A SLIDING WEAR MODEL AND ITS APPLICATION TO HEAT EXCHANGER TUBE WEAR

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Abstract

The objective of this study is to quantitatively determine wear and its main parameter relationships for heat exchanger tube wear. A model has been proposed.

The literature of the current state of heat exchanger tube wear study, wear mechanisms, and wear models were reviewed.

Tube/disc sliding wear tests were conducted with an impact-fretting testing rig incorporated with an advanced control system and an accurate data acquisition system. These ensured that the test results are reliable. The relationships between wear and normal load, wear and sliding distance, wear and frictional work were determined.

Advanced surface analysis techniques were used to better understand the heat exchanger tube wear problem. It was found that the roughness and its standard deviation of the tube and the disc were much the same after wear, but surface roughness was neither directly related to the sliding distance nor to the normal load. Plastic deformation was observed. Oxidation became an important mechanism for carbon steel disc/Incoloy tube combination even at room temperature.

Based on the experimental results obtained, it was found that the dynamic model by Lin and Cheng was quite suitable for the heat exchanger tube wear. The calculated results satisfactorily matched the test results. This model has been extended to calculate tube wear depth.
Table of Contents

Abstract

Tables of Contents

List of Tables

List of Figures

Acknowledgement

1 Introduction

2 Literature Survey

2.1 Heat Exchanger

2.2 Wear Mechanisms

2.3 Some Results of Wear Research

2.4 Sliding Wear Models

3 Experimental Tests

3.1 Test Procedure

3.2 Test Rig

3.3 Test Conditions and Material Properties

3.4 Test Results
4 Surface Analysis

4.1 Material Transmission 53
4.2 Plastic Deformation 55
4.3 Surface Profile with Number of Sliding Cycles 64
4.4 Surface Profile with Normal Load 69
4.5 AECL High Temperature Worn Tube Analysis 72

5 A Model for the Heat Exchanger Tube Wear 79

5.1 Some Previous Model Analysis 79
5.2 Wear Phenomena and Test Results Analysis 84
5.3 Modelling 90
5.4 Calculations 95
5.5 Equation Extension 107

6 Conclusions and Further Study 111

Bibliography 114

Appendices

A Test Results 120

B Calculation of Jain and Bahadur Model 124

C Equations of Gaussian Distribution 128
# List of Tables

<table>
<thead>
<tr>
<th>Table</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>Archard Wear Equation Extension</td>
<td>24</td>
</tr>
<tr>
<td>3.1</td>
<td>Chemical Composition of Materials</td>
<td>44</td>
</tr>
<tr>
<td>3.2</td>
<td>Mechanical Properties</td>
<td>44</td>
</tr>
<tr>
<td>4.1</td>
<td>Surface Parameters For The Group 1 Brass Samples</td>
<td>68</td>
</tr>
<tr>
<td>4.2</td>
<td>Surface Parameters For The Group 3 Brass Samples</td>
<td>68</td>
</tr>
<tr>
<td>5.1</td>
<td>Topographic Parameters</td>
<td>89</td>
</tr>
<tr>
<td>5.2</td>
<td>Initial Input Data</td>
<td>96</td>
</tr>
</tbody>
</table>
List of Figures

1.1 Classification of Wear 2
1.2 Typical Nuclear Power Station Schematic Diagram 4
1.3 Steam Generator Schematic Diagram 5
1.4 Worn Tube in Heat Exchanger 6
2.1 Multispan Test Rig 10
2.2 High Temperature Test Rig 11
2.3 Effect of Temperature on the Wear Rate of Incoloy 800 Tubing 11
2.4a Effect of Support Land Length on Tube Wear 12
2.4b Tube Fretting in Interrupted Hole Supports 12
2.5 Single Span Tube Test Rig 13
2.6 Tube to Support Plate Region 14
2.7 Distribution of Contact Pressure for a Smooth Surface 19
2.8 Distribution of Contact Pressure for a Rough Surface 20
3.11 Normal Load vs Wear Loss

3.12 Total Frictional Work vs Wear Loss (brass disc)

3.13 Total Frictional Work vs Wear Loss (stainless steel disc and tube)

3.14 Total Frictional Work vs Wear Loss (carbon steel disc, cutting fluid lubricant)

3.15 Total Frictional Work vs Wear Loss (tube, with carbon disc, cutting fluid lubricant)

3.16 Total Frictional Work vs Wear Loss (carbon steel disc and tube, distilled water)

4.1 EDX Analysis of Material Transmission

4.2 SEM Analysis of Material Transmission

4.3 Brass Disc Wear Scar Map

4.4 Plastic Flow on Worn Brass Disc

4.5 Original Grain Boundaries of The Brass Disc

4.6 The Deformed Grain Boundaries

4.7 3-Zone Schematic Diagram

4.8 Zone 3 and Cracks

4.9 Contour Plot of $\tau_{\text{oct}}/P_0$ for $\mu=0.164$
4.10 The Coordinate Schematic Diagram

4.11 The Microstructure of Y-Z Plane

4.12 Grain Deformation Near The Contact Centre

4.13 Contour Plot of $\tau_{yz}/P_0$ for $\mu=0.164$

4.14 Contour Plot of $\sigma_x/P_0$ for $\mu=0.164$ (y-z plane)

4.15 Contour Plot of $\sigma_y/P_0$ for $\mu=0.164$ (y-z plane)

4.16 Original Ground Surface

4.17 Brass Disc Surface After 9900 Cycles

4.18 Brass Disc Surface After 30,000 Cycles

4.19 Brass Disc Surface After 60,000 Cycles

4.20 Brass Disc Surface After 150,000 Cycles

4.21 Brass Disc Surface After 300,000 Cycles

4.22 Brass Disc Surface After 450,000 Cycles

4.23 Brass Disc Surface Under 50 N

4.24 Brass Disc Surface Under 100 N
4.25 Brass Disc Surface Under 200 N 70
4.26 Brass Disc Surface Under 300 N 71
4.27 Brass Disc Surface Under 400 N 71
4.28 Tube Section (tested with carbon steel A36 ring) 73
4.29 Tube Section (tested with Inconel 600 ring) 74
4.30 Tube Cross Section Ni Composition Spectrum 75
4.31 Tube Cross Section Cr Composition Spectrum 76
4.32 Tube Cross section Fe Composition Spectrum 77
4.33 Tube Cross Section Mn, Al, Ti composition Spectra 78
5.1 Schematic Diagram of Energy Distribution 83
5.2 Three Types of Wear 85
5.3 Three Regions of Wear 86
5.4 Brass Discs Against Incoloy 800 Tubes Test Results 86
5.5 The Relationship Between Tangential Load and Wear 87
5.6 Comparison of Test Results and Calculation Results (brass disc, diluted cutting fluid lubricant) 97
5.7 Comparison of Test Results and Calculation Results (ss410 disc, diluted cutting fluid lubricant) 98
5.8 Comparison of Test Results and Calculation Results (tube, tested with ss410 disc, diluted cutting fluid lubricant) 98
5.9 Comparison of Test Results and Calculation Results (carbon steel 12L14 disc, distilled water lubricant) 99
5.10 Comparison of Test Results and Calculation Results (tube, tested with carbon steel 12L14 disc, distilled water lubricant) 99
5.11 Comparison of Test Results and Calculation Results (carbon steel 12L14 disc, diluted cutting fluid lubricant) 100
5.12 Comparison of Test Results and Calculation Results (tube, tested with carbon steel 12L14 disc, diluted cutting fluid lubricant) 100
5.13 Difference Between Test Results and Prediction (brass disc, diluted cutting fluid lubricant) 101
5.14 Difference Between Test Results and Prediction (ss410 disc, distilled water lubricant) 101
5.15 Difference Between Test Results and Prediction (tube, against ss410 disc, distilled water lubricant) 102
5.16 Difference Between Test Results and Prediction (carbon steel disc 12L14 disc, distilled water lubricant) 102
5.17 Difference Between Test Results and Prediction (tube, against carbon steel 12L14 disc, distilled water lubricant) 103

5.18 Difference Between Test Results and Prediction (carbon steel 12L14 disc, diluted cutting fluid lubricant) 103

5.19 Difference Between Test Results and Prediction (tube, against carbon steel 12L14 disc, diluted cutting fluid lubricant) 104

5.20 The Relationship Between Wear Rate and Sliding Distance 105

5.21 The Relationship Between Antiwear Strength and Sliding Distance 105

5.22 σ Influence on Antiwear Strength U 106

5.23 h Influence on Antiwear Strength U 107

5.24 Tube/Ring Schematic Diagram 110

5.25 Tube/Ring Misalignment Schematic Diagram 110

xii
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Chapter 1

Introduction

Wear, as a phenomenon, has been recognized for a long time. However, it has been a relatively well defined discipline of science for only about thirty years. Wear is generally defined as "the progressive loss of substance from the operating surface of a body occurring as a result of relative motion at the surface."[1]. Wear is one of the major disciplines in tribology, the other two are friction and lubrication.

Wear is an extremely complex process. The variables which affect wear life have been compared to as many as those which affect human life [15]. Material properties, surface topography, component geometry, environmental factors such as lubricant, temperature, load, speed, sliding distance, and chemical reactions can affect the substance loss of the two contacting bodies. These parameters are related in a very complex way which is often hidden behind a curtain of mystery. Unlike most other subjects in mechanical engineering, the wear process usually happens on a microscopic level, i.e. the substance loss is not in tons or kilograms, but in milligrams or even less; the dimensions are not in meters or centimetres, but in millimetres or microns. Therefore, assumptions of "homogeneous", "isotropic", "continuity" which engineers are familiar with have to be reconsidered. For example, the material properties at the contact surface are usually very different from the bulk properties, since the microstructure of the material has been changed due to machining, heat, lubrication, and even chemical reaction. It is obvious that special considerations are needed to deal with wear.

In order to solve certain complex scientific problems, a knowledge of several branches of science is needed. Tribology is one such example, it requires a knowledge of material science, surface physics, mathematics, mechanics and chemistry. Many papers have been published in the
field of tribology over the past thirty years. Advancements made in different research fields have helped to clarify our understanding of the problem. Further clarification is due to the advancement in modern scientific instruments which play a very important role in wear study, since the micro-process of wear can not be observed directly.

Wear depends on numerous conditions, which make the classification of wear very difficult. As Ko pointed out [2], the simple definition of wear embraces a multitude of complex mechanisms involving a diversity of processes, such as adhesion, abrasion, fatigue and corrosion etc. One, or a combination of these processes may be in operation in any particular instance of wear. The interaction among these processes together with a variety of complex conditions at and below the contacting interface further adds to the complication. Among the various schemes to classify wear are: (1) by relative motions, (2) by the mechanism of particle removal and (3) by the type of wear. Figure 1.1 shows these various schemes. In his paper, Ko [2] also outlined the various wear modes and wear mechanisms.

Figure 1.1 Classification of Wear
Chapter 1. Introduction

In engineering practice, there is a need for wear equations in order to develop wear models which can be used to design mechanical components for better performance and reliability leading to reduced running cost and maintenance. Studies have been made which show the potential savings that tribology can have in modern industry. A report [3] issued in the U.K. in 1966 estimated that there were savings of £515 million/year to be achieved through better use of available tribological knowledge in the design, production and maintenance operations of industry. In Canada, the result of a thorough investigation [4] in 1982 estimated the cost of the tribological losses to the Canadian economy was about $5 billion. It is thought that these losses could be reduced by about 25% through cost-effective programmes of research, development and technology transfer.

A concern in the nuclear power industry is the failure of the steam generator tubes by wear. This problem has become a high priority item in Canadian nuclear generating stations. Figure 1.2 is a schematic diagram of a typical nuclear power station: in the nuclear reactor, the dissociation of atoms generate large amounts of heat which is carried away by the closed circuit heavy water cooling system. At the next stage in the steam generator, the heavy water transfers the heat to the feedwater which is turned into steam for the turbines to generate electricity. As Figure 1.3 shows, a nuclear steam generator may contain over 4000 tubes each about 20 feet long and supported by several horizontal plates. Flow induced vibration causes the tubes to wear by impacting and rubbing on the support plates and/or with adjacent tubes. The demand for high efficiency has resulted in higher flow rate with a consequential increase in the dynamic forces between the tubes and their supports. Since the tubes separate the heavy water and the light water, the resulting wear may lead to heavy water leakage from the tubes (see Fig. 1.4). This simple mechanical failure may result in contamination of the environment and radioactive exposure to station personnel. The current practice is to simply plug a worn tube so that it no longer function. Downtime of the station is extremely costly and must be avoided. A statistical investigation [5] carried out in a CANDU power station showed that the power output loss due to the failure of major heat exchangers was 2300 GW.h in 1979 alone.
Figure 1.2 Typical Nuclear Power Station Schematic Diagram
Figure 1.3  Steam Generator Schematic Diagram
Since the early '70s, there has been a major effort to develop analytical techniques to predict flow-induced vibration and wear phenomena. Ko [7] [8], Blevins [9] [10] Connors [14] and recently Magel [17] have performed research on this kind of problem. The objective of this research project is basically to extend Ko and Magel's work to reach two goals: the first, to understand the main factors that cause tube wear, by working on a further fundamental investigation; the second, to find out the main quantitative relationships among wear parameters in order to build up a sliding wear model which can reasonably predict laboratory test results and which may then be used for real engineering design.
Chapter 1. Introduction

This study is concerned with pure sliding wear only, chapter 2 will include a literature survey of the tube wear problem and previous fundamental investigations, chapter 3 will describe the new testing rig, testing conditions, and material properties and some test results, chapter 4 will include surface analysis such as surface parametric analysis, cross section microstructure analysis and other techniques. Chapter 5 will include some previous model analysis, wear phenomena analysis, the new wear model for the heat exchanger tube wear, the discussion of the calculation, and the model extension. The last part of the thesis, i.e. chapter 6, is the conclusions and suggestions of further study.
Chapter 2

Literature Survey

Steam generator and heat exchanger tube wear is mainly a flow-induced fretting wear problem. As the diagram in Figure 1.1 shows, fretting wear may involve several wear mechanisms: adhesion, abrasion, delamination, fatigue and corrosion. In the 1970's, Ko, Blevins and other researchers started to explore the tube wear problem. They conducted extensive controlled laboratory tests. Their research results have provided valuable information for subsequent analyses of tube wear mechanism and tube wear modelling. In order to develop a good wear model, it is important to understand their previous studies. This chapter includes a four-part literature survey: 1) Heat Exchanger, 2) Wear Mechanisms, 3) Results of Some Fundamental Study 4) Sliding Wear Models.

2.1 Heat Exchanger

2.1.1 The Environment Parameters of Heat Exchanger

Based on the information gathered by Mahail and Dueck [5] [6], physical data for the Canadian nuclear steam generators include the following:

- The temperature of the heavy water: 107~304°C
- Tube materials: Incoloy 800, Monel 400, Inconel 600, 70/30 Cu/Ni
- Support materials: carbon steel (A245B or SA.516 grade70), 70/30 Cu/Ni, brass 60/40
- Tube diameter: 12.7~19 mm
- Tube wall thickness: 1.12~1.65 mm
- Water pressure: 5171~9650 KPa.
Chapter 2. Literature Survey

2.1.2 Heat Exchanger Tube Parametric Study

Research into heat exchanger tube wear has progressed at several stages, each stage corresponding to the development of a new testing system. The first stage study relates to the flow induced vibration analysis. In 1980, Pettigrew et al. [7] studied flow induced vibration excitation mechanisms. Combining the dynamic flow analysis and multi-span room temperature dynamic testing (Fig. 2.1), they found that the main tube vibration frequency is about 20-40 Hz [8]. In reference [7], Pettigrew considered the vibration response to random turbulence excitation of a hypothetical four span tube and found the maximum random vibration response was 33 μm rms. He also estimated the impacting force between tube and support to be about 1.2 N rms. These figures indicate that tube wear is the result of low load, small amplitude vibrations - the conditions of fretting. Around the same period, Blevins [9] [10] [11] studied the fretting wear of heat exchanger tubes in a nitrogen/air mixture at room temperature. The effects of tube/tube support clearance, eccentricity, vibration frequency and mid-span displacement were investigated. Connors [14] also did valuable analysis of flow-induced vibration and wear of steam generator tubes.

The second stage study was a high temperature wear study. A testing facility used by Ko [8] consisted of a series of high temperature autoclaves connected to a pressurized loop simulating the high temperature/pressure operation environment (Fig. 2.2). This high temperature environment is especially important considering the effect of oxidation to the wear process. In this facility, each autoclave contains a single span tube with an end mounted vibration excitation system which drives the tube impacting and sliding on the support. Ko performed tube wear tests in pressurized water at temperatures up to 265 °C [8]. Figure 2.3 shows the wear relationship of Incoloy 800 tube to temperature. Two series of tests, one with carbon steel support specimens and the other with 304 stainless steel, were performed in two different excitation conditions. The results clearly show that the wear rate increases with temperature within the investigated range. Ko also studied the effect of support geometry on wear and concluded that the wear rate increases with the tube/support clearance, but wear depth decreases with increased contact length and support circumferential length (Fig. 2.4a, 2.4b).
Figure 2.1  Multispan Test Rig
Chapter 2. Literature Survey

Figure 2.2 High Temperature Test Rig

Figure 2.3 Effect of Temperature on the Wear Rate of Incoloy 800 Tubing
Chapter 2. Literature Survey

Figure 2.4a Effect of Support Land Length on Tube Wear

Figure 2.4b Tube Fretting in Interrupted Hole Supports
The third stage study involved more fundamental studies. Ko [12][16][44] studied the effect of impact force on wear at room temperature using a single-span tube test rig (Fig. 2.5) and later with a more advanced impact-fretting test rig (Fig. 3.2). He found that the wear rate increased linearly with tangential impact energy for brass and 304 stainless steel (the tangential impact energy = friction force x the total sliding distance). Ko emphasized that impact force alone cannot be directly correlated to wear rate when other parameters such as tube and tube support clearance and the type of motion are varying.

Figure 2.5  Single Span Tube Test Rig
2.1.3 Surface Analysis Techniques

In 1985, Hogmark et al. [13] had a rare opportunity to examine heat exchanger tubes taken from a nuclear power station. They applied surface analysis techniques (Energy Dispersive X-ray spectroscopy (EDX) and Auger Electron Spectroscopy (AES)) to analyze the worn tube. They found the tube was covered by a 0.1–2 μm oxide layer and the wear surface had a wavy topography. No plastic deformation was detected near the worn surface. Hogmark also found that the oxide layer is thinner (0.05–0.1 μm) in the worn area (Fig. 2.6 area 2) and thicker (0.5–2 μm) in the unworn area of the tube to support plate region (Fig. 2.6, area 3) as compared to the oxide layer thickness of the reference surface (Fig. 2.6, area 1). Therefore, they concluded that wear occurred by flow enhanced removal of material from the superficial layer of the oxide film.

Figure 2.6  Tube to Support Plate Region (dimensions in mm) [13]
Chapter 2. Literature Survey

2.2 Wear Mechanisms

Heat exchanger tube wear is primarily the result of impact-fretting which is related to adhesive, abrasive, delamination, fatigue and oxidative wear mechanisms. The definition and basic character for each of these mechanisms has been explained in many references [2][17][18]. This section will briefly discuss the mechanisms often associated with the heat exchanger tube wear.

2.2.1 Fretting Wear

Fretting itself is not a mechanism. Rather, it is a type of wear which arises when contacting surfaces undergo oscillatory tangential displacements of small amplitude (in microns). It represents the interaction of several wear mechanisms. Three names are often associated with fretting: fretting wear, fretting fatigue and fretting corrosion. Fretting wear is usually concerned with larger amplitudes of motion than fretting fatigue. Fretting fatigue occurs in situations where the combined effect of small vibration amplitude and surface friction results in only microslip at the contacting area. Fretting corrosion is due to combined mechanical and chemical actions resulting in the continuous removal of oxidized surface layers. As Quinn discussed in his study [48], fretting corrosion is clearly a form of oxidational wear, occurring under conditions where small oscillatory tangential motion occurs between two surfaces which are held together in such a way that the oxide wear particles cannot escape, thereby causing severe stresses to be set up in the constraining mechanism (e.g. in the bolt holding together two plates subjected to vibrations with extremely small amplitude (of the order of microns)).

Johnson [19] studied the mechanics of microslip as the fundamental cause of fretting. The oscillating force is a controlling parameter.

2.2.2 Adhesion

Adhesion is considered the most fundamental wear mechanism because it is a basic
phenomenon that takes place whenever two solid surfaces are in dry sliding contact. Since surfaces are always microscopically rough, in dry sliding contact, the load is carried by surface asperities. As the local pressures are very high, the yield strength of the softer material may be exceeded. The high local pressures and traction force generated by relative sliding may cause minute welds or junctions to form at these local contacts.

2.2.3 Abrasion

A abrasive wear is produced by hard particles or asperities which cut or groove one of the sliding surfaces. Harder particles may be from one of the two rubbing surfaces or may be formed by chemical and thermal related processes such as oxidation or the precipitation of carbides. The abrasive wear mechanism also includes third body wear. This type of wear occurs when hard, abrasive particles are trapped between two sliding surfaces. These hard particles may be products of wear (i.e. work hardened wear particles) or may otherwise be introduced from outside sources (dust, sand, soil). Moore [20] describes two mechanisms by which material may be removed during the abrasive wear process: wear particles can be removed as prows which form in front of a moving asperity due to plastic deformation or as chips due to fracture with limited plastic deformation.

2.2.4 Delamination Wear

Delamination wear is defined as the loss of material in the form of flakes, caused by the formation and propagation of subsurface fatigue cracks running parallel to the surface. This mechanism was first discussed by Suh and his co-workers at MIT [21]. In Suh's theory [21] [22] he assumes that a metal under a slider wears layer by layer similar to the removal of an onion skin. Each layer consists of many sheets. The creation of these wear sheets is assumed to be a cumulative process which results from the surface shearing a small amount with each passing asperity. A wear sheet may be created after a large number of asperities have passed each point on the surface.
2.2.5 Fatigue Wear

Fatigue is a broad term applied to the failure phenomenon where a solid is subjected to cyclic loading above a certain critical stress. The number of stress cycles necessary to cause failure increases with decreasing stress.

Fatigue wear on a microscopic scale is associated with individual asperity contacts. Failure occurs due to the repeated deformation and the relative displacement of the rubbing bodies.

2.2.6 Oxidation

Oxidation is the most common form of chemical attack on metallic wear surfaces. Corrosive or oxidative wear is the result of both corrosion and rubbing. It is usually considered a mild wear. Quinn and Sullivan [23] developed an oxidation wear model in which they assumed that wear occurs when the sliding system is attacked by atmosphere oxygen, and the corroded layer on the contact surface is subsequently rubbed off. Under static conditions, the oxide film acts as a barrier between the base metal and the absorbed oxygen inhibiting further oxidation, hence reducing the oxidation rate. In the dynamic case, i.e., with sliding, the oxide layer is rubbed away continuously. The oxidation rate therefore remains at the initial high value.

2.3 Some Results of Wear Research

2.3.1 The Effect of Surface Roughness

The effect of surface roughness on friction has been investigated by many researchers. Whitehouse [45] studied surface topography and quality and its relevance to wear. Stout et.al [46] studied the micro-geometry of lubricated wear. Gupta [47] discussed several key points in the general area of surface interaction and pointed out "a small change in sampling distance could result in significant changes in surface description". Jeng [24] found that lower roughness height yields lower friction, and that transverse roughness has lower friction than longitudinal
roughness and surface roughness effects become increasingly significant as the film thickness decreases.

Wang et al. [25] studied the relative surface conformation between two surfaces in sliding contact. Their research shows that the relative surface conformation rises with increasing test duration, during running in.

2.3.2 Reciprocating and Continuous Sliding Wear

In 1970, Ward [26] compared the wear rate of continuous and reciprocating sliding under similar load, speed and nominal area of contact. Higher wear rates were observed during reciprocating sliding for both mild and severe wear cases, the rate of increase of wear rate with load for severe wear was also greater under reciprocating conditions. The increased wear rate for reciprocating sliding was explained in terms of abrasion caused by the presence of loose wear debris i.e. the third body effects.

2.3.3 Contact Stresses for Nonconforming Surfaces

The stress distribution associated with smooth surfaces in contact are rarely experienced in practice. Factors such as surface roughness, lubricant films, and third body particulate are known to influence the state of stress and the resulting rolling contact fatigue life. Bailey and Sayles [27] developed a numerical technique for evaluating the complete subsurface field of stress based on the measurements of surface profiles resulting from the elastic contact of nonconforming rough bodies. The result revealed that the presence of asperities within the contact range resulted in highly stressed regions on the surface (Fig.2.7, 2.8), and that plastic deformation of these asperities during running in reduced the peak contact pressures. During sliding and rolling, the cyclic nature of these stresses often initiates fatigue cracks which will eventually cause failure of the contact surfaces.
Figure 2.7 (ref. [27])  Distribution of contact pressure and subsurface orthogonal and principal shear stresses for a smooth elastic Hertzian contact. (a) Contact geometry and surface pressure distribution; (b) Isometric view of orthogonal shear stresses; (c) Contour plot of orthogonal shear stress distribution shown in (b); (d) Effect of coefficient of friction $\mu = 0.1$ on stress distribution shown in (c); (e) Isometric view of principal shear stresses; (f) Contour plot of principal shear stress distribution shown in (c); (g) Effect of coefficient of friction $\mu = 0.1$ on stress distribution shown in (f).
Figure 2.8 (ref. [27]) Distribution of contact pressure and subsurface orthogonal and principal shear stresses for the simulated elastic contact of the unrun ground surface.
2.3.4 Misalignment

For a line contact case, misalignment is a common problem. It is mainly due to the difficulties in machining and mounting components to the accuracy of micron levels. The serious effect on wear by misalignment was studied by Kannel et al. [28]. Figure 2.9a, 2.9b, 2.9c show the relationship of misalignment and contact pressure. Kannel pointed out that at a misalignment level on the order of 0.86 to 1.25x10⁻³ radians, edge stresses on the order of 2 GPa have been measured and predicted. These stresses are more than twice those which would occur for ideal line contact. The high stress region will certainly result in increased wear.

![Graph showing experimental points and theoretical predictions for pressure profile with misalignment.](image)

Figure 2.9a (ref. [28]) - Axial pressure profile for nearly cylindrical disk at a load of 2224 N (500 lb) with no misalignment.
Figure 2.9b (ref.[28]) — Axial pressure profile for nearly cylindrical disk at a load of 2224 N (500 lb) and a misalignment of $0.86 \times 10^{-3}$ rad.

Figure 2.9c (ref. [28]) — Axial pressure profile for nearly cylindrical disk at a load of 2224 N (500 lb) and a misalignment of $1.25 \times 10^{-3}$ rad.
2.4 Sliding Wear Models

In the large quantity of published literature, hundreds of wear models have been suggested, each relating to a specific wear system. There is no generally accepted wear model available. The following sections briefly describe some of these models to provide different approaches for wear modelling.

2.4.1 Archard Wear Model

Archard's wear model which was published in 1953 [29], is the most widely quoted adhesive wear model. It is in a very simple form:

\[ V = k \frac{P}{H} \]

(2.1)

where \( V \) is wear volume which is proportional to the normal load \( P \) and travelled sliding distance \( L \) but inversely proportional to the softer material hardness \( H \). In this equation, \( k \) is a dimensionless constant known as the wear coefficient and can vary from \( 10^{-1} \) to \( 10^{-10} \).

The Archard wear model dominated both the theoretical and experimental approaches in the early period of wear study. In some cases, the Archard model can reveal the main factors that affect wear volume loss without getting into too much detail. The Archard model, which was originally for adhesive wear, was later extended to cover other wear mechanisms such as abrasive wear, fatigue wear, oxidation wear etc. (see table 2.1). In these equations, the wear coefficient \( k \) and its definition changes depending on the wear mechanism considered.
# TABLE 2.1 ARCHARD WEAR EQUATION EXTENSION

<table>
<thead>
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<th>TYPE OF WEAR</th>
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| Adhesive Wear | \( \frac{V}{L} = K_{adh} \frac{P}{H} \) | Erosive Wear | \( \frac{V}{L} = K \left[ \frac{1}{m} \frac{mv^2}{2} \right] \frac{1}{H} \)
| | \( K_{adh} = K \) | m = mass of particle | v = impact velocity |
| | \( L \) | for \( v \gg 1 \), | \( V = \frac{K}{6} \frac{mv^2}{H} + K \frac{v}{5} \) |
| Abrasive Wear | \( \frac{V}{L} = K_{abr} \frac{P}{H} \) | Fatigue Wear | \( \frac{V}{L} = K_{fat} \frac{P}{H} \)
| | \( \text{where } K_{abr} = K \frac{a}{\pi} \phi \) | (for plastic contact) | \( \text{where } \)
| | \( a \approx 1 \text{ to } 2 \) | | \( K_{fat} = \frac{0.18a}{n} \left[ \frac{h}{R} \right]^{1/2} \)
| Delamination Wear | \( \frac{V}{L} = K_{del} \frac{P}{H} \) | | \( \alpha = \text{ratio of nominal area of contact to frictional area of contact.} \)
| | \( \text{where } K_{del} = \frac{-K_{1}b_{1}}{C_{1}} + \frac{K_{2}b_{2}}{C_{2}} \) | n = ratio of plastic strain for fracture to the effective strain to the power \( t \), | \( t = \frac{\varepsilon_{f} - \varepsilon_{0}}{\varepsilon_{0}} \)
| | \( h = \text{thickness of layer} \) | | \( t = \text{fatigue curve exponent power} \)
| | \( C = \text{critical plastic displacement} \) | \( h = \text{depth of penetration} \) | \( R = \text{radius of curvature of asperity} \)
| | \( 1,2, \text{denote the two contacting components} \) | | \( \text{R} = \text{radius of curvature of asperity} \) |
| Oxidation Wear | \( \frac{V}{L} = K_{oxid} \frac{P}{H} \) | | \( \text{R} = \text{radius of curvature of asperity} \) |
| | \( \text{where } K_{oxid} = \frac{\beta e^{-[Q_{p}/RT]}}{[\eta \rho_{o}]^{2}} \nu \) | | \( \text{R} = \text{radius of curvature of asperity} \) |
| | \( \beta e^{-[Q_{p}/RT]} \) | | \( \text{R} = \text{radius of curvature of asperity} \) |
| | \( \text{parabolic oxidation constant} \) | | \( \text{R} = \text{radius of curvature of asperity} \) |
| | \( \beta, Q_{p} = \text{constants for given material} \) | | \( \text{R} = \text{radius of curvature of asperity} \) |
| | \( R = \text{universal gas constant} \) | | \( \text{R} = \text{radius of curvature of asperity} \) |
| | \( T = \text{temperature} \) | \( t = \frac{\varepsilon_{f} - \varepsilon_{0}}{\varepsilon_{0}} \)
| | \( f = \text{fraction of oxide which is oxygen} \) | \( t = \text{fatigue curve exponent power} \)
| | \( \eta = \text{critical oxide thickness} \) | \( h = \text{depth of penetration} \)
| | \( \rho_{o} = \text{average density of oxide} \) | \( \text{R} = \text{radius of curvature of asperity} \)
| | \( v = \text{velocity of sliding} \) | \( \text{R} = \text{radius of curvature of asperity} \)
2.4.2 Jain and Bahadur Wear Model

The wear model by Jain and Bahadur [30] was developed for the sliding wear of polymeric material. It was later verified experimentally [31]. They conjectured that the interactions between asperities that occur during the sliding of surfaces lead to cyclic contact and reversals of principal tensile stress, and this tensile stress was responsible for the nucleation and propagation of fatigue cracks.

Their wear model included the following assumptions:

1) The height of asperities on both surfaces in contact varied randomly. It was later assumed to be a Gaussian distribution.

2) The asperities have spherically shaped tips.

3) The asperities on one surface are aligned with those on the other surface and have the same pitch in the sliding direction.

4) The deformation in the contact zone is of an elastic nature.

5) The discrete contact zones were sufficiently separated to act independently of each other.

Four steps were used to derive the wear equation:

First, the real contact area is determined by applying the Hertzian equation for each individual asperity.

The Hertzian equation for elastic contact of a sphere with a smooth plane provided the following expression for the contact radius $a_1$, the area $A_1$ and the load $P_1$ [32]:

\[ a_1 = \beta^{1/2} \omega^{1/2} \]  \hspace{1cm} (2.2)

\[ A_1 = \pi \beta \omega \]  \hspace{1cm} (2.3)
\[ P_1 = \frac{A}{3} \left( E' \beta^{y_2} \gamma^2 \right) \]  \hspace{1cm} (2.4)

where

\[ \frac{1}{E'} = \frac{1-\gamma_1^2}{E_1} + \frac{1-\gamma_2^2}{E_2} \]  \hspace{1cm} (2.5)

\[ \gamma \] is the Poisson's ratio, \[ \omega \] is the compliance (distance by which the points outside the contact zone move closer owing to deformation), \[ \beta \] the radius of the spherical asperity and \[ E \] the modulus of elasticity. The subscripts 1 and 2 refer to the two surfaces in contact.

Figure 2.10  Schematic Representation of Sliding Contact Between Two Rough Surfaces
Greenwood and Williamson [33] extended the above single contact equations to the case of contact between a rough and a smooth surface (Fig. 2.10) to develop expressions for the number of the discrete contact zones \( n_0 \), the area of real contact \( A_r \), and load \( P \). By normalizing with respect to the standard deviation of the asperity height distribution, Jain and Bahadur arrived at the following:

\[
n_0 = \eta A_0 \int_d^\infty \phi(z) dz = \eta A_0 F_0(h) \tag{2.6}
\]

\[
A_r = \pi \eta A_0 \beta \int_d^\infty (z-d) \phi(z) dz = \pi \eta A_0 \beta \sigma F_1(h) \tag{2.7}
\]

\[
P = \frac{4}{3} \eta A_0 E' \beta^{1/2} \int_d^\infty (z-d)^{3/2} \phi(z) dz = \frac{4}{3} \eta A_0 E' \beta^{1/2} \sigma^{3/2} F_{3/2}(h) \tag{2.8}
\]

where \( \eta \) is the surface density of asperities, \( A_0 \) is the nominal area of contact, \( \phi(z) \) is the distribution of asperity heights and \( d \) is the distance between the reference planes of the surfaces in contact. In this equation, \( h \) and \( s \) are standardized variables, which were obtained by normalizing the heights \( d \) and \( z \) with respect to the standard deviation \( \sigma \) of the asperity height distribution such that \( h = d/\sigma \) and \( s = z/\sigma \). The generalized function \( F_n(h) \) is defined as

\[
F_n(h) = \int_h^\infty (s-h)^n \phi(s) ds
\]

For the case of two rough surfaces in contact, equivalent values for \( \beta \) and \( \sigma \) are modified as:

\[
\beta = \frac{\beta_1 \beta_2}{\beta_1 + \beta_2}, \quad \sigma = \left(\sigma_1^2 + \sigma_2^2\right)^{1/2} \tag{2.9}
\]

(The subscripts 1 and 2 refer to the two contact surfaces.)
The second step was to determine the principal stresses in a circular contact zone by using the Hamilton and Goodman analysis [34]. This analysis showed that two of the principal stresses in the contact zone were largely compressive in nature. As such, these would not be expected to contribute to the initiation of a fatigue crack. The third principal stress was tensile in front of the contact zone and compressive behind it. This stress was assumed to be responsible for the fatigue failure. The expression derived by Hamilton and Goodman for the maximum tensile stress, $S$, is

$$S = \frac{3P_1}{2\pi a_1^2} \left( \frac{\mu}{8} (4+\gamma) \pi + \frac{1-2\gamma}{3} \right)$$  \hspace{1cm} (2.10)$$

where $\mu$ is the friction coefficient, $\gamma$ is the Poisson's ratio. In order to determine the tensile stress $S$, the load $P_1$ and the contact radius $a_1$ for an individual asperity are needed. Greenwood [35] has shown that the average size of discrete microcontacts was nearly constant for an elastic contact situation. Therefore $P_1$ and $a_1$ could be determined from the following equations:

$$P_1 = \frac{P}{\eta A_0 F_0(h)}$$  \hspace{1cm} (2.11)$$

$$a_1 = \left( \frac{A_r}{\pi n_0} \right)^{1/2} = \left( \frac{\beta \sigma F_1(h)}{F_0(h)} \right)^{1/2}$$  \hspace{1cm} (2.12)$$

Substituting equations 2.11 and 2.12 into equation 2.10, the tensile stress $S$ is expressed as

$$S = \frac{3k_t P}{2\pi \eta A_0 \beta \sigma F_1(h)}$$  \hspace{1cm} (2.13)$$
where

\[ k_1 = \frac{\mu}{8} (4 + \gamma) \pi + \frac{1 - 2 \gamma}{3} \]

The third step involved the asperity fracture criterion, i.e. the fatigue failure criterion to determine the failure in an asperity. The bulk fatigue property of a material was represented by Wöhler's curve, the equation for which is [36]

\[ N_f = \left( \frac{S_0}{S} \right)^\gamma \]  

(2.14)

where \( N_f \) is the number of cycles to failure, \( S_0 \) is the failure stress corresponding to the application of a single stress cycle, \( S \) is the applied cyclic stress and \( t \) is a material constant. The factors \( S_0 \) and \( t \) for a material were determined from the plot of its fatigue data.

Substituting equation 2.13 into equation 2.14:

\[ N_f = \left( \frac{2 \pi S_0 \eta A_0 \beta \sigma F_1(h)}{3 k_1 P} \right)^\gamma \]  

(2.15)

The last step was the final wear equation. The starting point of the wear equation is

\[ V_R = N_w v_p \]  

(2.16)

where \( V_R \) is the volume wear rate, \( v_p \) is the average volume of a wear particle assuming the asperities were all in a spherical shape and \( N_w \) is the number of wear particles formed per unit time.

It is assumed that one surface in the wear system is moving, while the other is stationary. If \( \eta_L \) is the line density of the asperities on the moving surface and \( L \) is the sliding distance, then \( \eta_L L \) is the number of asperities considered on the moving surface. Remember \( n_0 = \eta_A F_0(h) \) is the number of asperities for the unmoving surface. The asperity interactions in a sliding
distance \( L \) between these two surfaces is \( L \eta \eta A_o F_0(h) \). The asperity encounter rate \( N_R \) for a sliding speed \( u \) will be

\[
N_R = u \eta \eta A_o F_0(h) \quad (2.17)
\]

The number of wear particles \( N_w \) formed is expressed in terms of the asperity encounter rate \( N_R \) divided by the number of loading cycles \( N_f \) needed to cause fracture, i.e.

\[
N_w = \frac{N_R}{N_f} \quad (2.18)
\]

Substituting equations 2.15, 2.17, 2.18 into equation 2.16, gives the volume wear rate \( V_R \)

\[
V_R = \eta \eta \mu F_0(h)(k_1 P)^t(A_0 \eta)^{1-t} \\
\left[ \left( \frac{2\pi}{3} \right) S_0 \beta \sigma F_1(h) \right]^t
\]

Substituting \( P \) from equation 2.8 into \( V_R \), the wear volume \( V \) for a sliding distance \( L \) is in a very complex form:

\[
V = \frac{K_1 P L \eta \eta \mu}{2S_0} \left( \mu \left( \frac{4+\gamma}{8} \right) \pi + \frac{1-2\gamma}{3} \right) \quad (2.19)
\]

where

\[
K_1 = \frac{2k_1 E'}{\pi S_0} \left( \frac{1}{\beta} \right)^{t-1} \left( \frac{1}{\gamma} \right)^{t-1} \sigma \gamma^2 \left( \frac{F_3 \sigma(h)}{F_1(h)} \right)^{t-1} \frac{F_0(h)}{F_1(h)}
\]

2.4.3 Lin and Cheng Wear Model

The wear model by Lin and Cheng [37] is a recent model which considers the dynamic process of wear.
In this model, it is postulated that the wear rate is proportional to a forcing term $F$ induced by the shear force at the asperity contacts, and inversely proportional to a wear resistance term $U$, which is related to the surface antiwear strength. Both $F$ and $U$ are time dependent or wear dependent because wear progress causes the material strength at various layers to change.

The wear equation is of the form:

$$ W_R = \frac{k F}{U} \quad (2.20) $$

where $W_R$ is the wear volume per unit sliding distance and $k$ is a dimensionless constant which may be determined either experimentally or theoretically.

The shear force $F$ was obtained by averaging the shear force over $h$ which is the elastically and plasticly deformed layer beneath the loaded asperity. The shear force was expressed by the shear stress $\tau_i(x,y,z)$ (at point $(x,y,z)$ and time $t$) integral in the area, i.e.

$$ F = \frac{1}{h} \int_{0}^{h} \left( \int_{0}^{h} \tau_i(x,y,z) dx \right) dy dz \quad (2.21) $$

For a rough surface contact, the shear force at $z$ in equation (2.21) can be represented by an average shearing stress over all asperity areas at $z$, $\tau_{i_{avg}}(z)$, times $A_0 \Phi_i(z)$ where $A_0$ is the nominal area and $\Phi_i(z)$ is the time dependent cumulative probability function of the asperity height distribution at time $t$. Thus, $F$ becomes

$$ F = \frac{A_0}{h} \int_{0}^{h} \Phi_i(z) \tau_{i_{avg}}(z) dz \quad (2.22) $$

where

$$ \tau_{i_{avg}}(z) = \frac{\sum_j \int_{A_j} \tau_i(x,y,z) dA_j}{\sum_j A_j} $$

$A_j$ is the area of asperity $j$ at depth $z$. 

The antiwear strength $U$ is defined as

$$U = \frac{1}{A_0 h} \int_0^h \int \int \hat{\delta}_f(x,y,z) \, dx \, dy \, dz$$  \hspace{1cm} (2.23)$$

where $h$ is the deformed layer beneath the loaded asperity, and is of the order of the average radius of the asperity contacting area. The flow strength $\delta_f(x,y,z)$ represents the ability of a material to resist plastic flow at point $(x,y,z)$ and time $t_i$. The flow strength is a microscopic property and can be related to the microhardness or the yield strength of the material.

Since the geometrical area of contact is composed of many asperity contact areas, the integration of $\delta$ is the summation of the integral for each asperity contact area. Thus

$$U = \frac{1}{A_0 h} \int_0^h \left( \sum_j \int_{A_j} \int \hat{\delta}_f(x,y,z) \, dA_j \right) \, dz$$

$$= \frac{1}{h} \int_0^h \Phi(z) \delta_{i, \text{avg}}(z) \, dz$$  \hspace{1cm} (2.24)$$

where $A_j$ is the area of asperity $j$ at depth $z$, and

$$\delta_{i, \text{avg}}(z) = \frac{\sum_j \int_{A_j} \hat{\delta}_f(x,y,z) \, dA_j}{\sum_j A_j}$$

is the average flow strength of the material at depth $z$. For a homogeneous material, the average flow strength can be the yield strength. As shown in equation 2.24, the antiwear strength combines both geometrical and mechanical properties of the material. The flow strength $\delta_{i, \text{avg}}(z)$ may increase due to work hardening or decrease due to the initiation and propagation of cracks.
Chapter 2. Literature Survey

For pure adhesive wear of a homogeneous material under constant friction and a steady wear condition, Lin and Cheng were able to reduce their model to the Archard equation. From the preceding definitions, wear, antiwear strength and averaged shear force are time/wear dependent. Therefore, this model is able to describe the nonlinear running-in wear behaviour.

2.4.4 Magel Wear Model

Magel's model [17] was originally derived for a single hard sphere sliding on a flat surface. Later, the model was applied to multiasperity heat exchanger tube wear. The model is based on the consideration of a certain amount of plastic deformation, i.e. shakedown, during the wear process.

When a hard indenter is pressured against a softer flat surface, high pressures induced by the spherical contact may exceed the elastic yield point and plastic deformation will commence. For the first few cycles the system will experience certain plastic deformation. With continuing deformation, an indentation is formed as the material conforms to the geometry of the indenter. The maximum pressure beneath the indenter decreases as the pressure distribution changes to resemble that of a line contact. At the same time residual stresses are induced. The specimen will no longer deform if the pressure beneath the indenter does not exceed the shakedown pressure $p_s$.

After several cycles, the deformed material reaches elastic shakedown and the indentation will have a geometry such that the maximum stress at all points of contact between the indenter and the material is $P_0$. Also, steady state deformation will be entirely elastic (with a new, effective elastic yield point of $P_0$). The area of contact between the indenter and conformal indentation will appear as a long ellipse and for an ellipticity ratio greater than five, the contact can be considered a line contact. The resulting pressure distribution will be as that shown in Figure 2.11.
The total load applied by the indenter is represented by the area under the pressure distribution, i.e.

\[ P(x) = \frac{\pi}{4} P_0^S (2a - 2b) = \pi P_0^S ab \]  

(2.25)
From Hertzian contact theory, the line contact formulae are

\begin{equation}
\frac{a}{2} = \left( \frac{P'R_L}{\pi E'} \right)^{1/2} \tag{2.26}
\end{equation}

\begin{equation}
\frac{P_0}{\pi R_L} = \left( \frac{P'/E'}{E'} \right)^{1/2} \tag{2.27}
\end{equation}

Magel assumed \( P_0 = P_0^s \) by shakedown. Notice that \( P' \) is the load/unit length, \( a \) is the semi-contact width, and combining equations 2.26 and 2.27 yields:

\begin{equation}
\frac{a}{R_L} = \frac{2P_0^s}{E'} \tag{2.28}
\end{equation}

where

\begin{equation}
E' = \left( \frac{1-\gamma_1^2}{E_1} + \frac{1-\gamma_2^2}{E_2} \right)^{-1}
\end{equation}

and

\begin{equation}
R_L = \left( \frac{1}{r_L} - \frac{1}{P} \right)^{-1}
\end{equation}

where \( r_L \) is the radius of the indenter in the longitudinal direction, which is assumed constant, \( \rho \) is the longitudinal curvature of the wear scar.

Substituting equations for the semi-contact width \( a \) and the semi-line contact length \( b \), equation 2.25 can be expressed as
where \( r_T \) is the transverse (perpendicular to the sliding direction) radii of the wear scar curvature and \( l \) is the half sliding distance in one pass. Magel assumed that cracks initiate and propagate at the location of maximum octahedral shear stress, therefore the thickness of a wear particle can be calculated.

Magel assumed the basic wear parameter relationship is

\[
P(x) = \frac{2\pi l}{E'} \left( \frac{P_0}{P} \right)^2 r_T \left( \frac{1}{r_L} - \frac{1}{\rho} \right)^{-1} \left( 1 - \frac{x^2}{l^2} \right)
\]

where \( V \) is the total wear volume, \( W \) is the total frictional work input, \( v_p \) is the particle volume, and \( E_p \) is the tangential input energy to create the particle \( v_p \). Note that \( E_p \) is equal to \( \tau_y A_p l \) (where \( A_p \) is the particle area, \( \tau_y \) is the yield shear strength, \( l \) is the sliding amplitude).

Magel extended the model to non-smooth surfaces in sliding, following the method of Jain and Bahadur (see section 2.4.2, equation 2.3). According to Magel's experimental results, the single indentation geometry is well predicted, but the multi asperity wear model dramatically overestimates the wear volumes primarily due to the simplified evaluation of \( E_p \).
Chapter 3

Experimental Tests

Experimental tests are very important in the study of wear. It enables researchers to understand the relative importance of various parameters. However, test results must be interpreted carefully, since it is easy to jump to wrong conclusions. In order to isolate each parameter during a test, a well controlled wear test system is very important. It is necessary to document clearly test conditions and material properties so that the tests can be repeated and the results compared with others. This chapter describes the test rig, test conditions, material properties, test procedure, and test results.

3.1 Test Procedure

Figure 3.1 shows the overall path of specimen preparation and analysis. Specimen preparation consists of machining of specimens, polishing the testing surfaces to the desired finish, cleaning and weighing prior to testing.

**Machining:** Since wear is sensitive to material properties and heat treatment, it is important that the material for each group of specimens comes from the same batch. In this study, the material used for the discs are cut from 1 inch diameter rods and the disc surfaces are machine ground.

**Polishing:** For some wear studies, particularly those involving wear particle formation mechanisms, controlled precision polishing of each specimen surface is necessary. For the present study, the majority of samples were polished by 1000 grit SiC paper, some of the samples were polished to a 1 micron diamond finish. SEM analysis samples were all polished with 0.05 μm aluminum microfinish.
Cleaning: The specimens were cleaned to eliminate surface films and contaminants prior to testing. A second cleaning removed loose wear particles that remained at the surface after the test. The cleaning procedure involves placing the specimens in an ultrasonic bath containing ethenol three times.

Weighing: Wear losses were obtained by weighing the specimen before and after testing, in both cases after cleaning. An analytical balance with digital readout to 10 μg was used, the reliability being about 0.02 mg.

For analysis, it is usually necessary to know the specimen microstructure changes, surface profile or even perform Auger and EDX analyse and Micro -Prob Analysis. Detailed surface analysis in this study will be discussed in the next chapter.
3.2 Test Rig

In this thesis, most of the tests were carried out in the NRC Fretting-Impacting Test Rig, a detailed description of which may be found in references [12] [17]. Figure 3.2 shows that excitation is from the two shakers mounted at 90° to each other. Figure 3.3 shows the holder detail. The aluminum ring shaped dynamic specimen holder is connected to both shakers. Shaker 1 provides the sliding displacement and shaker 2 provides the normal load between the dynamic and the static specimens. The normal load and frictional load can be measured by the triaxial load cell located under the stationary specimen (e.g. Fz=normal load, Fx=friction load, Fy=0, in this special application). The stationary specimen assembly includes the load cell attached to a semi-circular bar with its curved surface anchored at the lower half of the autoclave. A holder which mounts a spherical specimen is fixed in the dynamic holder. By adjusting the DC level of shaker 2, the preload between the two contact specimens can be set. In this study, most of the tests have a 5 N preload.

The two shakers with phase-locked signals control the dynamic specimen loading and provide impacting, sliding, oblique impacting or combined impact/sliding motion. For pure sliding tests, the two signals are phase locked at 270°.

The original NRC plug-in dynamic specimen holder was for a spherical shaped specimen. Since tube wear is the main concern in this work, a tube specimen holder (see Fig. 3.4) that can be fixed in the dynamic specimen holder was designed. The tube holder has self-aligning capability for providing near perfect line contact. An isometric diagram of the self-aligning tube holder is shown in Figure 3.4. It contains a main body and a self-aligning component, the main body slide fits into the ring shaped dynamic holder, and the self-aligning part with the mounted tube specimen is connected to the main part by two pins so that it can rotate freely for self-aligning.
Chapter 3. Experimental Tests

Figure 3.2 NRC Fretting-Impacting Test Rig

Figure 3.3 Dynamic Holder For Sphere Specimen
Figure 3.4 The Self-Aligning Tube Holder

Figure 3.5 is a schematic diagram of the test system. The system consists of three parts:

1. **Input Parameter Control System**: This system provides the signal for actuating the shakers. It produces the normal load and displacement signals from computer 1. The shape of the waveform, signal frequency and test duration can be selected. The attenuator is for adjusting the normal load and displacement in the open-looped system. More recently, the open-looped system has been converted to a close-looped system with much improved control accuracy. Thus, the input for normal load and displacement can be prescribed initially and controlled throughout the test with the closed-loop system. For the present series of tests, the normal load is always a half sine wave and the displacement a full sine wave.

2. **Test Section**: This part includes the test rig and signal sensors for monitoring the normal force, friction force and displacement. The force signals are conditioned with charge amplifiers. The displacement limitation has been increased from 3mm to 6mm.
Chapter 3. Experimental Tests

Figure 3.5 Full Test System Schematic Diagram
3. **Data Acquisition System**: Computer 2 is used to store and analyze the full-term test results with the help of an external 100 Mbyte hard drive. An improved version of the acquisition software has auto-triggering, adjustable sampling rate, selective sample length and force gain functions. The data acquisition system includes an analysis package that calculates several parameters including the maximum, minimum, and average displacement, normal and shear force; average friction coefficient, number of running cycles, and the total frictional work. Figure 3.6 shows an example of friction coefficient vs running cycles.

Calibration of the data acquisition system has been performed by comparing the analytical values with those obtained by the data acquisition. The differences were found to be less than 2% for the normal force, shear force, displacement, and less than 3% for the total work input (sine wave frequency is 20 Hz, sampling frequency is 300 Hz).

![Diagram of Friction Coefficient vs Cycles](image)

**Figure 3.6** Diagram of Friction Coefficient vs Cycles
Chapter 3. Experimental Tests

3.3 Test Conditions and Material Properties

Except where indicated, all tests were lubricated with diluted cutting fluid or with distilled water. The test parameters - normal load, sliding distance, numbers of cycles for each test are given in Appendix A. Tables 3.1 and 3.2 list respectively the chemical compositions and the mechanical properties of several materials of interest.

In all tests, the tube material is Incoloy 800, the counterface materials are brass C360000 or carbon steel 12L14 or stainless steel 410. The other material listed in Table 3.1 and Table 3.2 will be discussed in Chapter 4. In earlier tests, alloy steel ball AISI 52100 (hardened to 64-66 Rc, ground and polished) were used in place of the Incoloy 800 tube.

Table 3.1 Chemical Compositions of Materials

<table>
<thead>
<tr>
<th>Material</th>
<th>Fe</th>
<th>Ni</th>
<th>Cr</th>
<th>C</th>
<th>Al</th>
<th>Cu</th>
<th>Pb</th>
<th>Ti</th>
<th>S</th>
<th>Mn</th>
<th>Si</th>
<th>P</th>
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<td>C36000</td>
<td>0.35 max</td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>2.5-3.7</td>
</tr>
<tr>
<td>12L14</td>
<td>rest</td>
<td>0.15 max</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.85-1.15</td>
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<tr>
<td>A36</td>
<td>rest</td>
<td>0.25-0.29</td>
<td></td>
<td>0.2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.05 max</td>
<td>0.6-0.9</td>
<td></td>
<td>0.04 max</td>
</tr>
<tr>
<td>410S.S</td>
<td>rest</td>
<td>11.5-13.5</td>
<td>0.15 max</td>
<td>0.2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.03 max</td>
<td>1.0 max</td>
<td>1.0 max</td>
</tr>
<tr>
<td>304S.S</td>
<td>rest</td>
<td>8-10.5</td>
<td>18-20</td>
<td>0.08</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>0.03 max</td>
<td>2.0 max</td>
<td></td>
<td>1.0 max</td>
</tr>
<tr>
<td>Incoloy 800</td>
<td>39.5 min, 30-35</td>
<td>19-23</td>
<td>0.1 max</td>
<td>0.15-0.6</td>
<td>0.75 max</td>
<td>0.15-0.6</td>
<td>0.015 max</td>
<td>1.5 max</td>
<td>1.0 max</td>
<td></td>
<td></td>
<td></td>
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Table 3.2 Mechanical Properties of Materials

<table>
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<tr>
<th>Material</th>
<th>$\sigma_T$ MPa</th>
<th>$\sigma_Y$ MPa</th>
<th>Elongation %</th>
<th>Poisson Ratio</th>
<th>Reduction in area</th>
<th>Hardness</th>
<th>$T_e$ MPa</th>
<th>Fatigue strength</th>
<th>$\rho$ g/cm$^3$</th>
<th>E GPa</th>
</tr>
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<tbody>
<tr>
<td>C36000</td>
<td>400</td>
<td>310</td>
<td>25</td>
<td>50</td>
<td>136 Hv</td>
<td>235</td>
<td>140 MPa</td>
<td>8.5</td>
<td>97</td>
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<tr>
<td>12L14</td>
<td>540</td>
<td>415</td>
<td>10</td>
<td>35</td>
<td>200 Hv</td>
<td></td>
<td></td>
<td></td>
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<td></td>
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<tr>
<td>A36</td>
<td>400-550</td>
<td>250</td>
<td>20</td>
<td>35</td>
<td>200 Hv</td>
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<td></td>
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<tr>
<td>410S.S</td>
<td>1525</td>
<td>1225</td>
<td>14.5</td>
<td>63.5</td>
<td>315 Hv</td>
<td>7.8</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>304S.S</td>
<td>690</td>
<td>415</td>
<td>60</td>
<td>70</td>
<td>212 HRB</td>
<td>8.0</td>
<td>193</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Incoloy 800</td>
<td>517-690</td>
<td>207-414</td>
<td>50-30</td>
<td>0.339</td>
<td>220 Hv</td>
<td>228 MPa</td>
<td>7.94</td>
<td>195</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inconel 600</td>
<td>552-690</td>
<td>172-345</td>
<td>55-35</td>
<td>0.29</td>
<td>83 HRB</td>
<td>310 MPa</td>
<td>8.42</td>
<td>207</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
3.4 Test Results

3.4.1 Preliminary Test Results

These tests were carried out with an alloy steel ball sliding on a flat brass disc with cutting fluid lubricant at room temperature. These tests were an extension of previous studies [17]. It also served to familiarize the author with the testing techniques.

Some of the early tests did not have properly recorded data. These results are therefore somewhat confusing, and some of them do not repeat well (see Appendix A, Table A.1). Figure 3.7 also shows a rather random relationship between the total frictional work input and wear mass loss. Since material from the brass disc has been found to have transferred to the alloy sphere, the mass change of the sphere was not considered.

The results in Figure 3.8 show that under similar load and sliding distance wear increases as the dilution of the cutting fluid lubricant is increased. It is suggested that better lubricant will reduce the frictional force and reduce wear damage for the same travelled sliding distance and normal force.

![Figure 3.7 Total Frictional Work vs Wear Loss (Early Tests)](image-url)
3.4.2 The Line Contact Tests Between Incoloy Tubing and Brass Flat

The line contact tests between a tube and brass flat were originally designed to investigate plasticity deformation under pure sliding conditions. The brass specimen is a disc with a convex strip (Fig. 3.9). The tests were arranged into five groups, each group related to a certain lubricant condition. And the results are listed in Table A.2.

Figure 3.8 The Relationship of Lubricant and Wear Loss

Figure 3.10 The Shape of Brass Specimen For Tube Test
Chapter 3. Experimental Tests

Group 1 involved 7 tests at 100 N normal force and 2 mm sliding amplitude. All the tests were lubricated with the same dilution of cutting fluid. The purpose is to investigate the relationship between sliding distance and the amount of wear.

Group 2 involved 4 tests at 100 N normal load and 1 mm sliding amplitude. The same lubricated condition as group 1 was used. The purpose is to study the effect of sliding amplitude on wear (compared with the group 1).

Figure 3.10 shows that the two groups of results almost coincide and that the sliding distance vs wear relationship is linear.

Group 3 included 9 tests each subjected to 300,000 sliding cycles and 2mm sliding amplitude. This group of tests is mainly for investigating the normal load and wear loss relationship. Since the friction coefficient has a strong effect, the results have been averaged. Figure 3.11 shows a linear relationship between wear loss and normal load.

Group 4 and group 5 are additional results. In the tests of group 1 to 4, there is misalignment between the contact surfaces. After modification group 5 tests achieved almost perfect line contact.

The preceding results provide some fundamental relationship between wear and some wear parameters. But it is important to note that these tests are subjected to the same lubrication condition.

Figure 3.12 shows an important relationship between wear and the total frictional work. It includes all of the "Incoloy tube-brass disc" tests. The linear relationship is an important result for pure sliding wear. An interesting finding is that the misalignment problem did not affect the linear relationship between total frictional work and wear. It appears that the higher load side had deeper wear, and the lower load side had shallower wear, the effect of misalignment was averaged out.
Figure 3.10 Travelled Sliding Distance vs Wear Loss

Figure 3.11 Normal Load vs Wear Loss
3.4.3 Line Contact Between Incoloy Tube and Stainless Steel Disc

In order to investigate tube wear performance, a series of tests has been carried out with Incoloy 800 tube sliding on stainless steel 410 disc with distilled water as the lubricant. The results are listed in Appendix A Table A.3.

Since the disc material is relatively hard and the lubrication poor, the tube wear rate and friction coefficient are much higher. For better control, lower normal loads have been applied (20 N, 50 N, 80 N).

Figure 3.13 shows the relationship between frictional work input and wear mass loss for both the stainless steel disc and tube. The tubes have higher wear than the stainless discs, and the relationship is close to linear.
3.4.4 Line Contact Test Between Incoloy Tube and Carbon Steel Flat

In many CANDU steam generators, Incoloy 800 tubing and carbon steel support structures are implemented. A simplified configuration of an Incoloy tube sliding on a flat carbon steel disc was used in this investigation. Two different lubricant conditions have been applied, the results are listed in Appendix A, Table A.4 and A.5.

The tests with cutting fluid lubricant involved carbon steel discs with two different surface finishes. One set was ground to 1000 grit SiC paper finish and the other set to a 6 micron diamond finish. The relationship between mass loss and the total frictional work input is fairly linear (see Fig. 3.14). The unpolished condition appears to have lower mass loss as a group because the preload is lower (average 4.5 N compare to the polished group 5 N in preload). In these tests, the tube become noticeably worn after hours of testing and the wear amount is easily measured. Figure 3.15 shows the approximate linear relationship between tube wear loss and frictional work input.
Chapter 3. Experimental Tests

Figure 3.14 Total Friction Work vs Wear Loss
(Carbon Steel, Cutting Fluid)

Figure 3.15 Total Frictional Work vs Wear Loss
(Tube, Cutting Fluid)
The tests with distilled water lubricant are only for the polished carbon steel discs. In this case, the tube wear rate is lower compared to the stainless steel disc cases. The relationship between the frictional work input and tubing and carbon steel disc mass loss is quite linear (see Fig. 3.16).

From the above study, a conclusion can be drawn: for all the material combinations tested in this program, wear has a linear relationship with the frictional work.
Surface analysis is a very important component in tribology research. The wear process, because of its complexity and small scale, is very difficult to observe. Analyzing the specimen surface after testing is one way to better understand the actual wear mechanisms. There are many surface analysis techniques that include topographical analysis, microstructure analysis, and chemical composition analysis. With advanced modern instruments, more and more evidence has been found for explaining the wear mechanisms, and helping to guide new research initiatives.

This chapter describes some observed surface phenomena from the sphere/disc test samples, tube/disc test samples, and some of the AECL high temperature (265°C) worn tube samples. In this study, Stylus Profilometer analysis, Scanning Electron Microscopy (SEM) analysis, Energy Dispersion X-ray (EDX) analysis, Electron Micro-Probe analysis and X-ray Photoelectron Spectroscopy (XPS) analysis have been applied.

4.1 Material Transmission

In the tests of a steel sphere against a brass disc or Incoloy 800 tubing against a brass flat, certain amounts of brass were transferred to the contact surfaces of the sphere and tube. By visual examination, it was observed that the worn sphere or tube surface had turned to a brassy colour. EDX analysis confirmed that the worn sphere surface had elements of brass coating (Fig. 4.1). Examination by SEM also showed that brass material adhered to the worn sphere surface (Fig. 4.2). One explanation is that the softer brass wear particles were trapped between the contact surfaces, and eventually forming an adhered layer on the spherical specimen surface after repeated passes.
Chapter 4. Surface Analysis

Figure 4.1 EDX Analysis of Material Transmission (ball #114)

Figure 4.2 SEM Analysis of Material Transmission (ball #114)
4.2 Plastic Deformation

Figure 4.3 is a typical brass disc wear scar map after rubbing against a sphere. Along the sliding direction, brass material has been pushed forward to the front tip. On the worn surface, plastic flow can be observed by SEM as shown in Figure 4.4.

Figure 4.5 shows the cross-section of an unworn area of a brass disc sample. After test, the specimen was sectioned, polished (0.05 μm aluminum finishing) and etched to show the undeformed grains. Figure 4.6 is from the worn area, where the sample is sectioned along the sliding direction. The microstructure of the material has been highly deformed and strained in the sliding direction, especially near the surface area. It resembles that displayed in Rice's morphological analysis [41]. He suggests a zone structure for describing the deformed grain structure under repetitive load as illustrated in the schematic diagram of Figure 4.7. Zone 1, furthest from the contact region, consists of undisturbed base material. Zone 2, the intermediate region, is plastically deformed. Zone 3 is usually homogeneous and very finely structured.

At the edge of the worn surface (zone 3), grain boundaries can not be observed any more, instead, a very fine layer has been formed (Fig. 4.8). In Figure 4.8, some cracks can be seen, these cracks have been blunted by the soft nature of the material. In wear processes, fracture problems do appear, but the crack behaviour is complicated by the changing material properties and cyclic forces.

Using Hertzian contact equations for a sphere contact, the octahedral shear stress can be determined. Figure 4.9 shows, when μ = 0.164, the maximum shear stress is about 0.34 P₀ located at a depth about z = 0.5 a. The related coordinate is shown in Figure 4.10.

Another interesting investigation involved sectioning the worn area in the transverse direction, i.e. perpendicular to the sliding direction. Figure 4.11 shows the deformed grain structure. Figure 4.11a and 4.11c are the two edges of the contact zone, 4.11b is the contact centre. Near the centre of the surface, the grains have been highly compressed, from round or long shape to flat shape (Fig. 4.12).

The reason why the micro-structure has this kind of change may be explained by the related stress analysis. Figure 4.13 shows the shear stress field τᵧz which is on the plane that is
perpendicular to the sliding direction (i.e. on the Y-Z plane). Figure 4.14 shows the normal stress field of \( \sigma_z \) and Figure 4.15 shows the stress \( \sigma_y \) on the Y-Z plane. These stress contours show that \( \sigma_z \) is the dominant stress which compress the material greatly at the contact centre. Off the contact centre, \( \tau_{yz} \) played a more important role.

sliding direction

![Diagram](image)

Figure 4.3 Brass Disc Wear Scar Map
Figure 4.4  Plastic Flow on Worn Brass Disc
Chapter 4. Surface Analysis

Figure 4.5  Original Grain Boundaries of The Brass Disc

Figure 4.6  The Deformed Grain Boundaries
Figure 4.7  3-Zone Schematic Diagram

Figure 4.8  Zone 3 and Cracks
Figure 4.9  Contour Plot of $\frac{\tau_{\infty}}{P_0}$ for $\mu=0.164$ (brass disc #114)

Figure 4.10  The Coordinate Schematic Diagram
Figure 4.11 The Microstructure of Y-Z Plane (for brass disc #114, perpendicular to the sliding direction)
Figure 4.12  Grain Deformation Near The Contact Centre

Figure 4.13  Contour Plot of $\tau_{xz}/P_0$ for $\mu=0.164$ (brass disc #114)
Figure 4.14  Contour Plot of $\sigma_y/P_0$ for $\mu=0.164$ (brass disc #114)

Figure 4.15  Contour Plot of $\sigma_y/P_0$ for $\mu=0.164$ (brass disc #114)
4.3 Surface Profile vs Number of Sliding Cycles

For the Incoloy 800 tube/brass bar tests mentioned in the Chapter 3, group 1 has revealed the relationship between wear mass loss and sliding distance (see Appendix A, Table A.2). A related series of photomicrographs clearly illustrates the surface condition due to repeated sliding with a 100 N normal load.

Figure 4.16 shows the profile of the original ground surface. After 9900 cycles of sliding, the surface has become smooth (Fig. 4.17).

Figures 4.17 to 4.22 show the surface conditions under different sliding cycles. A surface roughness measurement has been performed for the group 1 samples by stylus profilometry. The results are listed in Table 4.1. There is no obvious relationship between surface roughness and the number of sliding cycles.

Figure 4.16  Original Ground Surface
Figure 4.17  Surface After 9900 Cycles (brass bar #1, 100 N)

Figure 4.18  Surface After 30,000 Cycles (brass Bar #2, 100 N)
Figure 4.19 Surface After 60,000 Cycles (brass bar #3, 100 N)

Figure 4.20 Surface After 150,000 Cycles (brass bar #4, 100 N)
Figure 4.21  Surface After 300,000 Cycles (brass bar #10, 100 N)

Figure 4.22  Surface After 450,000 Cycles (brass bar #6, 100 N)
### Table 4.1  Surface Parameters For Group 1 Brass Samples

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<tr>
<th>NUM</th>
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<th>Ra</th>
<th>Skewness</th>
<th>Kurtosis</th>
<th>Sigma</th>
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<td>0.3007</td>
<td>0.8353</td>
<td>3.8973</td>
<td>0.395</td>
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<tr>
<td>2</td>
<td>30000</td>
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<td>4</td>
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<td>0.2886</td>
<td>0.793</td>
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### Table 4.2  Surface Parameters For Group 3 Brass Samples

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<th>Skewness</th>
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<th>Sigma</th>
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<td>0.9301</td>
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<td>5</td>
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<td>7</td>
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<td>9</td>
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<td>0.2802</td>
<td>0.5811</td>
<td>3.1835</td>
<td>0.3583</td>
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</table>
4.4 Surface Profile vs Normal Load

The relationship between surface profile and normal load has been studied with the aid of SEM. The samples are from the Incoloy 800 tube and brass flat test group 3 (see Table A.2). Table 4.2 lists surface conditions measured by stulus profilometry for group 3 samples. Figures 4.23-4.27 and the data from Table 4.2 show no obvious relationship between normal load and surface condition.

Figure 4.23  Surface Under 50 N (brass bar #5, 300,000 cycles)
Figure 4.24  Surface Under 100 N (brass bar #11, 300,000 cycles)

Figure 4.25  Surface Under 200 N (brass bar #7, 300,000 cycles)
Figure 4.26  Surface Under 300 N (brass bar #8, 300,000 cycles)

Figure 4.27  Surface Under 400 N (brass bar #9, 300,000 cycles)
4.5 AECL High Temperature Worn Tube Analysis

Ko studied tube fretting wear in high temperature (265°C) water using a CRNL high temperature autoclave rig (Fig. 2.2) and found that the wear rate of the two different tube/tube support material combinations can differ by a factor of ten, even though their vibration levels are the same [8].

With curiosity, this author studied two tube samples that were from Ko's previous high temperature tests. One was a piece of Incoloy 800 tube that was tested for 120 hours with a carbon steel A36 ring in water at 265°C and 900 psi. The other one was a piece of Incoloy 800 tube that was tested for 30 hours with an Inconel 600 ring under similar temperature and pressure. The former had little wear and the tube surface appeared to be very smooth (Ra=0.3786 m, longitudinal), the latter had severe wear and the surface was very rough (Ra=1.9139 m, longitudinal). The sectioned surfaces showed that the tube tested with the carbon steel A36 ring had no apparent microstructural disturbance near the worn surface (Fig. 4.28), but the tube tested with the Inconel 600 ring showed significant microstructure deformation near the worn surface (Fig. 4.29).

With the help of Dr. Matti Raudsepp at Department of Geological Science UBC, the chemical composition of a tube (tested with carbon steel ring) cross section was investigated with a Cameca 5x50 Electron Microprobe. Figures 4.30, 4.31, 4.32 and 4.33 are Ni, Cr, Fe and Mn, Al, Ti spectra respectively. The results show that for the tube/carbon steel combination, there is no obvious chemical composition change across the section in this case, tube wear only happened at the very top surface.

Usually the colour of the tube surface alters during high temperature tests. It is interesting to note that different tube/ring material combinations resulted in different tube surface colours. A XPS analysis has been done at the Department of Chemistry of UBC with help from Dr. Philip Wong. It was found that the worn tube surfaces were fully covered by oxide and the material element composition (Fe, Cr, Ni mainly) has been significantly changed. This result coincides with Sture Hogmark's AES analysis [13] (mentioned in chapter 2). Since the tube samples are relatively old and not very well preserved, further investigation would not be fruitful. But two
things are worth noting. First, at high temperature, heat exchanger tube wear has a stronger oxidation than at room temperature; second, further investigation of the kind of oxide formed at the tube surface for each tube/ring material combination may lead to the understanding of why the wear rates are different. This information may reveal more about the nature of the heat exchanger tube wear problem.

Figure 4.28  Tube Section (tested with carbon steel A36 ring)
Figure 4.29  Tube Section (tested with Inconel 600 ring)
Figure 4.30 Tube Cross Section Ni Composition Spectrum
Figure 4.31  Tube Cross Section Cr Composition Spectrum
Figure 4.32 Tube Cross Section Fe Composition Spectrum

SX - Spectra Acquisition and Plotting
15-keV 40-nA IN5 S15 Linear traverse spectrum
Date: 16-MAY-92

Length: 58.80 Microns
Step: 1.02 Microns
Counting time: 380 ms

First point coordinates: 6985 26652 26692 6985
Last point coordinates: 6985 26652 26692 6985

Counting time: 380 ms
Figure 4.33 Tube Cross Section Mn, Al, Ti Composition spectra
Chapter 5

A Model For The Heat Exchanger Tube Wear

In spite of several decades of efforts, wear modelling remains one of the most challenging areas in tribology. A complete fundamental principle is not yet available. However, there is a great demand for wear equations in engineering practice.

For wear modelling, it is important to determine which of the many parameters have the greatest effects on the wear system. Compromise has to be made at this stage. As a result, some simple empirical models can be available for some special wear problems.

In this chapter, some early models are analyzed for heat exchanger tube wear applications. Based on the heat exchanger tube wear phenomena, test results and previous studies, a model has been proposed and discussed. The calculated results are well matched to the experimental results. In order to make this model more suitable for the engineering application, an extension to the model has been made so that the tube life can be estimated and the tube/ring misalignment problem can be considered.

5.1 Some Early Analytical Models

Many wear models have been published, usually, for specific conditions. With due respect to the previous studies, some of these models have been applied to the heat exchanger tube wear.

5.1.1 Archard Model Analysis

As we mentioned in chapter 2, the Archard model is the most widely accepted adhesive wear model. It is in a very simple form:
$V = k \frac{P}{H} L$

where wear volume $V$ is proportional to the normal load $P$ and the travelled sliding distance $L$, inversely proportional to the softer material hardness $H$, $k$ is a dimensionless constant which could be in a very wide range (from $10^{-1}$ to $10^{-10}$).

In the heat exchanger tube/brass disc experimental tests, the wear amount is not always proportional to the normal load. However, if the friction coefficient is a constant, wear volume does have the linear relationship with normal load. During the rubbing process, the friction coefficient varies (see Fig. 3.6), which may be caused by the change of wear surface topography, wear particles trapped between the contact surfaces, and the change in the lubricant film thickness.

5.1.2 Jain and Bahadur Model Analysis

The Jain and Bahadur wear model [30] was proposed in 1980 for polymer wear. The key point is to assume that the fatigue failure of asperity interactions cause wear. They found that the computed steady state wear rates are in excellent agreement with the experimental results for the three polymers tested [31]. It is worthwhile to test this model for its suitability for the heat exchanger tube wear problem.

In chapter 2, the basic concepts of the model have been explained. In order to calculate the wear amount for brass/tube tests, more detailed assumptions are added to the original model:

1). Asperity heights have a Guassian distribution
2). Both surfaces have a similar asperity distribution i.e.

$$\eta_1 = \eta_2; \ \eta_{L1} = \eta_{L2}; \ \beta_1 = \beta_2; \ \sigma_1 = \sigma_2$$

(where $\eta$ is asperity density per area, $\eta_{L}$ is linear asperity density, $\beta$ is asperity radius, $\sigma$ is
Chapter 5. A Model For The Heat Exchanger Tube Wear

asperity standard deviation, the subscripts 1 and 2 refer to the two surfaces in contact). This assumption can be explained by the conformity of the two contact surfaces.

3). The distance between the reference plane of the surfaces in contact (Fig. 2.10) is assumed to be the mean asperity height, i.e. \( d = R_a \).

4). The failure stress corresponding to the application of a single cycle is assumed to be the true fracture strength for necking [40], i.e.

\[
S_0 = \frac{\sigma_0}{(1 - \text{Area Reduction } \%)}
\]

Detailed wear calculation for brass/tube specimen #1 is listed in the Appendix B. The results show a significant difference between the theoretical prediction and the experimental measurement.

Many factors might be responsible for this difference. The main reason could be that metal behaves very differently from a polymer. For example, metal wear profiles are very different from that of the polymers (compare \( \eta \) and \( \eta_n \) of the two cases). In the model, the bulk material fatigue property has been used to judge the failure of asperities. But for metal, the contact surface material property is very different from the bulk property. On the other hand, the measurement of the surface parameters has become a significant problem, for instance, the same specimen can have an asperity radius 174.8 \( \mu \)m at 5 \( \mu \)m per sampling step, and a radius of 627 \( \mu \)m at 10 \( \mu \)m per sampling step. This kind of problem has been pointed out by P. Gupta [47] in his discussion of surface topography in 1978.

5.1.3 Magel Model Analysis

In this model, Magel applied Herzian line contact theory, shakedown theory and fracture mechanics to predict wear for a sphere sliding against a flat surface.

Herzian line contact theory provides a good opportunity to describe the pressure distribution and stress fields within elastic range. Shakedown theory allows the application of elastic theory under small amount of plastic deformation. By combining the shakedown and
Herzian theory, Magel can successfully predict the wear geometry of the indentation created by a sphere on a flat. But the prediction of the heat exchanger tube wear is claimed to be significantly larger than the experimental results.

The expression for predicting the amount of wear is given by equation 2.30, i.e.

\[ V = \frac{W}{F} v_p \]

This is a straightforward equation, the relationship among the parameters being quite obvious.

The difficult part is to get an accurate particle size \( v_p \) and the energy to generate the particle \( E_p \).

In order to remove the obstacle, Magel made the following assumptions:

1). cracks initiate and propagate at the depth \( Z_e \) of the maximum octahedral shear stress
2) the energy input \( E_p \) required to form a wear particle of size \( A_p \) and thickness \( Z_h \) is equal to \( \tau_y A_p L_i \)

Magel calculated \( A_p \) as the scar area and \( Z_h \) by the location of the maximum octahedral shear stress. He assumed that applying energy \( E_p = \tau_y A_p L_i \) can generate a wear amount \( v_p = A_p Z_h \). The evaluation of the wear amount is significantly higher than the experimental results.

By looking into each step of the equations, the \( E_p/V_p \) ratio has been found to be very far from the testing evaluation.

In reality, to generate observable wear, usually thousands of cycles are needed under mild normal load with lubricant. Before fracture happens, a large amount of energy goes into generating heat and plastic deformation. Therefore, a relatively small amount of shear energy \( E_p = \tau_y A_p L_i \) can not generate wear volume \( v_p = A_p Z_h \). H.Vetz and J.Föhl discussed this phenomena in their early paper [43]. They described the energy absorbing process on a pair of metallic materials (Figure 5.1). They point out "The main energy term, at least for metallic materials, comes from deformation. The fracture energy is estimated to amount to only a few percent of the total absorbed energy". This may explain why the calculated tube wear is much higher than the real measurement.
Chapter 5. A Model For The Heat Exchanger Tube Wear

5.1.4 Lin and Cheng Model Analysis

The Lin and Cheng model is a dynamic wear model which permits the wear rate to vary with time. Therefore, it is able to explain the commonly observed running-in, steady-state, or accelerated wear phenomena.

The model can be expressed as

$$ W = c \frac{FL}{U} $$
where \( W \) is the wear amount, \( L \) is the sliding distance, \( F \) is the forcing term which represents the wear causing agent, \( U \) is an antiwear strength term that represents the wear resisting agent, \( c \) is a nondimensional constant.

The antiwear strength \( U \) is defined as the average material strength within a layer of thickness \( h \) near the surface. For some simple cases, Lin and Cheng suggested that \( U \) can be related to the measurable material property such as the hardness or the yield strength (see equation 2.23). The average shear force is defined as the shear force averaged over a thin layer of thickness. Both \( F \) and \( U \) are time and wear dependent.

Since this model includes time dependent parameters such as the shear stress \( \tau(x,y,z) \) and flow strength \( \delta(x,y,z) \), for asperity involved conditions, the calculation could be rather complicated, especially when plastic deformation is also considered.

This model has not been directly test proved. But by comparing the model calculation to some experimental results obtained by others, the authors claimed that the model could give a rather good wear prediction.

### 5.2 Wear Phenomena and Test Results Analysis

In spite of the complexity of the wear problem, some wear phenomena have been generally recognized and classified. The experimental results of this study have revealed several relationships among the wear parameters which provided the fundamental criteria of heat exchanger tube wear.

#### 5.2.1 Wear Volume and Sliding Distance

"The longer the component used, the larger the wear." is a common wear phenomenon. This phenomenon has been classified by Hirst and Lancaster [49]. Figure 5.2 shows several possible relationships between wear and time. Figure 5.3 is a typical wear-time curve which is generally recognized. There are three wear stages shown in Figure 5.3. Stage one is the running-in regime, in which wear and time relationship is nonlinear, usually it is the wear of type I, of
which the wear rate is initially high but later decreases to a lower value. Stage two is the steady state wear process. In stage two, the wear-time function is linear, i.e. type II of which the wear rate is a constant. Wear stage three is the accelerated wear regime, a catastrophic surface damage may occur in a short time because wear increases beyond a critical wear value, usually it is the wear of type III, in which wear rate linearly increases.

A curve that has a similar pattern to Figure 5.3 can be drawn from data presented in Figure 5.4 under the condition of constant load. Otherwise, the results can be in a random pattern as shown in Figure 5.4.

5.2.2 The Load and Wear Relationship

In wear studies, load has been considered an important wear causing agent. For many wear cases, wear is proportional to the applied normal load. This has been described by the Archard model. For the heat exchanger tube wear tests, it is true only under a constant frictional coefficient conditions. Further data analysis shows, that under the same sliding distance and environmental conditions, the tangential load and wear relationship is linear (see Figure 5.5).
Chapter 5. A Model For The Heat Exchanger Tube Wear

Figure 5.3 Three Regions of Wear

Figure 5.4 Brass Discs Against Incoloy 800 Tubes Test Results
5.2.3 Wear State

As Figure 5.3 shows, the wear process experiences three stages. Running-in is an initial break-in period in which wear behaves differently from that during the steady state. At this break-in period, the wear system is in an adjusting stage, many variables such as surface topography, lubricant characteristics, near surface temperature, micro structure, and hardness change with time. It has been reported [50] that running-in will depend strongly on the initial material structure and properties and on the surface conditions such as the surface finish and the nature of any film present. During this transition period, films may be destroyed or modified, the material may work-harden, phase transformations may occur, temperature will rise as a result of frictional heating, energy will be stored and the micro structure and surface topography will evolve toward a steady-state condition. All these changes affect the frictional coefficient.
Chapter 5. A Model For The Heat Exchanger Tube Wear

At steady state, wear variables will be relatively constant with time. Compared to the running-in state, steady state last much longer. If one waits long enough, and if the design allows the achievement of steady-state conditions, any initial surface finish should evolve to the surface finish appropriate to steady state for the given system. The choice of initial surface finish may affect the time needed to achieve steady-state, but the same steady state should be reached eventually.

The third wear state is the accelerating stage. At this stage, wear failure is indicated.

From the heat exchanger tube experimental observation, the steady state is the dominant state. Running-in is not obvious (Figure 3.15-3.18). The third stage, i.e. the accelerate state, is not considered here, because the application must be stopped at this stage to avoid severe damage to the heat exchanger tubes.

5.2.4 Hardness

In wear study, hardness is an important material property. For many cases, it is claimed that wear is inversely proportional to the softer material hardness, such as in the Archard equation. In other literature, it has been found that wear hardness does not inversely proportional to the material hardness [51].

For the heat exchanger tube experimental tests, under similar testing conditions, higher hardness materials experienced less wear. However, tests with carbon steel 12L14 disc are the exceptions, since oxidation problems become another significant factor on wear.

5.2.5 Roughness

The effect of surface roughness on wear has been recognized for decades. However, mathematically describing the way it affects wear is still difficult, because different film thickness, material, and speed have different interactions and effects.

The majority part of the wear state for heat exchanger tube wear is steady state, in steady state, the original surface condition does not affect wear (Fig. 3.17). But during the running-in
stage, the original surface finishing has a strong influence on wear. This is also observed in Knowles work [52].

Table 5.1 shows some topography results from the heat exchanger tube test. It has been found that under similar grinding processes, harder material exhibit finer surface finishing, and no matter what kind of material the disc is, no matter what the original surface roughness the disc has, the Ra and σ values of the disc and tube after wear are much the same. It indicates that the contact surfaces finally reached conformity, i.e. the test reached the steady state.

The surface roughness Ra and standard deviation σ are relatively stable parameters compared to other surface topographic parameters such as asperity radii.

<table>
<thead>
<tr>
<th>Table 5.1 Topographic Parameters</th>
</tr>
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<tbody>
<tr>
<td></td>
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<tr>
<td></td>
</tr>
<tr>
<td>Tube</td>
</tr>
<tr>
<td>SS410 (S3) Tube</td>
</tr>
<tr>
<td>12L14 (P8*) Unpolished Tube</td>
</tr>
<tr>
<td>12L14 (P3) Polished &amp; cutting fluid Tube</td>
</tr>
<tr>
<td>Tube</td>
</tr>
<tr>
<td>12L14 (P8) Polished &amp; Water Tube</td>
</tr>
<tr>
<td>Brass Disc (B4)</td>
</tr>
</tbody>
</table>
5.3 Modelling

5.3.1 Criteria

From the previous discussion, it is clear that the following parameters need to be considered for heat exchanger tube wear:

- wear is proportional to the sliding distance \( L \)
- wear is proportional to the frictional force \( F \)
- wear may be inversely proportional to the material hardness \( H \)
- the running-in stage is related to the topography
- at the steady stage, wear variables will be relatively constant with time

Therefore the wear equation can be written as

\[
V = c \frac{F \cdot L}{U} = c \frac{E}{U} \quad (5.1)
\]

where \( E = F \cdot L \).

This is the Lin and Cheng wear model.

5.3.2 Discussion

In equation 5.1, each term has a certain physical meaning.

The parameter \( F \) is the frictional force, \( L \) the accumulated unit direction sliding distance (the backward movement is load free). Reciprocating sliding wear is not considered here. In equation 5.1, both \( F \) and \( L \) are the external wear causing agents and both are measurable parameters. \( E \) may be viewed as the external energy input to generate wear.

The parameter \( c \) is a non-dimensional wear coefficient which contains plenty of undetermined information such as lubricant film thickness (external factor), material microstructures (internal factors) and the surface temperature, oxidation (both external and internal factors). As previously discussed, at the running-in stage, these parameters are time dependent.
Therefore, strictly, at the initial stage, c should be a time dependent function instead of a constant. At the steady wear stage, these parameters are independent of time, thus c will be a constant. Latter, the parameter c may be separated into many parameters or functions such as \( c = f_1 \cdot f_2 \cdot f_3 \cdot \ldots \cdot f_n \).

The antiwear strength U is a wear resisting term. It represents the energy required to generate a unit wear volume, similar to the term \( E_p/V_p \) in Magel’s model (i.e. equation 2.30). U can also be called a specific wear energy. Jahanmir [53] pointed out "the specific wear energy is composed of surface energy, cutting energy and plowing energy. The contribution of each component to the specific wear depends on the wear mechanism, but in most cases the plastic strain energy for subsurface deformation predominates". For the whole wear process, U is time and wear dependent. At the running-in state, the wear system is in an adjusting stage, wear parameters vary with time, so that U changes until the two surfaces reach conformity. At the accelerating state, the component is close to failure, the antiwear strength U decreases drastically. Some papers have considered that U is a function of material properties [53].

5.3.3 Equations

Equation 5.1 simply describe the relationship of the wear causing agent and the wear resisting agent. More detailed calculation is based on further assumptions.

5.3.3.1 Assumptions

1. The material is simplified to be homogeneous and time independent
2. At the running-in stage, the surface topography is the main factor to cause nonlinear wear behaviour. In other words, the wear coefficient c is considered a constant.
3. The asperity distribution is exponential i.e. the asperity height probability function is

\[ \phi (z) = \lambda \exp (-\lambda z) \]
4. The asperity distribution at time \( t_i \) and \( t_{i+1} \) is assumed to be the same, i.e.

\[
\phi_i(z) = \phi_{i+1}(z)
\]

5. The antiwear strength is defined as

\[
U = \frac{H}{h} \int_{z_i}^{z_i + h} \Phi(z) \, dz
\]

In these assumptions, \( z_i \) is the wear depth at time \( t_i \), \( h \) is the thickness of the surface layer that has been deformed elastically and plastically, \( H \) is the material hardness, \( \phi(z) \) is the asperity height distribution, \( \Phi(z) \) is the cumulative probability distribution.

### 5.3.3.2 Derivation

Combining equation 5.1 and the assumptions, at \( t_i \), the following equations can be established:

\[
V_i = c \frac{E_i}{U_i} \quad (5.2)
\]

\[
V_i = A \int_{z_i}^{z_i + h} \Phi(z) \, dz \quad (5.3)
\]

\[
E_i = F_i \Delta L_i \quad (5.4)
\]

\[
U_i = \frac{H}{h} \int_{z_i}^{z_i + h} \Phi(z) \, dz \quad (5.5)
\]
where $A$ is the nominal contacting area.

If the asperity height probability distribution is exponential, then the cumulative probability function is

$$\Phi (z) = \int_0^z \phi (z) \, dz = 1 - \exp \left( -\lambda z \right) \quad (5.7)$$

From the definition of the exponential distribution, $\lambda$ is related to the root mean square $\sigma$, i.e. $\lambda = 1/\sigma$.

Substituting the equations 5.3, 5.4, 5.5 into equation 5.2, it arrives at

$$A \int_{z_l}^{z_{r+1}} \Phi (z) \, dz = \frac{cF_i \Delta z_i}{H} \int_{z_l}^{z_{r+1}} \Phi (z) \, dz$$

$$\int_{z_l}^{z_{r+1}} \Phi (z) \, dz = K h_i \frac{1}{\int_{z_l}^{z_{r+1}} \Phi (z) \, dz} \quad (5.3)$$

where

$$K = \frac{cF_i \Delta L_i}{HA}$$

Substitute equation 5.7 into equation 5.8 and integrate, the following equations can be obtained:
Chapter 5. A Model For The Heat Exchanger Tube Wear

\[
(z_{i+1} - z_i) + \frac{1}{\lambda} (\exp(-\lambda z_{i+1}) - \exp(-\lambda z_i))
\]

\[
= Kh/((h+\exp(-\lambda z_i))(\exp(-\lambda h) - 1)/\lambda)) \tag{5.9}
\]

let

\[
z_{i+1} = z_i + \Delta z_i
\]

then

\[
\exp(-\lambda (z_i + \Delta z_i))
\]

\[
= \exp(-\lambda z_i) - \lambda \exp(-\lambda z_i) \Delta z_i \tag{5.10}
\]

Substituting equation 5.10 into equation 5.9, then

\[
\Delta z_i = \frac{Kh}{(1-\exp(-\lambda z_i))(h+\frac{1}{\lambda} \exp(-\lambda z_i)(\exp(-\lambda h) - 1))} \tag{5.11}
\]

The antiwear strength can be written as

\[
U_i = \frac{H}{h} \int_{z_i}^{z_i+h} (1-\exp(-\lambda z)) \, dz
\]

\[
= \frac{H}{h} (h+\frac{1}{\lambda} \exp(-\lambda z_i)(\exp(-\lambda h) - 1)) \tag{5.12}
\]
5.4 Calculations

5.4.1 Data Input

Table 5.2 listed the original data input of the calculation for all the testing conditions.

In this table, $\lambda$ is the inverse of $\sigma$. For two rough surface in contact, $\sigma=\left(\sigma_1^2+\sigma_2^2\right)^{1/2}$, then $\lambda=1/\left(\sigma_1^2+\sigma_2^2\right)^{1/2}$. Since the running-in stage is related to the initial surface topography, here, the unworn surface parameters $\sigma_i$ ($i=1,2$) is used. $H$ is the material hardness, $A$ is the nominal wear area, $c$ is an empirical wear coefficient, and $h$ is the deformed depth of the material which can be obtained by the observation of the cross section using SEM. In equation 5.11, $z_0=0$ is a singularity point, this problem comes from the original assumption of the exponential distribution of asperity height. For Gaussian distribution, this problem can be avoided, but the calculation for $\Delta z_i$ is more complicated, detailed derivation is explained in Appendix C.
Table 5.2 Initial Input Data

<table>
<thead>
<tr>
<th></th>
<th>λ</th>
<th>H</th>
<th>A</th>
<th>C</th>
<th>h</th>
<th>Zo</th>
<th>F_i</th>
<th>dL_i</th>
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<td>Brass</td>
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<td>1.3328x10^9</td>
<td>12.7x10^-6</td>
<td>3.05x10^-5</td>
<td>3x10^-4</td>
<td>1x10^-8</td>
<td>200</td>
<td>2x10^-3</td>
</tr>
<tr>
<td>SS410 Tube*</td>
<td>0.4361552x10^-6</td>
<td>3.087x10^9</td>
<td>31.75x10^-6</td>
<td>2.376x10^-5</td>
<td>1x10^-4</td>
<td>1x10^-8</td>
<td>200</td>
<td>2x10^-3</td>
</tr>
<tr>
<td>12L14 (W)**</td>
<td>0.4283855x10^-6</td>
<td>1.960x10^9</td>
<td>31.75x10^-6</td>
<td>1.592x10^-5</td>
<td>2x10^-4</td>
<td>1x10^-8</td>
<td>200</td>
<td>2x10^-3</td>
</tr>
<tr>
<td>Tube*</td>
<td>0.4283855x10^-6</td>
<td>2.156x10^9</td>
<td>31.75x10^-6</td>
<td>1.076x10^-6</td>
<td>2x10^-4</td>
<td>1x10^-8</td>
<td>200</td>
<td>2x10^-3</td>
</tr>
<tr>
<td>12L14 (L)**</td>
<td>0.4593895x10^-6</td>
<td>1.960x10^9</td>
<td>31.75x10^-6</td>
<td>1.12x10^-5</td>
<td>2x10^-4</td>
<td>1x10^-8</td>
<td>200</td>
<td>2x10^-3</td>
</tr>
</tbody>
</table>

* tube material is Incoloy 800
** W—water lubricant, L—diluted cutting fluid lubricant.
5.4.2 Comparison and Difference Analysis

Seven groups of data have been calculated: brass disc wear, 410 stainless steel disc wear, Incoloy 800 tube wear (against SS410 discs), carbon steel 12L14 disc wear (with distilled water lubricant), Incoloy 800 tube wear (against the 12L14 discs, with distilled water lubricant), carbon steel 12L14 disc wear (with diluted cutting fluid lubricant), Incoloy 800 tube wear (against 12L14 discs, with diluted cutting fluid lubricant).

Figures 5.6 to 5.12 show the comparison of the calculated results to the testing results. The differences are described in Figures 5.13 to 5.19 respectively. Generally, the more significant errors happen at regions of low frictional work. These errors are mainly due to the wear loss being too small to accurately measure. Even under accurately contolred conditions, if the wear amount is too small, it is very easy to get significant error. Figure 5.17 is an exception, the wear underestimation is caused mainly by oxidation which is not considered for running-in in this model. But for carbon steel case, oxidation is an very important wear mechanism, modification is needed in future study. Fortunately, what we are really interested in the heat exchanger tube wear problem is long-term wear. From this point of view, the calculated results appears to be acceptable.

![Wear Volume Comparison (theory vs test)](image)

Figure 5.6 Comparison of Brass Disc/Tube Test Results and Calculation Results (Brass Disc, Diluted Cutting Fluid Lubricant)
Figure 5.7  Comparison of SS410 Disc/Tube Test Results and Calculated Results (SS410 Disc, Distilled Water Lubricant)

Figure 5.8  Comparison of SS410 Disc/Tube Test Results and Calculation Results (Tube, Distilled Water Lubricant)
Chapter 5. A Model For The Heat Exchanger Tube Wear

Wear Volume Comparison (theory vs test)

12L14 disc vs Incoloy 800 tube ($W$) (12L14 disc)

![Graph showing wear volume comparison (theory vs test).]

Figure 5.9  Comparison of Carbon Steel 12L14 Disc/Tube Test Results and Calculation Results (Disc, Distilled Water Lubricant)

Wear Volume Comparison (theory vs test)

12L14 disc vs Incoloy 800 tube ($W$) (Incoloy 800 tube)

![Graph showing wear volume comparison (theory vs test).]

Figure 5.10  Comparison of Carbon Steel 12L14 Disc/Tube Test Results and Calculation Results (Tube, Distilled Water Lubricant)
Wear Volume Comparison (theory vs test)
12L14 disc vs Incoloy 800 tube (L) (12L14 disc)

Figure 5.11 Comparison of Carbon Steel 12L14 Disc/Tube Test Results and Calculation Results (Disc, Diluted Cutting Fluid Lubricant)

Wear Volume Comparison (theory vs test)
12L14 disc vs Incoloy 800 tube (L) (Incoloy 800 tube)

Figure 5.12 Comparison of Carbon Steel 12L14 Disc/Tube Test Results and Calculation Results (Tube, Diluted Cutting Fluid Lubricant)
Chapter 5. A Model For The Heat Exchanger Tube Wear

Wear Volume Comparison (theory vs test)

brass disc & Incoly 800 tube (brass disc)

Figure 5.13  Difference Between Test Results and Prediction  
(Brass Disc, Diluted Cutting Fluid Lubricant)

Wear Volume Comparison (theory vs test)

ss410 disc

Figure 5.14  Difference Between Test Results and Prediction  
(SS410 Disc, Distilled Water Lubricant)
Figure 5.15  Difference Between Test Results and Prediction  
(Tube, against SS410, Distilled Water Lubricant)

Figure 5.16  Difference Between Test Results and Prediction  
(Carbon Steel 12L14 Disc, Distilled Water Lubricant)
Chapter 5. A Model For The Heat Exchanger Tube Wear

**Figure 5.17** Difference Between Test Results and Prediction
(Tube, against carbon steel disc, Distilled Water Lubricant)

**Figure 5.18** Difference Between Test Results and Prediction
(Carbon Steel 12L14 Disc, Diluted Cutting Fluid Lubricant)
5.4.3.1 Running-In

From Figures 5.6 to 5.12, a non-linear process is not observable. This is because running-in happens only at the very beginning.

Figure 5.20 shows wear rate (i.e. wear volume/sliding distance) decreases when sliding distance increases (with constant frictional force). After a certain time, the wear rate becomes a constant.

Figure 5.21 shows antiwear strength $U$ against sliding distance $L$. It describes how the antiwear strength changes before conformity.
Figure 5.20  The Relationship Between Wear Rate and Sliding Distance

Figure 5.21  The Relationship Between Antiwear Strength and Sliding Distance
5.4.3.2 \( \sigma \) and \( \lambda \)

Figure 5.22 shows the effect of \( \sigma \) on antiwear strength U. As the surface gets rougher (\( \sigma \) increases), the antiwear strength U is smaller.

Figure 5.23 shows the effect of \( h \) on antiwear strength U. When elastic and plastic deformation goes deeper, which means more material is used to resist wear. For the same material, \( h \) increases, antiwear strength increases.

\( \text{comparison} \)

\text{different sigma's (h=10 um)}

![Graph showing the effect of different sigma values on antiwear strength U.](image)

**Figure 5.22** \( \sigma \) Influence on Antiwear Strength U
Chapter 5. A Model For The Heat Exchanger Tube Wear

5.5 Equation Extension

The previous model is for a tube sliding on a flat disc with a perfect alignment. It can be extended to a more realistic form for application to heat exchanger tube wear.

As Figure 5.24a shows, $D_i$ is the tube inside diameter, $D_o$ is the tube outside diameter, $B$ is the ring thickness. According to the observation of the worn tubes from the previous study...
Chapter 5. A Model For The Heat Exchanger Tube Wear

(Fig. 1.4), the tube wear scar is almost uniform around the contacting area. If the worn depth of the tube is \(d\), then

\[ V = \pi \bar{D} B d \quad (5.13) \]

where

\[ \bar{D} = D_0 - d \]

Substitute equation 5.13 into equation 5.1, i.e

\[ \pi \bar{D} Bd = c \frac{E}{U} \quad (5.14) \]

It is postulated that the tube has a critical worn depth \(d^*\), after this depth, further wear causes unacceptable damage. In practice, the frictional force \(F\), sliding amplitude are random. In order to simplify this problem, the averaged frictional force \(F_a\), calculated sliding distance \(L\) and the averaged sliding speed \(u_a\) are used. Therefore, the external energy input for causing wear is

\[ E = F_a L = F_a u_a t \quad (5.15) \]

where \(t\) is the contacting time.

Substitute equation 5.15 into equation 5.14

\[ d = c \frac{F_a u_a t}{\pi U DB} \quad (5.16) \]
Chapter 5. A Model For The Heat Exchanger Tube Wear

For the critical worn depth $d^*$, the heat exchanger tube life $T$ is

$$T = \frac{\pi UDBd^*}{c F_a u_a} \quad (5.17)$$

For the tube/ring misalignment case as shown in Figure 5.25, if $B$ is contact width, the wear volume can become:

$$v = \pi \overline{D} \frac{d^2}{2} (\tan \alpha + \frac{1}{\tan \alpha}) \quad (5.18)$$

where $\alpha$ is the tilted angle, $d$ is the wear depth.

Let $d=d^*$, and substitute the equation 5.18 into 5.17, the tube wear life is

$$T = \frac{\pi UBDd^*}{2 c F_a u_a} (\tan \alpha + \frac{1}{\tan \alpha}) \quad (5.19)$$
Chapter 5. A Model For The Heat Exchanger Tube Wear

Figure 5.24  Tube/Ring Schematic Diagram

Figure 5.25  Tube/Ring Misalignment Schematic Diagram
Chapter 6

Conclusions and Further Study

In this study, room-temperature tube/disc sliding wear has been investigated with the aid of a testing rig incorporated with improved systems of testing control and data acquisition. Quantitative relationships among wear and its parameters have been determined and a heat exchanger tube wear model has been proposed.

The following is a summary of the main conclusions of this research work:

1. Under constant load and similar frictional conditions, cumulative sliding distance and wear are related linearly. This is true for both the 2mm and 1mm sliding amplitude cases. As well, under constant sliding distance and similar frictional conditions, the relationship between wear and normal load is linear. Further more, under constant sliding distance, the tangential load and wear relationship is also linear.

2. Frictional work, which equals frictional force times sliding distance, is proportional to wear. This has been confirmed for the tube/brass disc, tube/ss410 disc, tube/carbon steel disc combinations under both diluted cutting fluid lubricant and distilled water lubricant conditions.

3. Tube/ring misalignment does not appear to affect the above relationships. With same wear volume, the higher load side will have deeper wear, the lower load side will have shallower wear. The effect of misalignment is averaged out.

4. For alloy tube against brass disc, material transmission has been observed on the tube surface. For the sphere/disc tests, plastic deformation and cracks have been observed on the brass disc surface and its cross section. Stress analysis show $\sigma_z$ is the dominant stress to compress the
material at the contact centre. Away from the centre, the shear stresses $\tau_{xz}, \tau_{yz}$ play an important role.

5. After wear, the roughness $Ra$ and its standard deviation $\sigma$ of the tube and the disc are much the same. But surface topography analysis show that surface roughness neither directly relates to the sliding distance nor to the normal load.

6. At high temperature, with carbon steel A36 ring, the worn tube surface was very smooth, and the sectioned surface showed no observable disturbance of the microstructure, there was no obvious chemical change across the section. This indicates that the tube wear happened at the very top surface. With Inconel 600 ring, the worn tube surface was very rough, and the sectioned surface showed microstructure disturbance. It is believed that for high temperature tests, oxidation becomes an important mechanism for heat exchanger tube wear. An interesting low wear rate for carbon steel 12L14 disc and Incoloy 800 tube combination has been found even at room temperature.

7. A model for heat exchanger tube wear has been developed based on the wear model by Lin and Cheng. The calculated results satisfactorily match the test results. This model has been extended to predict tube life.

8. At the running-in stage, the new model can dynamically describe the non-linear process. At this stage, wear rate decreases with time because the antiwear strength increases with time until the two contact surfaces reach conformity. The standard deviation $\sigma$ and the deformed material thickness $h$ are the two factors affect the antiwear strength. It has been found by analyzing that for the same material rougher surface weakens the antiwear strength, but as the deformed thickness $h$ increases, the antiwear strength increases.
Chapter 6. Conclusions and Further Study

For engineering application, further studies may need to be carried out for the better approaches of the heat exchanger tube wear:

1. High temperature and high pressure study. It has been found from the AECL high temperature and high pressure tube/ring tests (Chapter 4), the worn tube surfaces are obviously oxidized, which is concordant to the Hogmark's study. The investigation of this thesis is only for the room temperature study, because of the limited facility. For the high temperature and high pressure application, modification is needed.

2. Surface analysis. Under high temperature and high pressure condition, the worn tube surfaces have different colour by the different ring material combination (Chapter 4). By the limitation of the provided worn tube specimen, a complete surface analysis can not be carried out. However, a further detailed observation and surface analysis may lead to fruitful understanding, and the results may provide a fundation to consider oxidation and improve this model.

3. Random process. In a real steam generator, the contacts of the heat exchanger tubes and the support plates are random. From the investigation of this thesis, it has shown that the product of frictional force times sliding distance is proportional to the wear amount. In reality, the frictional force and the sliding distance are both random. Therefore, the further investigation of the behaviour of the frictional force and sliding distance as functions of time may be necessary.

4. The improvement of the antiwear strength U. In this investigation, the antiwear strength U only include the original surface parameter σ and material hardness H. Other factors which affect the antiwear strength may need to be quantitatively analyzed.

5. Slip amplitude. In this study, all the tests are under 2mm or 1mm sliding amplitude which are closer to sliding wear than fretting. This is mainly due to the limitation of the test facility. In real heat exchanger tube wear, the sliding distance is much smaller. Modification may be needed for studies of smaller sliding amplitudes.
Bibliography


## Appendix A

### Test Results

#### Table A.3  410 Stainless Steel Disc and Incoloy Tube Test Results
(Lubricated With Distill Water)

<table>
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<td>2.056</td>
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<td>0.79</td>
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#### Table A.5  410 Carbon Steel Disc and Incoloy Tube Test Results
(Lubricated With Distill Water)

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## Table A.1 Sphere and Disc Test Results
*(Lubricated With Diluted Cutting Fluid)*

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N.B.: specimen #121, #122, #133, #108, #119 are not recorded well.
### Table A.2 Flat Brass and Incoloy Tube Test Results
(Lubricated With Diluted Cutting Fluid)

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N.B.: specimen #16, #18, #31 are not well controled.
### Table A.2 continued

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N.B.: specimen #1 is not recorded.

### Table A.4 410 Carbon Steel Disc and Incoloy Tube Test Results

(Lubricated With Diluted Cutting Fluid)

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<th>Dspl. (mm)</th>
<th>CF (N-m)</th>
<th>T-Work (N-m)</th>
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N.B.: *for unpolished carbon steel disc.
Appendix B

Calculation of Jain and Bahadur Model

In order to examine the availability of Jain and Bahadur wear model for heat exchanger tube wear, a specimen of brass C36000 disc/Incoloy 800 tube test is analyzed. The specimen is from the brass disc #1.

1. Material Properties

1.1 True Tensile Strength For Fracture

For brass C36000 disc, $\sigma_t=380$ N/mm², Area Reduction=52%, then

$$S_0 = \frac{P}{A_{true}} = \frac{P}{A_0 (1-\text{Area Reduction})} = \frac{\sigma_T}{(1-\text{Area Reduction})}$$

$$S_0 = 791.67 \text{ N/mm}^2$$

1.2 Fatigue Strength Property

For the material C36000, fatigue strength=140 MPa at $10^8$ cycles.

From equation 2.14, parameter $t$ can be obtained as

$$t = \frac{1T N_T}{1T S_T - 1T S} = 10.6$$
1.3 Equivalent Young's Modulus $E'$

brass, $E_1 = 97$ GPa, $\gamma_1 = 0.3$

tube, $E_2 = 196.5$ GPa, $\gamma_2 = 0.339$

From the equation 2.5

$$E' = \frac{1}{\left(\frac{1-\gamma_1^2}{E_1} + \frac{1-\gamma_2^2}{E_2}\right)} = 72.017 \times 10^3 \text{ N/mm}^2$$

2. Test Parameters

Frictional Coefficient $\mu = 0.18$

Travelled Sliding Distance $L = 19800$ mm

Nominal Contact Area $A = 7.36 \text{ mm}^2$

Normal Load $P = 100$ N

Sliding Speed $u = 80$ mm/s

Surface Roughness $R_a = d = 0.3007$ µm

Standard Deviation $\sigma_1 = 0.3950$ µm, $\sigma = 0.5586$ µm

Asperity Radius $\beta = 0.175$ mm (0.5 µm for sampling step), $\beta = 0.0875$ mm

$h = d/\sigma = 0.54$

Line Density of Asperities (longitudinal) $\eta_L = 2.4$ (1/mm)

Line Density of Asperities (transverse) $\eta_T = 28$ (1/mm)

Area Density of Asperities $\eta = \eta_L \eta_T = 67.2$ (1/mm²)

From the Table A1 of the reference [31]

$$F_1(0.54) = 0.186132, \quad F_{\eta/2}(0.54) / F_1(0.54) = 0.98084$$
F_0(0.54) = 0.294794, \quad F_0(0.54) / F_1(0.54) = 1.58379 \\
F_{3/2}(0.54) = 0.182566

3. Calculation of Wear Particle Volume \( V_p \)

In the reference [31], \( V_p \) is defined as a flattened sphere (special case of ellipsoid) where the radius of the spherical portions of the particle was taken equal to the radius of a discrete contact zone. Thus, the volume of a particle is given by

\[
V_p = \frac{2}{3} \pi a^2 c
\]

where \( a \) is the radius of a discrete contact zone and \( c \) is the thickness of a wear particle.

Assume the half maximum particle height above Ra is \( c = 0.7 \ \mu m \) which can be obtained from the bearing ratio curve.

From the equation 2.12, the radius \( a \) can be determined.

\[
a = \left( \frac{\beta \sigma}{F_0(h) / F_1(h)} \right)^{1/2} = 5.555E-3 \ \text{mm}
\]

\[
V_p = 45.24E-9 \ \text{mm}^3
\]

4. Final Calculation

\[
k = \mu \left( \frac{4 + \gamma}{8} \right) \pi + \frac{1 - 2\gamma}{3} = 0.437
\]
From the equation 2.8

$$P = 33.854 \text{ N}$$

From the equation 2.19

$$K_1 = 23.4 \times 10^6 (1/\text{mm}^2)$$

The calculated wear volume is

$$V = \frac{K_1 P L \eta L V}{2 S_0} k_1 = 470.038 (\text{mm}^3)$$

Since the material density is $\rho = 8.5 (\text{mg/mm}^3)$, the measured wear volume is

$$V = \frac{0.18 \text{mg}}{8.5 \text{mg/mm}^3} = 0.021 (\text{mm}^3)$$

The difference is significant!
Appendix C

Equations for Gaussian Distribution

For zero mean Gaussian distribution, the distribution function is

$$\phi(z) = \frac{1}{\sqrt{2\pi}\sigma} \exp\left(-\frac{z^2}{2\sigma^2}\right)$$

where $\sigma$ is the standard deviation.

The cumulative probability function of the asperity height is

$$\Phi(z) = \frac{1}{\sqrt{2\pi}\sigma} \int_{-\infty}^{z} \exp\left(-\frac{t^2}{2\sigma^2}\right) dt$$

Since about 99.9 per cent of all events lie within $\pm 3\sigma$, therefore, in practice, the distribution curve is truncated to finite limits. Then

$$\Phi(z) = \frac{1}{\sqrt{2\pi}\sigma} \int_{-3\sigma}^{z} \exp\left(-\frac{t^2}{2\sigma^2}\right) dt$$

Substitute $\Phi(z)$ and $z_{i+1} = z_i + \Delta z_i$ into the equation 5.8

$$\int_{z_i}^{z_{i+1}} \int_{-3\sigma}^{z} \phi(z) dz dz = \frac{Kh}{\int_{z_i}^{z_{i+1}} \int_{-3\sigma}^{z} \phi(z) dz dz}$$
When $\Delta z_i$ is small, then

\[
\Delta z_i \int_{-3\sigma}^{z_i} \phi(z) \, dz = \frac{K_h}{\int_{z_i}^{z_i+h} \int_{-3\sigma}^{z} \phi(z) \, dz \, dz}
\]

Therefore,

\[
\Delta z_i = \frac{K_h}{\int_{-3\sigma}^{z_i} \phi(z) \, dz \int_{z_i}^{z_i+h} \int_{-3\sigma}^{z} \phi(z) \, dz \, dz}
\]

where

\[
\phi(z) = \frac{1}{\sqrt{2\pi}\sigma} \exp\left(-\frac{z^2}{2\sigma^2}\right)
\]

and

\[
U_i = \frac{H}{h} \int_{z_i}^{z_{i+h}} \int_{-3\sigma}^{z} \frac{1}{\sqrt{2\pi}\sigma} \exp\left(-\frac{z^2}{2\sigma^2}\right) \, dz \, dz
\]