_{tl1} \tan (\Theta_1+\Theta_2) \quad (6.2)
\]
Figure 6.49 Failure mechanism model of the damaged cage
When the friction coefficient tends to zero, $\theta_2$ tends to zero, $F_{nt1}$ tends to $[F_{t1}\tan(\theta_1)]$, and $F_{nt1}$ reached a minimum value. As the friction coefficient tends to an increase, the friction force tends to an increase, $\theta_2$ tends $(90^\circ-\theta_1)$, and then $F_{nt1}$ tends to $\infty$ resulting in an adhesive wear between the balls and the cage until the cage is broken.

6.3.3.5 Experiment conclusions

According to the analysis of the test results of the coated bearings under no lubrication condition, the conclusions obtained are as follows:

1. The main failure mode observed for the coated bearings under no lubrication condition is cage damage.

2. The main reason for the damage under no lubrication condition is that the perpendicular component force $F_{nt1}$ acting on the cage increases continually with the increase of the friction force between the balls and the cage until the cage is broken.

6.4 Summaries of the experiment results of the test coated bearings

6.4.1 Rolling contact failure performances of the test coated bearings

1. Under the oil lubricant and grease condition.

Initial fatigue cracks occur near the substrate subsurface of the test coated bearing raceways under the action of the cyclic maximum shear stress. The initial fatigue crack extends to the surface of a test coated bearing raceway, whilst intermeshing cracks appear on the surface of the test coated bearing raceway. First, a small fatigue crack occurs on the surface of the raceways. Then, the intermeshing fatigue cracks are developed in a small zone. Finally, a piece falls off the surface of the test bearing raceway and a concave hole appears on the surface. Excessive cyclic maximum shear stress is one of the main reasons
for the initial crack to occur. According to the observations with SEM and optical microscopy, the coating appears to be present after the fatigue tests. Moreover it was not possible to find any edges in the coating between the contact and non-contact zones.

2. Under no lubricant (dry lubrication) conditions

The damage to the test coated bearing cages is the main failure affecting performance under the no lubricant condition. The damage to the test coated bearing cages occurs before the rolling contact fatigue failures of the test coated bearing raceways. The friction force between the cage and balls will increase continually until the cage has been damaged under no lubricant (Dry lubrication) conditions. The variation of friction force acting on the cage is related to the precision, load and speed of operation of the test coated bearings.

6.4.2 Factors effecting the rolling contact fatigue resistance of the coated test bearings

1. The distribution and maximum value of the maximum shear stress are the main factors affecting the rolling contact fatigue resistance of the test bearing raceways.

2. The fatigue resistance of the substrate subsurface of the coated bearing raceway plays a very important role in the improvement of rolling contact fatigue resistance of the coated components.

3. Insufficient traction forces on the contact surface between the rolling elements of a test coated bearing will cause gross skidding to occur, which leads to rapid wear, overheating and final failure of the test coated bearing.

4. The L50 fatigue life of the test coated bearing raceway decreases with increasing coating thickness, when the coating thickness is in the range of 1~3 microns.
5. The L50 fatigue life of the test coated bearing raceway decreases with increasing coating hardness of the test coated bearing raceway, when the coating hardness is in the range of 7~25 GPa.

6. The accelerated rolling contact fatigue test can provide equivalent fatigue test result rankings as with the normal rolling contact fatigue test. The accelerated rolling contact fatigue test is therefore a very efficient way to save the time consumed in the normal rolling contact fatigue test of the coated components.

7. The fatigue test machine can accurately detect and diagnose the initial fatigue crack and quick-stop the operation of the test machine to preserve the state of the initial fatigue crack for the coated components of superhard coating materials.

8. Some coatings wear away from the test bearing raceways (under the optical observation). Sliding must exist in the contact zone between the ball and the coated bearing raceway. The sliding results in an increase of the temperature of the raceway, and then the fatigue life $L_{10}$ decreases. Although it is not entirely clear when and how the coating of the test bearing raceway is worn away, it is evidence that wear does occur, and this wear is commensurate with the increase in sliding of the balls, compared to an uncoated bearing. It may also be the case that ball lateral rotation takes place (i.e. a gyroscopic effect) further increasing wear.
CHAPTER 7 Conclusions and recommendations

7.1 Conclusions

7.1.1 Rolling contact fatigue test machine

1. The rolling contact fatigue test machine which was designed and manufactured on the basis of the rolling contact fatigue mechanism and its model is suitable to evaluate the rolling contact fatigue resistance of coated components.

2. The measurement system of the rolling contact fatigue test machine can precisely detect and diagnose the initial cracks by combining the measurement of vibration with the measurement of friction torque in the contact zone of the test component geometry.

3. The control and transmission systems of the rolling contact fatigue test machine can quickly stop the operation of the rolling contact fatigue test machine to preserve the state of the initial fatigue cracks when such cracks occur, by applying the quick-stop device of the test machine.

7.1.2 Accelerated fatigue test methodology

1. The accelerated rolling contact fatigue test under the contaminated oil lubricant conditions can obtain equivalent fatigue test results as found with the normal fatigue test under the clean oil lubricant condition. The time required in the fatigue resistance test of coated components can be reduced substantially with the accelerated fatigue test method. The accelerated fatigue test method is thus a very efficient way to reduce the time required to compare the fatigue test life of many different coated components.

2. The mechanical transmission and lubrication systems of the rolling contact fatigue test machine can simulate the actual application situation of coated components. The
proportion coefficient $\delta^*$ of the normal fatigue life to the accelerated fatigue life can be obtained by using respectively the normal fatigue test method and the accelerated fatigue method. Therefore, the accelerated fatigue test can simulate the actual applications of the coated components, and the test results possess a good reliability.

7.1.3 Evaluation experiments of the fatigue resistance of the coated bearings

7.1.3.1 Rolling contact failure performance of the test coated bearings

1. Under the oil and grease lubricant condition.

First, the initial fatigue crack occurs near the substrate subsurface of the test coated bearing raceway under the action of the cyclic maximum shear stress. Then, the initial fatigue crack extends to the surface of the test coated bearing raceway, whilst a small fatigue crack occurs on the surface of the raceways. Then, the intermeshing cracks appear on the surface of the test coated bearing raceway. Further, the intermeshing fatigue cracks are developed in a small zone. Finally, a piece falls off the surface of the test bearing raceway and a concave hole appears on the surface. Excessive maximum cyclic shear stress is one of the main reasons causing the initial crack to occur.

2. Under no lubricant (Dry lubrication) conditions

The damage to the test coated bearing cages is the main failure affecting performance under the no lubricant condition. The damage to the test coated bearing cages occurs before the rolling contact fatigue failure of the test coated bearing raceways. The friction force between the cage and balls will increase continually until the cage has been damaged under no lubricant (Dry lubrication) condition. The variation of friction force acting on the cage is related to the precision, load and speed of operation of the test coated bearings.
7.1.3.2 Factors effecting rolling contact fatigue resistance of the coated test bearings

1. The distribution and maximum value of the cyclic shear stress are the main factors affecting the rolling contact fatigue resistance of the test bearing raceways.

2. The fatigue resistance of the substrate subsurface of the coated bearing raceway plays a very important role in the improvement of rolling contact fatigue resistance of the coated components. However, it should be noted that account has to be taken of the change in operating parameters, when comparing these tests with real in-service conditions.

3. Insufficient traction forces in the contact between the rolling element surfaces of a test coated bearing cause gross skidding to occur, which leads to rapid wear, overheating and final failure of the test coated bearing.

4. The L50 fatigue life of the test coated bearing raceway decreases with increasing coating thickness, when the coating thickness is in the range of 1~3 microns.

5. The L50 fatigue life of the test coated bearing raceway decreases as increasing coating hardness of the test coated bearing raceway as the coating hardness is in the range of 7~25 GPa.

6. Some coatings wear away from the test bearing raceways (under the optical observation). Sliding must exist in the contact zone between the ball and the coated bearing raceway. The sliding results in an increase of the temperature of the raceway, and then the fatigue life L10 decreases. Although it is not entirely clear when and how the coating of the test bearing raceway is worn away, it is evidence that wear does occur, and this wear is commensurate with the increase in sliding of the balls,
compared to an uncoated bearing. It may also be the case that ball lateral rotation takes place (i.e. a gyroscopic effect) further increasing wear.

7.2 Recommendations

1. The rolling contact fatigue test results demonstrate that the $L_{50}$ fatigue lives of the coated bearing raceways are lower than those of the uncoated bearing raceways. Further understanding of the effect factors on fatigue resistance of coated components is crucial to improve their fatigue resistance. These factors include the friction force on the contact surface, residual stresses at the interface, chemical compositions of the coating and substrate, coating material microstructures and coating methods.

2. Studies of the relationships between the fatigue resistance and the intrinsic properties of the coated components will help to understand the fatigue failure mechanisms of coated components. The rolling contact fatigue behaviours should depend on the intrinsic performance parameters of the coated components. Therefore the fatigue resistance of coated components should be related to the properties, i.e. Young's modulus, residual stress, hardness and fracture resistance of coated components; material composition, microstructure of coating material, topography, coating/substrate interfacial adhesion and coating methods.

3. The tribological behaviour of coated components under the lubrication condition of oil with added particulates should be further studied. It has been discovered that the fatigue resistance of coated bearings will be obviously affected by the size (and size distribution) of the particulates added under the accelerated fatigue test of coated bearings. When the particulate size of Silicon Carbide Powder is less than 1 micron, particulates contained in oil lubricant provide a net benefit to the fatigue resistance of the coated bearings. It appears that the particulates would take up a polishing role in the
rolling contact of the rolling balls particularly during ‘running-in’. However, when the particulates’ size in oil lubricant is more than 3 microns, there is an obvious decrease the rolling contact fatigue resistance of the test coated bearings. This performance demonstrates that the variety of the size of particulates will cause the changes of the rolling contact tribological behaviours of coated components. According to the mechanism, it is possible to seek a special lubricant for the benefits of the fatigue resistance of rolling contact coated components.
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APPENDIX 1 Weibull distribution

The two-parameter Weibull distribution is designed to describe bearing fatigue life based on an empirical fit to bearing fatigue life data.

1. Probability Function

Weibull probability function, known as the cumulative distribution function (CDF), takes two parameters, namely, the scale parameter $\eta$ and the shape parameter $\beta$.

\[ F(x) = \Pr(\text{life} < x) = 1 - e^{-(x/\eta)^\beta} \quad x, \eta, \beta > 0 \]

where $x$ is an arbitrary value representing the bearing life.

The probability density function (pdf) then has the form,

\[ f(x) = \frac{x^{\beta-1}}{\eta^\beta} e^{-(x/\eta)^\beta} \]

2. Mean Time Between Failure

The average or expected value of a random variable is defined as

\[ E(x) = \int_{0}^{\infty} x f(x) dx = \eta \Gamma\left(\frac{1}{\beta} + 1\right) \]

In reliability theory $E(x)$ is known as the mean time between failures and is commonly referred to as MTBF. It represents the average time between the failures of two consecutively run bearings.

3. Percentiles

CDF gives the probability that the observed value of a Weibull random variable is less than an arbitrary value. The inverse problem is to find the value of $x$, say $x_p$, for which the probability is a specified value $p$ such that the life will not exceed it. The term $x_p$ is defined implicitly as

\[ F(x_p) = 1 - e^{-(x_p/\eta)^\beta} = p \]

The solution is

\[ x_p = \eta \left[ \ln\left( \frac{1}{1 - p} \right) \right]^{1/\beta} \]

4. Explanation of terms

L10 represents the $x_{0.10}$;

L50 represents the $x_{0.50}$;

SLOPE represents the shape parameter $\beta$;

UCL represents the upper control limits;

LCL represents the lower control limits;

65%UCL and 65%LCL represent the value which the mean time will not exceed.