

**THE INFLUENCE OF SURFACE TOPOGRAPHY
ON SLIDING FRICTION IN BOUNDARY
LUBRICATION**

-by-

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ABSTRACT

This thesis investigates the influence of surface topography on friction in boundary lubrication. Hirst and Hollander's work provided the background of the work on the surface parameters σ and β^* and their relationship with surface damage in boundary lubrication. Their work was repeated and the similarities and differences in results are investigated. The work was extended by using various materials such as a steel or ceramic ball on stainless steel, gear steel or copper. It was shown that the transition from 'safe' to 'unsafe' sliding of boundary lubricated rough surfaces was related to plasticity and elastic conformity.

A numerical elastic-plastic contact model has been developed. This was used to study the effect of surface roughness on the contact states, such as the distribution of contact pressures, real area and spacings. This model was extended to study the plasticity and elastic conformity of rough surface contact in an attempt to derive both the plasticity and conformity indices as a function of normal loads, elastic-modulus and topographic parameters. Results from numerical contact analysis combined with friction tests were used to define the σ - β^* range of surfaces for 'safe sliding'.

It was found that there were three different states of surface contact, namely (i) predominantly plastic asperity stress with gaps between the contact spots, (ii) predominantly elastic asperity stress and (iii) complete elastic conformity where the real area of contact approaches the nominal area, depending on the values of σ and β^* . These three states of contact were also shown to have different mechanistic processes influencing sliding friction and surface damage.

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NOMENCLATURE

Symbol	Meaning
a	Radius of Hertzian contact Width of finite element
A	Contact area
A_{el}, A_a	Elastic, apparent area
A_r, A_p, A_n	Real, plastic and nominal area
b	Length of finite element
c	$1-0.8968/\alpha$
C_{ij}	Influence coefficient
D_p	Peak density
E	Young's modulus
E'	Reduced elastic modulus
$E\{ \}$	Expectation
f	Fraction of metal-metal contact
F	Normal force
$G\{\omega\}$	Power spectral density function (PSDF)
H	Surface hardness
h_{max}	Peak to valley height
K	Kurtosis
K_f	Fracture toughness
k	Metal's shear flow stress (Ch.3) Interfacial shear strength/the metal shear strength (Ch.9)
	$\sigma'^2/\sigma.\sigma''^{1.5}$ (Ch.3)
k'	0.4777λ

$K(k)$	Complete elliptical integral of the first kind
L	Profile length (Ch.2) Asperity loading (Ch.4)
m_n	nth moment of PSDF
N_c	Number of sub-divided Contacts of finite element
n	number of contact spots
p, p_m	mean pressure
P_L	Concentrated point force
$p()$	Probability density function
$Pr()$	Cumulative probability function
$p(x)$	Pressure distribution
P_c	Mean peak curvature (Ch.2) Mean pressure at total conformity (Ch.6)
p_h	Mean pressure at the onset of plasticity
P_k	Mean peak height
P_m	Uniaxial flow pressure
P_r	Mean normal stress
R	Summit radius (Ch. 2) Radius of profile curvature (Ch. 2) Electrical resistance (Ch. 3)
R_a	Centre line average roughness
R_m	Hertzian reduced radius
R_q	Rms roughness
R_t	Total roughness depth
R_{tz}	Mean peak-to-valley height
$R(\tau)$	Autocovariance function
r	distance between pressure and surface elements
S_k	Skewness

$S(\tau)$	Structure function
u	Surface displacement
W	Applied normal loading
w_j	Contact length of finite element
x, y	Local coordinates
y_h	plastic contact length of finite element
y_s	Yield stress of metal
$z(x), z$	Surface heights
Δ	Amplitude of wavy surface
$\Delta\gamma$	Surface energy
$\delta(x), \delta$	Geometric overlap distribution
$\rho(\tau)$	Autocorrelation function
α	Ratio of plastic area to nominal area (Ch.3)
	Bandwidth parameter $m_0 m_4 / m_2^2$ (Ch.5)
σ	Standard deviation
σ_{av}	Average nominal stress
σ_n	Nominal stress
σ_θ	Standard deviation of profile slope
σ^*	Standard deviation of peak heights
β^*	Correlation length
λ	$\sqrt{(m_{20}/m_{02})}$
	Wavelength
$ \theta , \tan\theta$	Absolute mean profile slope
μ	Friction coefficient
τ	Delay length
τ_i	Shear strength of interfacial film
τ_m	Shear strength of metal
ν	Poisson's ratio
ω	Frequency

ψ	Plasticity index
ψ_a	Adhesion index
ψ_c	Conformity index
Ω	$E'\sqrt{(m_{02}/\pi)}$
ζ	Topographic index

CHAPTER 1

INTRODUCTION

1.1 Background

Previous experimental work by Hirst and Hollander(1974) confirmed that sliding surface damage can be related to the surface roughness σ and the correlation length β^* . The principal theme of Hirst and Hollander's work is demonstrated in Figure 1a and shows safe and unsafe sliding regions on a σ , β^* plot, where σ is the rms roughness and β^* is the correlation length defined by the autocorrelation function $\rho=1/e$. The upper boundary from safe to unsafe sliding was shown to be defined by the plasticity index and represented a transition in the asperity contact conditions from a predominantly elastic state of contact stress in the safe region to a significant plastic condition in the unsafe region.

A lower boundary of safe sliding was also identified by experiment to occur with roughness values less than about 0.02 μm rms, and represented a 'too smooth' condition for sliding. The lower boundary appeared to be independent of β^* .

1.2 Project Objectives

The initial objectives of this work were to confirm Hirst and Hollander's upper boundary results, and to show that the lower boundary is also a function of σ and β^* and represents a transition to a third state of asperity contact where the interface is in almost

total contact, and in an elastic state of stress. This third contact state can be considered as very close to the theoretical smooth surface Hertzian state.

The σ - β^* map this research propose to be more realistic, and attempts to confirm by experiment, is shown in Figure 1b where the lower boundary is defined by a conformity index which represents a transition from partial elastic asperity contact to almost complete elastic contact.

1.3 Layout of thesis

In Chapter 2, a brief review of rough surface characterization is given. Surface properties of abraded surfaces, the type of surface being used in the friction tests, are studied. Methods of surface assessment are also discussed in an attempt to obtain an appropriate way to extract surface information of an abraded surface.

In Chapter 3, a literature survey of rough surface contact is given.

In Chapter 4, a numerical elastic-plastic contact model of a ball on directionally structured anisotropic rough surfaces is presented. This is used to study the effect of surface roughness on the contact states, both elastic and plastic. The distribution of contact pressures, real areas and spacings are also studied for different σ/β^* . The results are presented in Chapter 5.

In Chapter 6, two simple numerical contact models of wavy surfaces are studied. These models are used to study the plasticity and elastic conformity of rough surface contact in an attempt to derive both the plasticity index and conformity index as a function of normal loads, elastic-modulus and topographic parameters. Results are used to compare

with the results from the numerical contact model described in Chapter 4.

In Chapter 7, three different states of surface contact are studied. Those occur, when the contacts are in a state of (i) predominantly plastic stress, (ii) predominantly elastic stress and a (iii) complete elastic conformity occurs. These three states are used to study the mechanistic processes influencing sliding friction and surface damage in relation to surface roughness.

In Chapter 8, friction tests are performed to confirm Hirst and Hollander's results on the transition from safe to unsafe sliding of boundary lubricated rough surfaces and are extended to several more materials such as a steel ball or a ceramic ball on stainless steel, gear steel and copper. The differences and similarities in results are discussed. Results from numerical contact analyses combined with friction tests are used to define the $\sigma\text{-}\beta^*$ range of surfaces for 'safe sliding'.

In Chapter 9, friction theories are briefly reviewed. and used to analyse results from the friction tests of Chapter 8.

In Chapter 10, suggestions for further work are given. The implications of the work in explaining some common tribological phenomena are discussed.

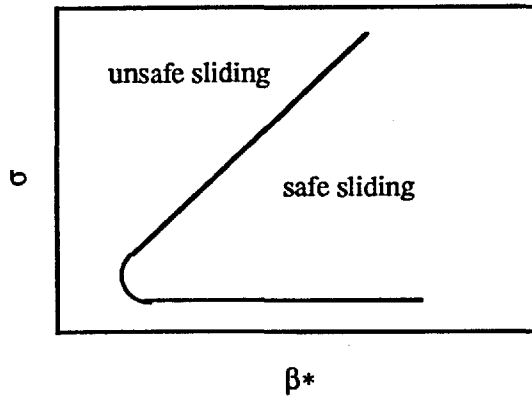


Fig. 1a

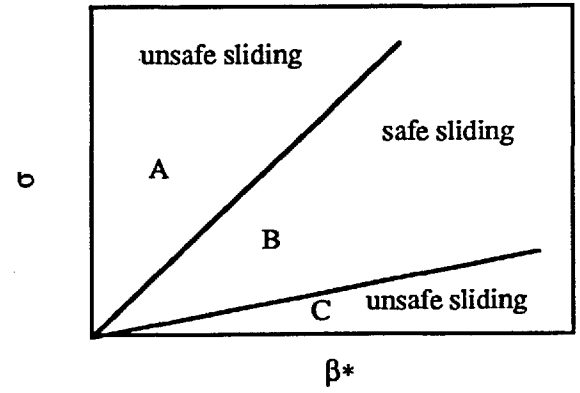


Fig. 1b

Figure 1 Maps of rms roughness σ and correlation length β^* indicating safe and unsafe sliding regions. Fig. 1a is due to Hirst and Hollander and Fig. 1b is proposed as the more correct map.

CHAPTER 2

SURFACE CHARACTERISATION

2.1 Introduction

It has been realised that surface topography plays an important part in understanding the nature of sliding surfaces. No matter how finely finished, almost all machinery rubbing surfaces are extremely rough from a microscopic viewpoint. As a result, the microtopography of a single surface and the nature of a contact between two surfaces form an essential background for developing a fundamental concept of the nature of friction between two sliding surfaces.

At one time measurement of complicated surface parameters was out of question. The manufacturer had to rely upon visual or tactile examination of the surface, using finger or thumb nail to give a qualitative assessment of the suitability of the surface for the desired function (Reason 1944).

In recent years, the availability of modern measuring technology and the advent of the digital computer have enabled us to measure and describe the shape of surfaces. The tribologist, at last, is equipped with what he needs to make new strides in understanding the influence of surface texture upon performance such as wear and frictional behaviour.

There are a large number of different types of surface measuring instruments (e.g. optical, electrical, hydraulic, pneumatic), a review of some of these is contained in Green

(1968). The most commonly used have been profilometer techniques using ever more sensitive stylus instruments to give highly magnified traces of the surface. Under ideal conditions a resolution of a few hundred angstroms can be accomplished.

With the advent of small but powerful micro-computers, a surface can be digitised through the use of fast analogue to digital conversion techniques and the large storage capacities available on modern disc drives means that a micro-computer linked to a surface measuring system, can collect and store a large library of surface topography information. The surface parameters can then be calculated within minutes of the data being recorded.

The digital method has become a popular way of surface analysis because of its flexibility and accuracy. The importance of surface texture has long been realised in its relation to the surface performance. But the question is, what features of the texture are important. Without having sufficient knowledge of the functional significance of roughness parameters and the misuse of computers, this could lead to almost limitless possibilities of defining new parameters, as has happened. Indeed the extent of the growth of new parameters has prompted one author to refer to the present state of affairs as a "parameter rash" (Whitehouse 1982). For example, R_a , R_q , R_t , R_{tz} etc, are all height parameters, where R_a is the average roughness, R_q the root mean square roughness, R_t the total roughness depth and R_{tz} the mean peak-to-valley height. It is apparent that some of the parameters are redundant or irrelevant or both.

The problem of obtaining sufficient information from the profile of a rough surface requires a knowledge of the influence of surface topography upon the function. A surface parameter which is best at describing a certain application may not be true for others. One could say that the average slope, the mean curvature of peaks, and bearing ratio, would be useful additions to the arithmetic average R_a for normal contact and plastic flow situations, but this would have to be modified to include the way in which the curvature changed with height for light loads. Similarly when oil films are present, for instance as in EHL, it is the

density of peaks at different heights rather than their curvature which would be expected to determine the degree of interaction.

For many engineering surfaces, such as spark eroded, shot blasted and ground, it has been realised that the asperity heights can be reasonably well approximated by a Gaussian distribution. The early realisation of this directed attention to the use of random process theory which seemed enable to the most suitable parameters for a given function to be identified. Peklenik (1967) began to classify surfaces by means of their autocorrelation function with the principal aim of better controlling manufacture.

Tallian et al (1964) used some aspects of random process theory to determine the density of crossings of the profile at some arbitrary level. He did this in an attempt to predict asperity interactions through a thin oil film.

In the problems of rough surfaces contact, a subject reviewed in Chapter 3, Greenwood and Williamson (1966), based on the experimental observation that for many surfaces the summit height distribution is approximately Gaussian, derived a plasticity index defined as:

$$\psi = \frac{E'}{H} \left(\frac{\sigma^*}{R} \right)^{\frac{1}{2}} \quad (2.1)$$

where

$$\frac{1}{E'} = \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \quad (2.2)$$

ν_1, ν_2, E_1 and E_2 are the Poisson's ratios and elastic moduli of the two surfaces, σ^* is the standard deviation of the peak heights, R is the summit radius and H is the hardness of the softer of the two surfaces. The plasticity index, in terms of topographical and material properties, is a good example of a parameter that is related to a particular

function. This gives a quantitative method of predicting the contact behaviour (either elastic or predominantly plastic) for a given material and topography.

In the Greenwood and Williamson plasticity index, the assumption that all asperities had the same contact radius was removed by Whitehouse and Archard (1970) who extended the model to allow peaks to have a distribution of curvatures and suggested that surfaces can be represented by two parameters, namely the rms roughness σ of the height distribution and the correlation length β^* , which describes the spatial structure of a surface.

Hirst and Hollander (1974) provided experimental evidence confirming the value of using σ and β^* . In their experiments they measured the critical loads and temperatures at which the onset of severe sliding damage occurred, which was then taken to correspond with the breakdown of boundary lubrication. They suggested that the breakdown occurs at a critical load which caused plastic deformation of the main surface texture, and not at loads which only damage the smaller asperities. This results showed that the nature of rough surface contact is determined by topographical features and material properties, an endorsement of the original conclusion of Greenwood and Williamson.

Other workers, such as Myers (1962), Koura (1980) and Moalic and Torrance (1986) also provided experimental evidence suggesting that the mean profile slope θ can be used to correlate with friction when a hard rough surface slides on a soft smooth counterface. However, this is in no way invalid^{to} the use of σ and β^* as surface parameters to represent a surface. Indeed, Whitehouse and Archard have shown that the statistical geometry of many random profiles, that is the distributions of slopes, curvatures, peak heights and so on, could be represented completely by only two parameters, σ and β^* .

Unfortunately, none of the parameters mentioned are intrinsic properties of a real surface. The numerical values of almost all parameters depends on the scale of

measurement because of the phenomenon of 'self similarity' (Mandelbrot 1977), whereby real surfaces reveal new complexities in their structure as this is examined in finer and finer detail. It has been argued elsewhere (Sayles & Thomas 1978) that this is an inescapable consequence of the mechanism of their formation.

In the manufacturing process, it usually happens that a surface consists of many different wavelengths, some caused by the inaccuracies of the machine tool, deformation of the work under the cutting forces, vibration of the machine, rupture of the material during chip removal and others caused by the actual cutting action. The complete texture of the surface may thus be very complex which usually includes errors of form, which is called the secondary texture, and waviness which is included in the primary texture. No particular wavelength can usually be specified to separate them.

The consequence can be resolved by what is called functional filtering (Thomas & Sayles 1978b). It is affected by applying some form of long and/or short wavelength rejection on a surface having a broad spectrum based on the physical problem involved. This form of 'functional filtering' takes account of the so-called scale-of-size problem, where features which appear as roughness in one application of a surface may well constitute waviness in another. Thus, before the surface roughness can be effectively quantified, an application must be defined.

The objective of this work is to study the influence of surface topography on sliding friction in boundary lubrication. A means of obtaining sufficient information from the profile of a rough surface is essential in order to correlate the appropriate surface parameters effectively with the frictional performance.

In this chapter, ways of rough surface characterisation in the problems of contact are presented. These techniques are used to study the surface features of abraded surfaces,

the types of surface used in the friction tests described in Chapter 7 and 8. As most surface parameters are not intrinsic properties of a surface, the long and/or short wavelength structure irrelevant to the application, need to be removed before the surface parameters are calculated. Methods of filtering to remove the unwanted surface information are discussed.

2.2 Surface Characterisation

2.2.1 Vertical Roughness Parameter

There are various ways to describe vertical roughness of a surface, such as the average roughness, root mean square roughness σ , total roughness depth R_t , mean peak-to-valley height R_{tz} etc, according to the national standard of each country. The two most widely used mean line roughness parameters are rms roughness R_q and the centre line average roughness R_a . R_a is more widely used because of its ease of measurement and is also convenient for the instrument maker because the integrating circuit which computes it is simpler.

The R_a value is defined as the average deviation of the profile, over length L , from the mean height of the surface, i.e.

$$R_a = \frac{1}{L} \int_L |z| dx \quad (2.3)$$

where z is the height measured from the mean line. Thomas (1981) suggests that the R_a has no physical significance. It is true that most of the statistical treatments, for example Whitehouse & Archard (1970), are based upon the rms value of the surface heights.

The R_q or σ is defined mathematically as:

$$\sigma = \left\{ \frac{1}{L} \int_L z^2 dx \right\}^{\frac{1}{2}} \quad (2.4)$$

For many surfaces, namely those having a Gaussian distribution of surface heights, there exists a simple relationship between the σ and R_a values:

$$\sigma \approx 1.25R_a \quad (2.5)$$

R_a and σ are insensitive to wavelength, to small changes in profile geometry, and to occasional high peaks or deep valleys which may be important in the problems of surface contact.

There are two surface parameters namely skewness S_k and kurtosis K which are sensitive to occasional deep valleys or high peaks. The skewness is defined mathematically as,

$$S_k = \frac{1}{\sigma^3} \int_{-\infty}^{\infty} z^3 p(z) dz \quad (2.6)$$

where $p(z)$ is the probability density distribution of z (the height distribution).

It is a useful parameter in defining variables with an asymmetric spread. A surface profile with as many peaks as valleys of the same height and depth has zero skewness. Profiles with peaks removed or deep scratches have negative skewness. Profiles with valleys filled in or high peaks have positive skewness.

The Kurtosis is defined mathematically as,

$$K = \frac{1}{\sigma^4} \int_{-\infty}^{\infty} z^4 p(z) dz \quad (2.7)$$

It represents the peakedness of the distribution. A Gaussian height distribution has a Kurtosis of 3. If $K < 3$, the distribution is said to be platykurtic and has relatively few high peaks and low valleys. If $K > 3$ distribution is said to be leptokurtic and has relatively many high peaks and low valleys.

S_k and K have both been used to characterise running-in and to correlate drag coefficients of rough surfaces (Musker & Lewkowicz 1978). Unfortunately, the values of S_k and K in themselves are distributed with a scatter over a large number of random samples. Nevertheless, they provide additional information about the shape of the height distribution.

2.2.2 Spatial Parameters

R_a and σ are examples of height dependent parameters. However, they do not give information about the variation in the horizontal plane. The knowledge of spatial variation in surface height is essential, as it affects the spatial distance of the asperity contacts. This in turn affects the asperity pressures and surface deflections which interact with each other according to their distance apart.

There are a number of parameters to describe the spatial structure of a surface profile, such as peak density, zero crossing density, average spacing between each zero crossing etc. Other parameters suggested include the mean slope, mean profiles curvature

and various random-process parameters such as the correlation length.

2.2.2.1 Autocovariance and Autocorrelation Functions

The autocovariance (ACVF) and autocorrelation functions (ACF) are popular techniques used to characterise the spatial variation of a surface. For a one-dimensional surface profile the ACVF is defined as:

$$R(\tau) = \frac{1}{L} \int_L z(x) \cdot z(x + \tau) dx \quad (2.8)$$

The function may also be written as an expectation,

$$R(\tau) = E \{z(x) \cdot z(x + \tau)\} \quad (2.9)$$

The ACF is simply the normalised form of the ACVF i.e.

$$\rho(\tau) = \frac{1}{\sigma^2 L} \int_L z(x) \cdot z(x + \tau) dx \quad (2.10)$$

or

$$\rho(\tau) = \frac{1}{\sigma^2} E \{z(x) \cdot z(x + \tau)\} \quad (2.11)$$

The ACVF and ACF of an abraded surface are shown in Figures 2.1 and 2.2. A model found to fit the autocorrelation function of many random surfaces is

$$\rho(\tau) = \exp\left(\frac{-\tau}{\beta^*}\right) \quad (2.12)$$

where $1/\beta^*$ is the decay rate of the function.

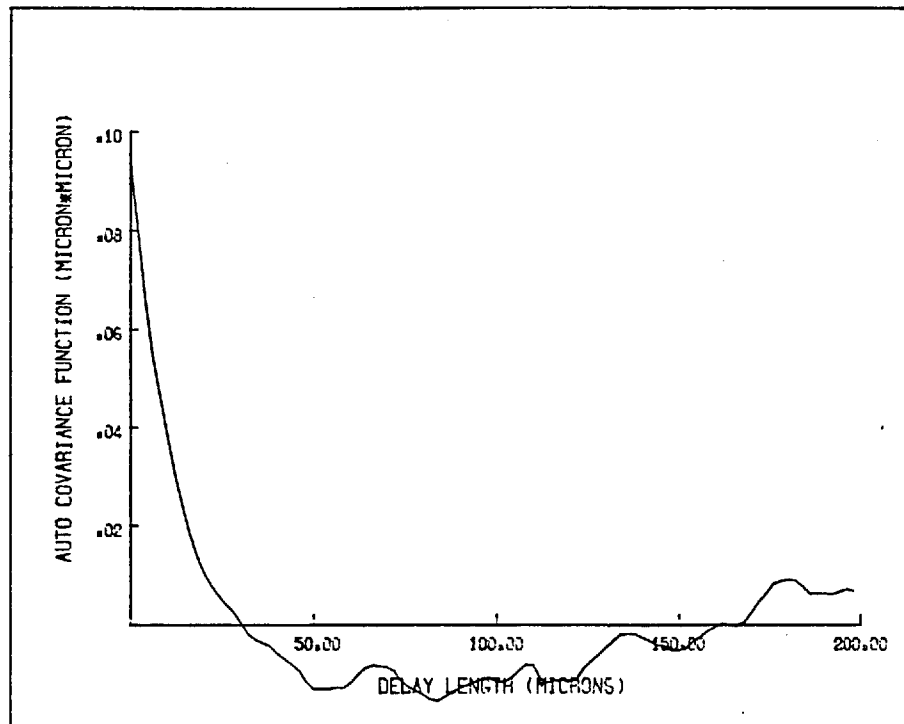


Figure 2.1 Autocovariance function of an abraded surface.

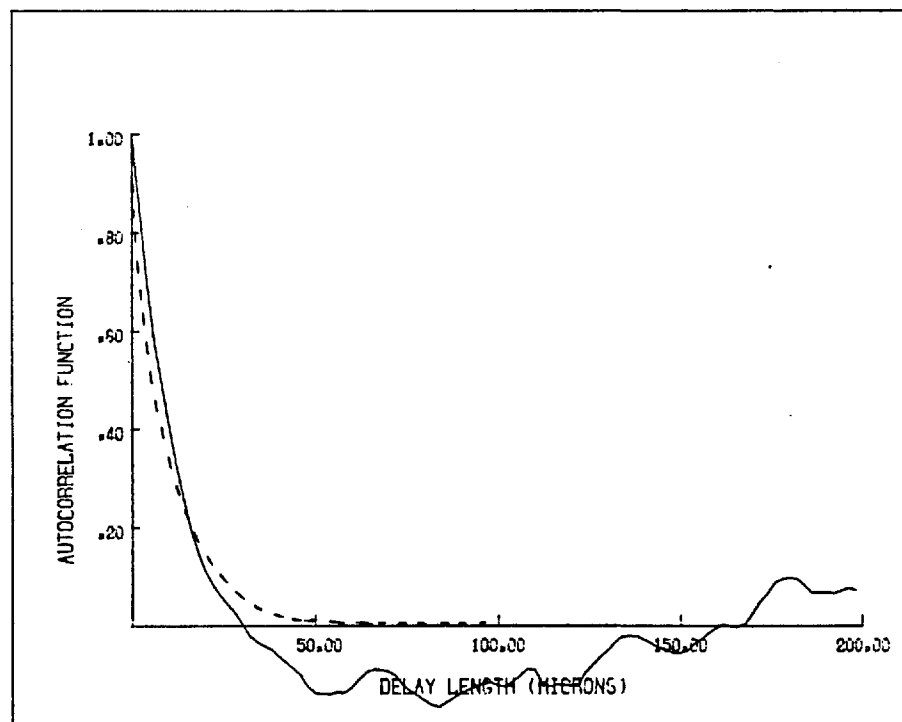


Figure 2.2 Autocorrelation function — of an abraded surface with an exponential function ----.

The correlation length can be defined at $\rho(\tau)=0.1$ (when $\tau=2.3 \beta^*$) or $\rho(\tau)=1/e$ (when $\tau=\beta^*$). Both forms are considered to be a measure of the length at which single height readings become statistically independent of one another. A practical example of the use of correlation length as a surface parameter is the work of Hirst and Hollander (1974) favouring the use of β^* .

2.2.2.2 Structure Function

An alternative method of assessing spatial properties was presented by Sayles and Thomas (1977), who proposed the use of the structure function. The structure function is a measure of the expected value of the square of the height difference as a function of delay length. It is defined as

$$S(\tau) = E \{ [z(x) - z(x + \tau)]^2 \} \quad (2.13)$$

By expanding the above equation, we can express the structure function in terms of the autocorrelation function.

$$S(\tau) = E \{ (z^2(x + \tau)) \} + E \{ z^2(x) \} - 2 \cdot E \{ z(x) + z(x + \tau) \} \quad (2.14)$$

and using $E \{ (z^2(x+\tau)) \} = E \{ z^2(x) \} = \sigma^2$, we have

$$S(\tau) = 2 \cdot \sigma^2 (1 - \rho(\tau)) \quad (2.15)$$

The structure function contains the same information as the autocorrelation function, but it does have some practical advantages. Firstly, it is independent of the mean

value as it is height differences that are calculated. Secondly, the technique can be used to isolate the peak or valley heights from the rest of the profile, Figure 2.3 to 2.5 show the profile, peak and valley structure functions for an abraded surface. This approach offers a means of representing the spatial effects of assymetric surfaces in terms of the upper and lower halves.

As an example in sliding contact, the two bodies are usually supported at discrete contact spots. The topographical changes occurring during a wear or running-in process on the peaks is thus likely to be greater than that on the valleys. The structure function offers a means to monitor the relative extent of these changes. For example, Figure 2.6 and 2.7 show the peak and valley structure functions of a surface before and after one traversal of sliding contact. These results have the same form as those presented by Thomas (1982).

2.2.2.3 Power Spectral Density Function

The power spectral density function (PSDF) is another method of characterising the spatial properties of random signals. The information is essentially the same as contained in the structure function and ACF analysis but is in a spectral form. The PSDF can be attained directly from the ACVF by a Fourier Transform.

$$G(\omega) = \frac{2}{\pi} \int_0^{\infty} R(\tau) \cos(\omega\tau) d\tau \quad (2.16)$$

where $G(\omega)$ denotes the PSDF. It may also be obtained directly from the profile using Fast Fourier Transformation (FFT) techniques.

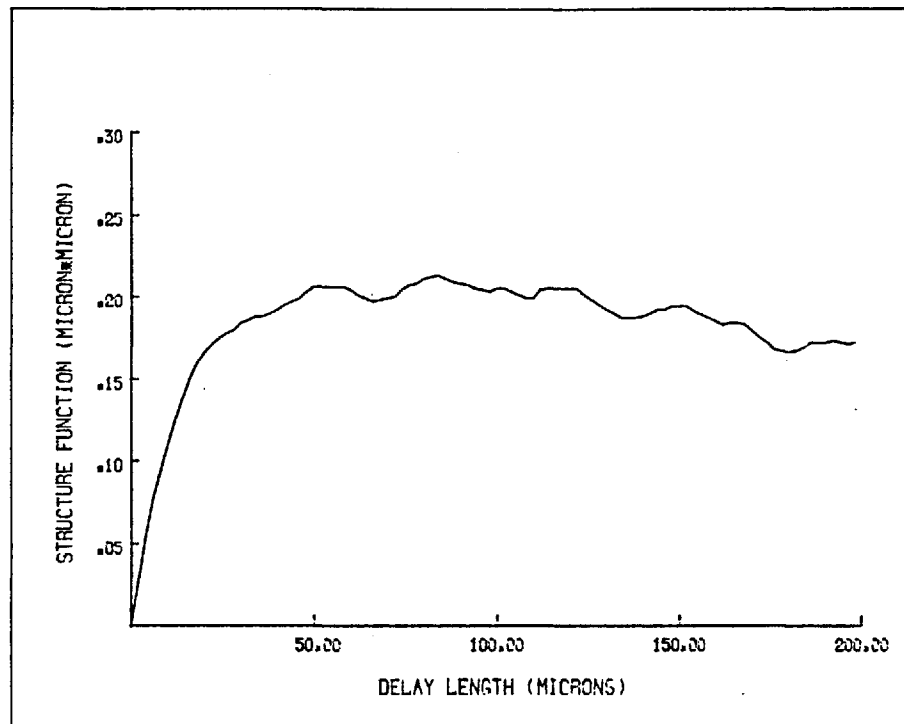


Figure 2.3 Structure function of an abraded surface.

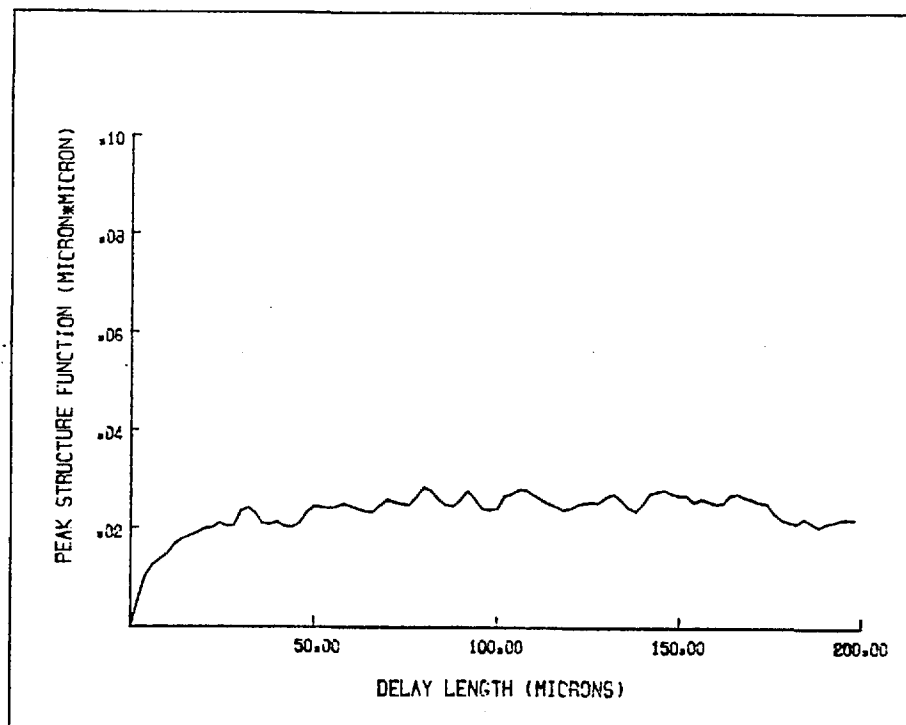


Figure 2.4 Peak structure function of an abraded surface.

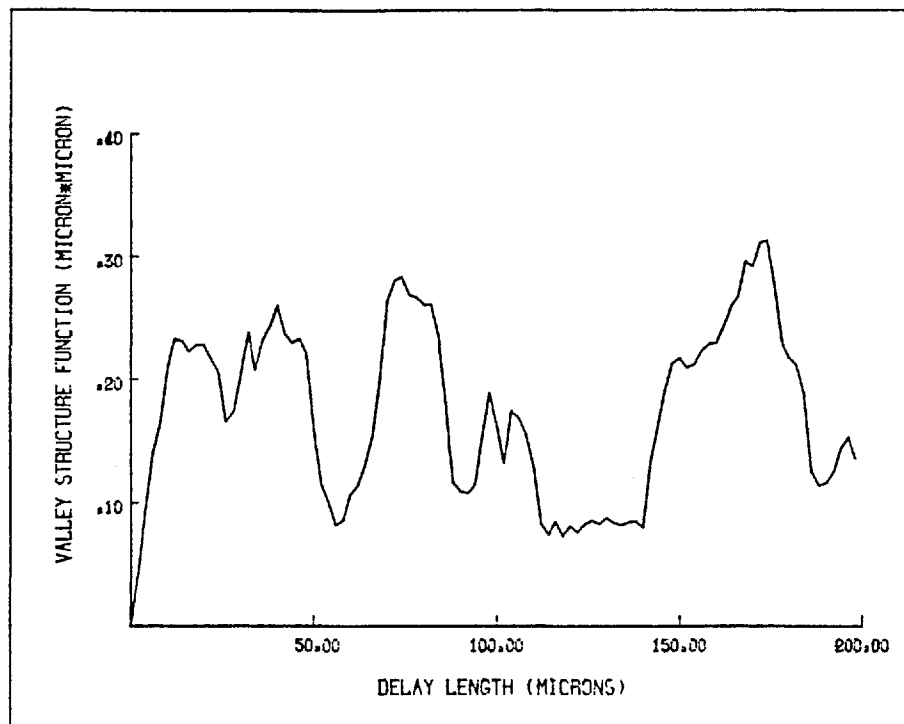


Figure 2.5 Valley structure function of an abraded surface.

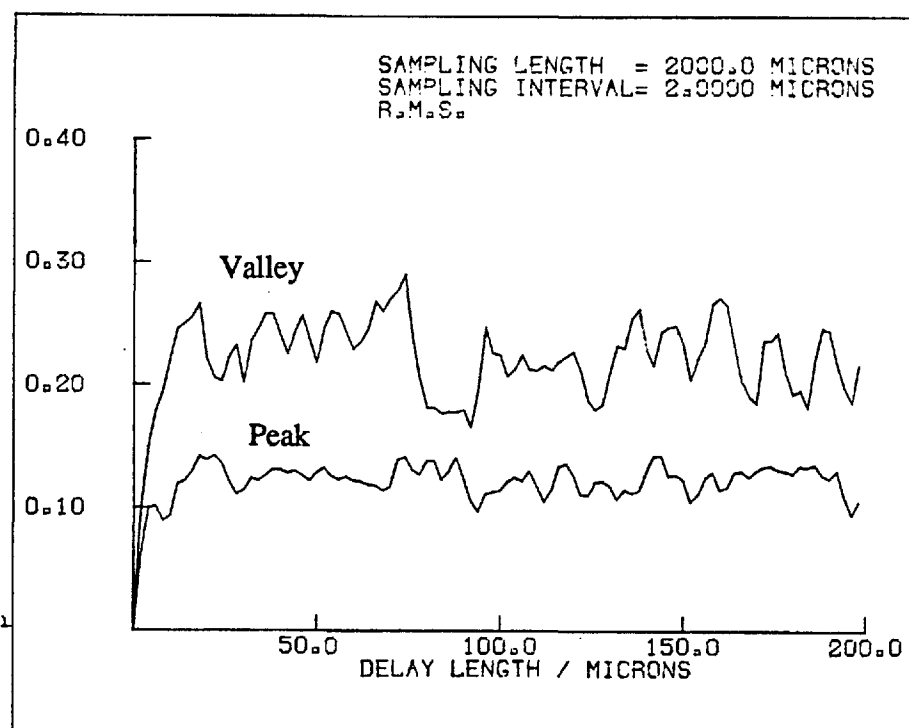


Figure 2.6 Peak and Valley structure function of an abraded surface.

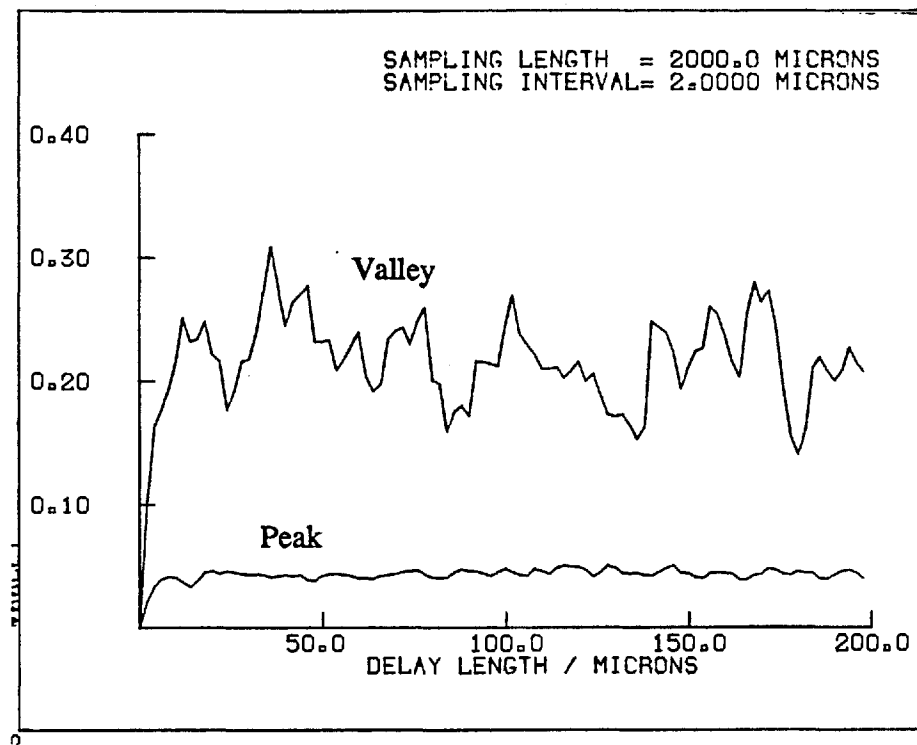


Figure 2.7 Peak and valley structure function of a worn surface profile of Figure 2.6.

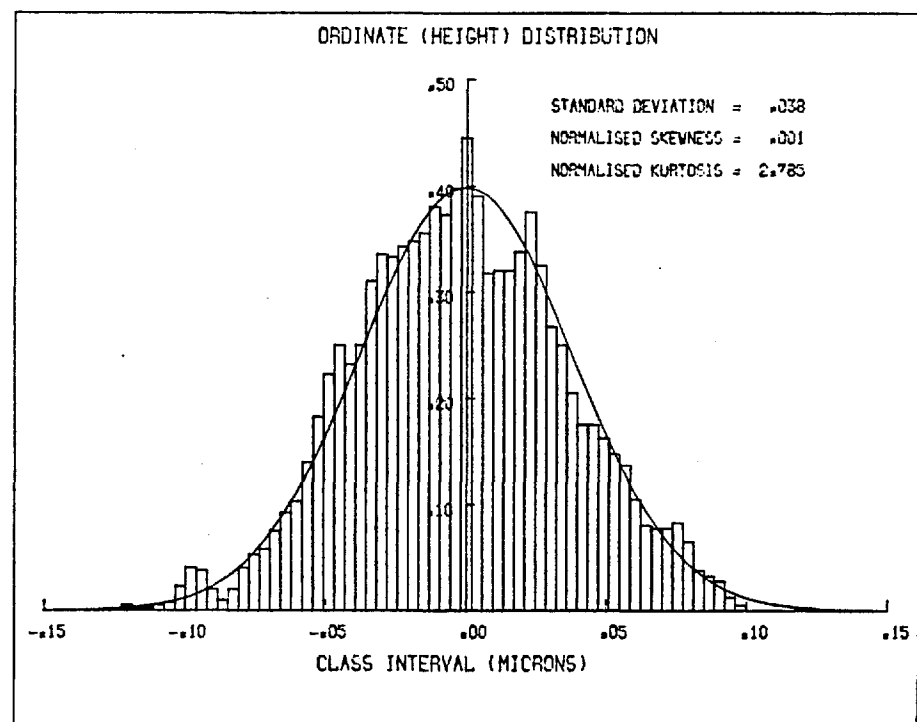


Figure 2.8 Ordinate height distribution of the surface S12 compared with a Gaussian distribution.

$$G(\omega) = \frac{1}{\pi} \int_{-\infty}^{\infty} z(x) \exp(i\omega x) dx \quad (2.17)$$

If the ACVF method is used the process is reversible, i.e.

$$R(\tau) = \int_0^{\infty} G(\omega) \cos(\omega \tau) d\omega \quad (2.18)$$

Thomas favours the use of ACF and structure function techniques suggesting that the sinusoidal representation implied by the use of the PSDF give a broad spectrum to account for the complexities of asperity shape. Peklenik (1967/68) gives experimental results for PSDF and ACF of a set of eleven different engineering surfaces and demonstrates the practical application of random process methods.

2.3 Surface Measurement and Analysis

In this section, surface topographic features of abraded surfaces are examined. Surface roughness parameters and their interdependence together with their consistency and variability are studied. The objective of the study is to obtain a comprehensive knowledge of the properties of abraded surfaces in order to correlate the appropriate surface parameters with surface performance and particularly in relation to sliding friction and surface damage.

2.3.1 Surface Production and Measurement

The surfaces are in a form of nominal flat and prepared using silicon carbide

abrasive papers of varying grade and condition to produce different roughness and spatial structure.

Measurements are made perpendicular to the lay direction using a Talysurf 4 with a fine tip stylus. The digitised recorded profiles are stored on an LS1-11/23 based micro-computer. An external datum called the optical flat attachment is used to provide a more accurate measurement. Each profile consisted of 1000 digitised points at a spacing of 2 μm , thus giving a profile length of 2 mm.

2.3.2 Analysis of Surfaces

50 abraded surfaces covering a range of roughness are subjected to the study of their height distribution, bearing area curve and contact parameters such as σ , β^* , peak curvature, profile slope etc. Results without filtering are summarised in Table 2.1

2.3.2.1 Height Distribution

The height distribution is constructed by plotting the number or proportion of surface heights lying between two specific heights as a function of the height. Figure 2.8-2.10 show the height distribution, peak height distribution and valley height distribution of Surface 12. A Gaussian distribution is superimposed for comparison, and the values of skewness and kurtosis indicate that the sample surface has an approximately Gaussian height distribution. However, some surfaces with height distributions which deviate from the Gaussian form were also recorded, (see Figure 2.11 which represents the distribution for Surface 33). The values of skewness and kurtosis also indicate that surface 33 is far from Gaussian.

Surfaces	σ (μm)	β^* (μm)	S_k	K	At $\tau=2.3\beta^*$			At $\tau=0.23\beta^*$		
					Peak height (μm)	Peak Curvature (μm^{-1})	Profile slope (rad)	Peak height (μm)	Peak Curvature (μm^{-1})	Profile slope (rad)
S01	0.19	5.51	-2.12	12.0	0.131	2.24E-3	7.47E-3			
S02	0.22	16.8	-0.84	4.56	0.175	3.58E-4	3.11E-3	0.11	7.97E-3	1.60E-2
S03	0.30	3.34	0.015	3.96	0.242	1.30E-2	2.20E-2			
S04	0.076	12.2	-0.18	3.38	0.063	2.30E-4	1.55E-3	0.04	1.30E-2	1.10E-2
S05	0.24	4.64	-0.14	5.55	0.186	4.65E-3	1.20E-2			
S06	0.135	18.3	-2.12	12.7	0.100	1.32E-4	1.68E-3	0.05	3.59E-3	7.73E-3
S07	0.33	11.1	-1.00	4.82	0.254	1.13E-3	7.19E-3	0.15	4.50E-2	4.30E-2
S08	0.31	11.0	-2.56	18.1	0.209	8.76E-4	6.51E-3	0.13	3.30E-2	3.40E-2
S09	0.18	2.90	-0.05	4.82	0.138	9.18E-3	1.50E-2			
S10	0.21	3.65	-1.25	7.38	0.149	6.14E-3	1.30E-2			
S11	0.10	3.97	-0.93	5.89	0.073	2.61E-3	5.71E-3			
S12	0.038	15.3	0.010	2.79	0.031	7.23E-5	6.11E-4	0.18	4.34E-3	4.05E-3
S13	0.054	5.81	-0.73	4.48	0.038	5.98E-4	2.04E-3			
S14	0.092	3.31	-0.90	5.28	0.070	3.50E-3	6.65E-3			
S15	0.11	3.58	-0.62	3.90	0.086	3.83E-3	7.57E-3			
S16	0.036	23.2	-0.36	3.35	0.030	2.71E-5	3.84E-4	0.19	1.99E-3	2.65E-3
S17	0.09	32.5	-0.95	4.80	0.065	7.26E-4	7.26E-4	0.031	7.38E-4	2.78E-3
S18	0.03	17.1	0.152	4.33	0.021	3.64E-5	4.36E-4	0.013	2.06E-3	2.46E-3
S19	0.04	4.89	-0.75	6.44	0.029	6.88E-4	1.82E-3			
S20	0.083	2.96	-0.50	4.46	0.064	4.13E-3	7.09E-3			
S21	0.24	3.56	-0.59	4.78	0.187	8.38E-3	1.60E-2			
S22	0.037	9.67	-0.45	3.53	0.027	1.54E-4	8.55E-4	0.02	1.00E-2	6.82E-3
S23	0.035	20.0	-0.23	5.90	0.023	3.02E-5	3.84E-4	0.017	2.02E-3	2.55E-3
S24	0.045	32.4	-0.22	4.31	0.035	1.63E-5	3.32E-4	0.021	1.05E-3	2.15E-3
S25	0.034	34.1	-1.03	6.62	0.025	1.13E-5	2.27E-4	0.016	7.15E-4	1.38E-3
S26	0.035	19.4	-0.50	4.01	0.028	3.48E-5	4.36E-4	0.018	2.51E-3	2.98E-3
S27	0.13	11.1	-0.20	4.62	0.098	4.35E-4	2.81E-3	0.048	1.80E-2	1.70E-2
S28	0.036	2.89	-0.08	2.87	0.026	1.90E-3	3.02E-3			
S29	0.032	10.2	-0.90	5.86	0.024	1.14E-4	6.63E-4	0.015	7.26E-3	4.97E-3
S30	0.039	13.8	-0.63	4.20	0.03	8.00E-5	6.46E-4	0.016	3.31E-3	3.79E-3
S31	0.255	20.1	-1.71	9.75	0.19	2.64E-4	2.85E-3	0.094	8.49E-3	1.50E-2
S32	0.143	12.3	-1.19	5.78	0.113	4.32E-4	2.90E-3	0.048	1.20E-2	1.40E-2
S33	0.68	18.6	-0.25	2.52	0.59	9.66E-4	1.00E-2	0.237	2.50E-2	4.80E-2
S34	0.088	4.80	-0.37	3.46	0.07	1.72E-3	4.40E-3			
S35	0.06	2.14	-0.15	3.43	0.045	6.13E-3	6.53E-3			
S36	0.07	5.03	-1.13	6.51	0.051	1.07E-3	3.28E-3			
S37	0.18	10.3	-1.57	10.8	0.124	5.97E-4	3.77E-3	0.082	0.036	0.025
S38	0.141	2.82	-0.38	3.69	0.112	7.90E-3	1.20E-2			
S39	0.084	7.41	-0.94	4.34	0.057	5.29E-4	2.57E-3			
S40	0.036	6.05	-0.76	4.91	0.027	3.98E-4	1.33E-3			
S41	0.043	39.6	-1.40	7.63	0.032	1.23E-5	2.62E-4	0.021	7.17E-4	1.64E-3
S42	0.021	26.3	-1.38	6.90	0.016	1.24E-5	1.92E-4	0.010	7.82E-4	1.19E-3
S43	0.028	3.48	-0.78	6.94	0.02	8.49E-4	1.73E-3			
S44	0.029	43.7	-0.68	4.27	0.024	7.95E-6	1.75E-4	0.013	2.51E-4	7.85E-4
S45	0.028	22.3	-1.38	10.97	0.021	2.00E-5	3.14E-4	0.013	1.58E-3	2.09E-3
S46	0.024	45.3	-0.62	5.32	0.021	5.20E-6	1.57E-4	0.011	2.41E-4	7.33E-4
S47	0.014	34.3	-0.42	2.69	0.012	5.87E-6	1.05E-4	0.006	1.82E-4	5.24E-4
S48	0.025	26.3	1.31	11.61	0.023	1.80E-5	2.62E-4	0.014	5.98E-4	1.12E-3
S49	0.054	19.0	-1.58	11.71	0.044	5.96E-5	7.16E-4	0.020	1.1E-3	3.32E-3
S50	0.047	22.6	-0.71	5.09	0.043	4.48E-5	5.41E-4	0.014	8.24E-4	2.57E-3

TABLE 2.1 Surface parameters of 50 abraded surfaces

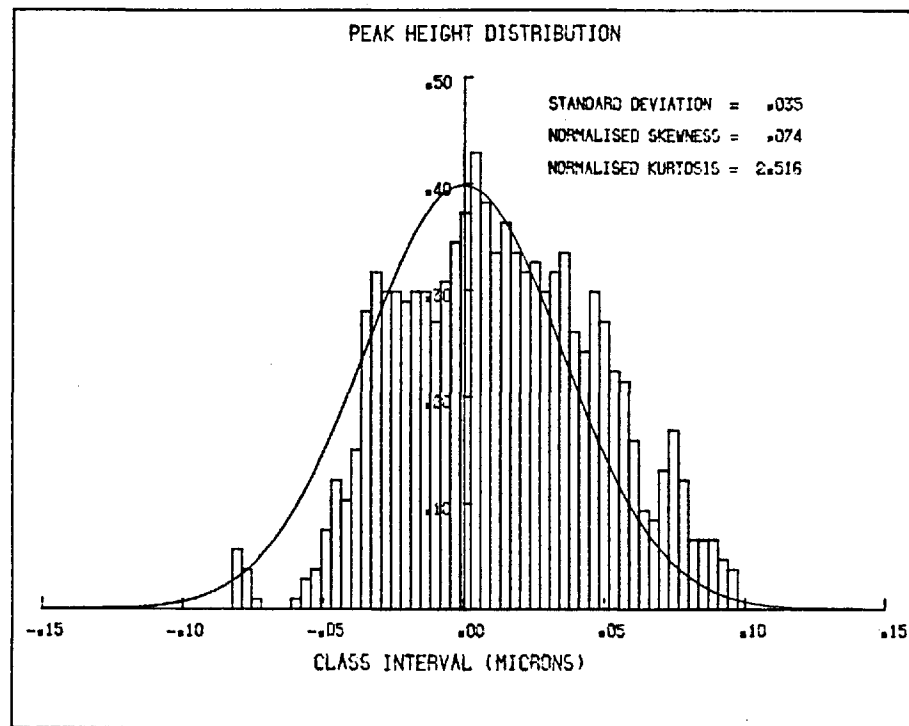


Figure 2.9 Peak height distribution of surface S12 compared with a Gaussian distribution.

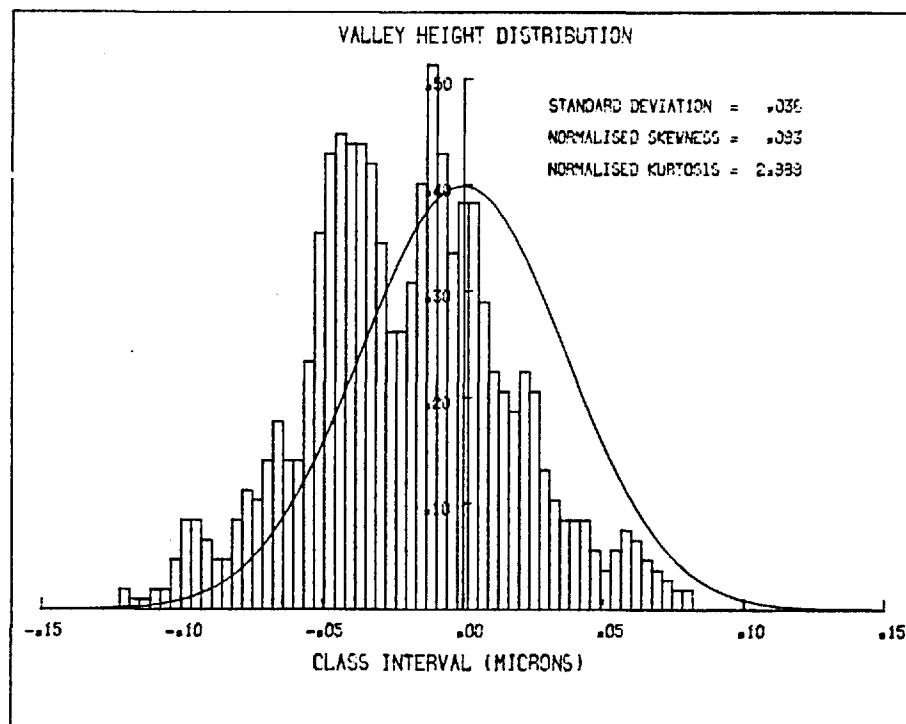


Figure 2.10 Valley height distribution of surface S12 compared with a Gaussian distribution.

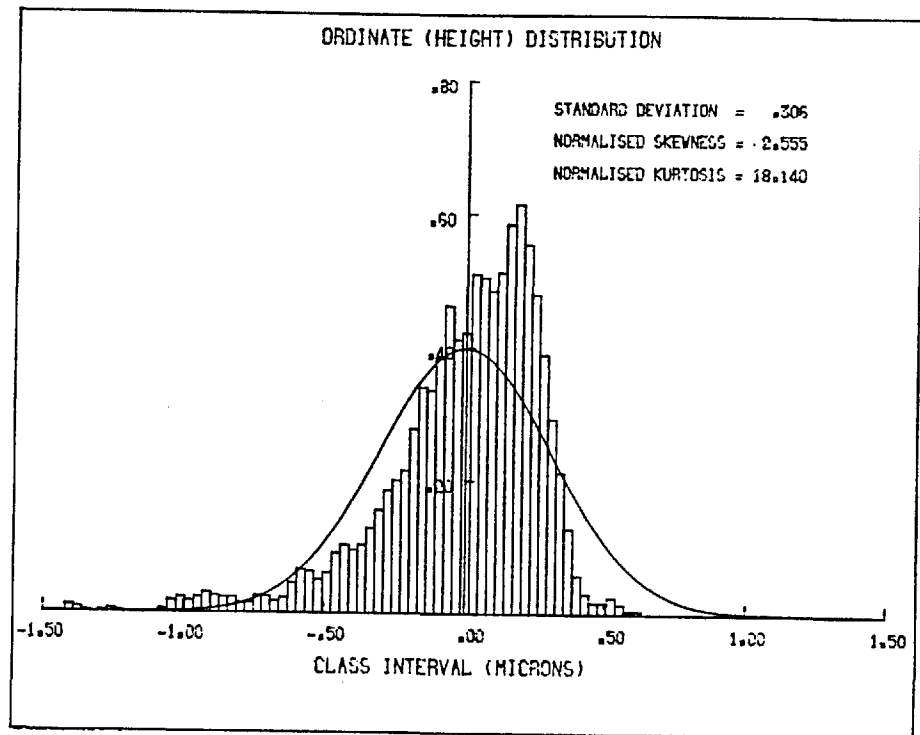


Figure 2.11 Ordinate height distribution of the surface S08 compared with a Gaussian distribution.

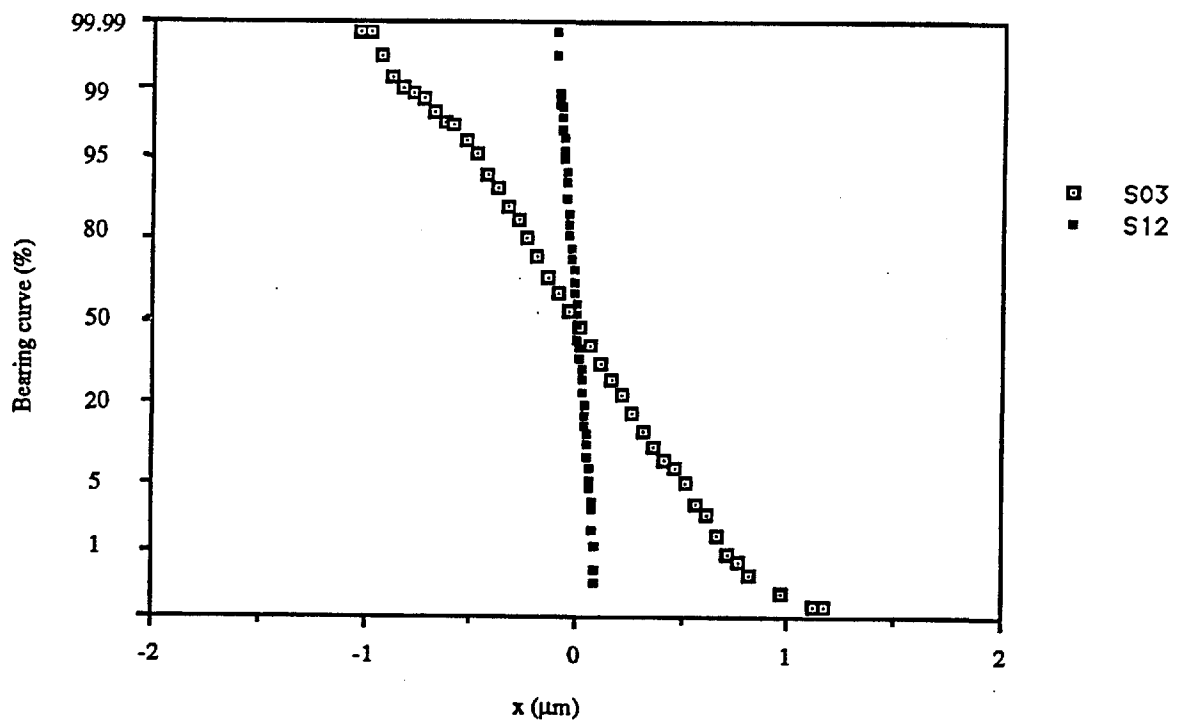


Figure 2.12 Bearing curve of surfaces S03 and S12 having a Gaussian distribution.

According to Thomas (1981), a surface height distribution with the values of $-1 < S_k < 1$ and $1 < K < 5$ can be regarded as closely Gaussian. Table 2.1 shows a large number of surfaces, those underlined have a close to Gaussian height distribution. Only a few surfaces have a high values of K . Most surfaces have a negatively skewed distribution. It has been reported both theoretically and experimentally (Sayles & Thomas 1976) that grinding and abrasion can produce surfaces having skewed distributions. In fact, many of the more common machining process produce negatively skewed distributions, such as honing, lapping and superfinishing.

The information about how closely surfaces can follow a Gaussian height distribution is important. Traditionally, stochastic techniques (Greenwood & Williamson 1966, Onions & Archard 1973, Bush & Gibson 1975) have been used to analyse rough surface contact problems by assuming surface heights or peaks to be Gaussian. By this approach, the interfacial force and contact area or bearing area can be predicted as a function of separation from the mean line.

2.3.2.2 Bearing Area

The bearing area curve is constructed according to that suggested by Abbott and Firestone (1933) of slicing the surface from the peaks down and plotting the cumulative land length as a function of the height of each slice. Figure 2.12 shows the bearing area curves for surfaces 3 and 12 which have approximately Gaussian height distributions. Figure 2.13 shows the bearing area curves for surfaces 3 and 8 which have a similar roughness but different K . The graph is scaled such that a straight line is produced when the distribution is Gaussian. By doing so, it is easy to see whether a surface has a Gaussian distribution, and in particular whether its uppermost strata follow the same distribution as the general surface. A further advantage of this representation is that the σ of the surface

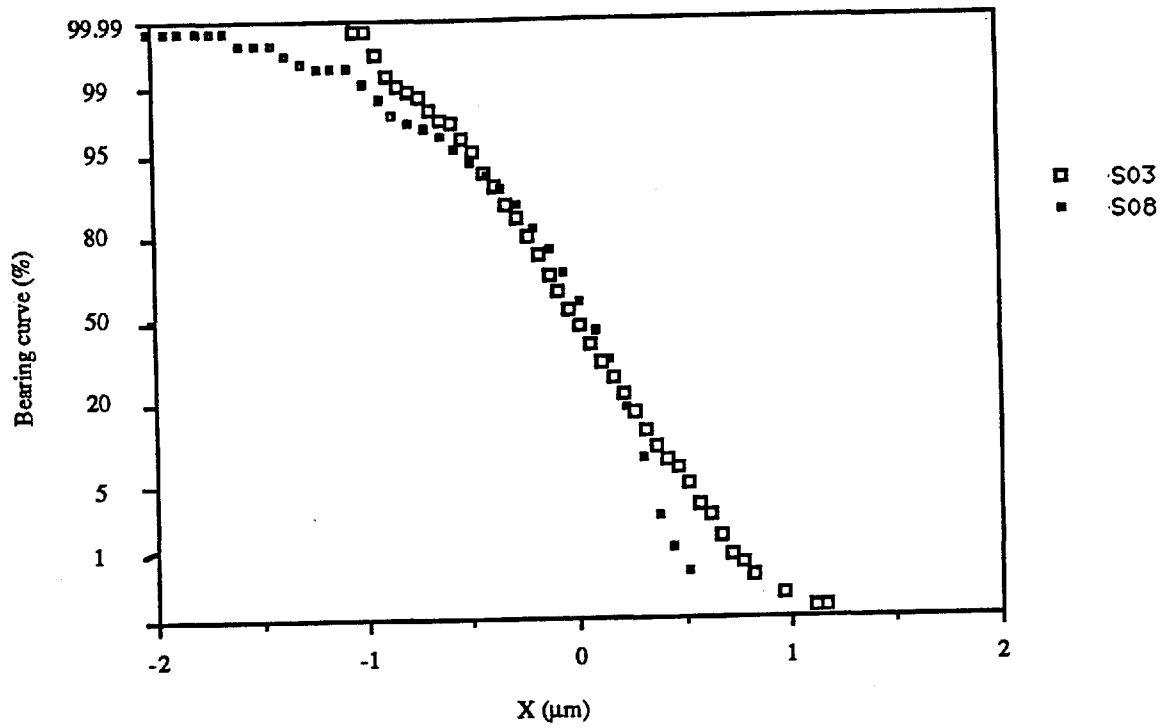


Figure 2.13 Bearing curve of surfaces S03 and S08 with a different kurtosis.

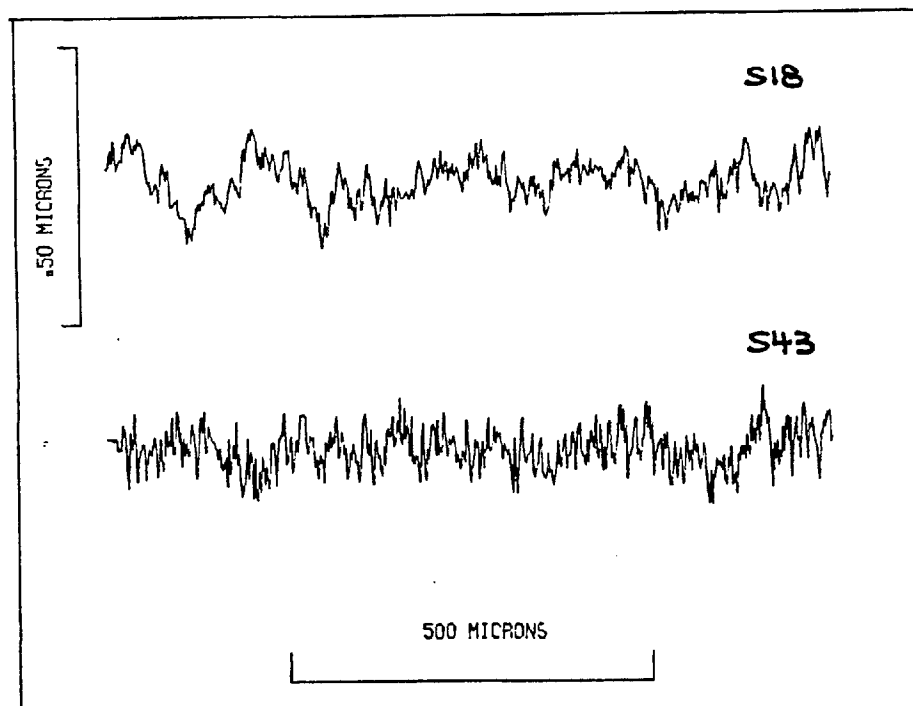


Figure 2.14 Surface profiles of surfaces S18 and S43 having difference spatial structure.

can be read directly from the slope of the line. The σ is given by difference between the height of the 95th percentile and that of the 5th percentile, divided by 3.29.

The physical significance of a bearing area curve is obvious. It represents the contact plateau area at a particular height if the hills were all worn down to that height. The advantage of this method is that it provides a simple way of estimating the contact area as a function of separation.

The shape of the bearing area curve depends on the value of skewness or kurtosis. At a separation above the mean line, the bearing area is greater for a surface with negatively skewed distribution than that of a surface with a closely Gaussian distribution. On the other hand, at a separation below the mean line, the bearing area for a surface with a closely Gaussian height distribution is greater. In many practical situations contact occurs on the topmost strata of the surfaces. Therefore, the contact area of a surface with negatively skewed distribution would be greater than that derived by Gaussian distribution assumptions.

2.3.2.3 Spatial Surface Parameters

Figure 2.14 shows the surface profiles of surfaces 18 and 43 which have the same roughness but clearly the spatial arrangement of surface heights is very different. This might represent an important functional difference in many practical engineering cases, this being particularly true with rough surface contact problems where asperity contacts interact with each other according to their vertical roughness as well as their spatial distance.

The autocorrelation function can be used to represent a surface's spatial variation. The correlation length can be defined at $\rho(\tau)=0.1$ (when $\tau=2.3\beta^*$) or $\rho(\tau)=1/e$ (when

$\tau=\beta^*$). Practically, the ACF near the region at $\tau=2.3\beta^*$ tends to be variable. Therefore, $\rho(\tau)=1/e$ is used to define the correlation length of a surface. Figure 2.15 shows a selection of surface profiles with increasing β^* . It can be seen that surfaces with larger β^* have less spatial height variation than the surfaces with smaller β^* .

2.3.2.4 Surface Contact Parameters

It has been suggested that peak height, peak curvature and profile slope are important parameters in the problems of surface contact. Greenwood and Williamson (1966) used the peak height and peak curvature to define what they called a plasticity index which determines the mode of surface deformation. Whitehouse and Archard (1970) later resolved the plasticity index in terms of σ and β^* . Blok (1952) and Halliday (1955) used the mean profile slope as a criterion for the onset of plastic flow. Experimentally, Myers (1962) claimed the mean profile slope gave the best correlation with friction among other surface parameters such as rms roughness and profile curvature. Hirst and Hollander (1974) provided experimental evidence that surfaces can be divided by the use of σ - β^* maps into those likely to fail catastrophically and those likely to remain undamaged during sliding.

The surface parameters mentioned can be expressed in terms of σ and β^* . Whitehouse and Archard (1969) using an exponential autocorrelation function model showed that the statistical geometry of many random profiles, that is the distribution of slopes, curvatures, peak heights and so on, could be represented completely by two parameters, the rms σ and the correlation length β^* . They used the sampling distance $\tau=0.23\beta^*$ to define the fine structure and $\tau=2.3\beta^*$ for the main structure and expressed the mean peak height p_k , mean peak curvature p_c , and the absolute mean profile slope $|\theta|$ in terms of σ and β^* .

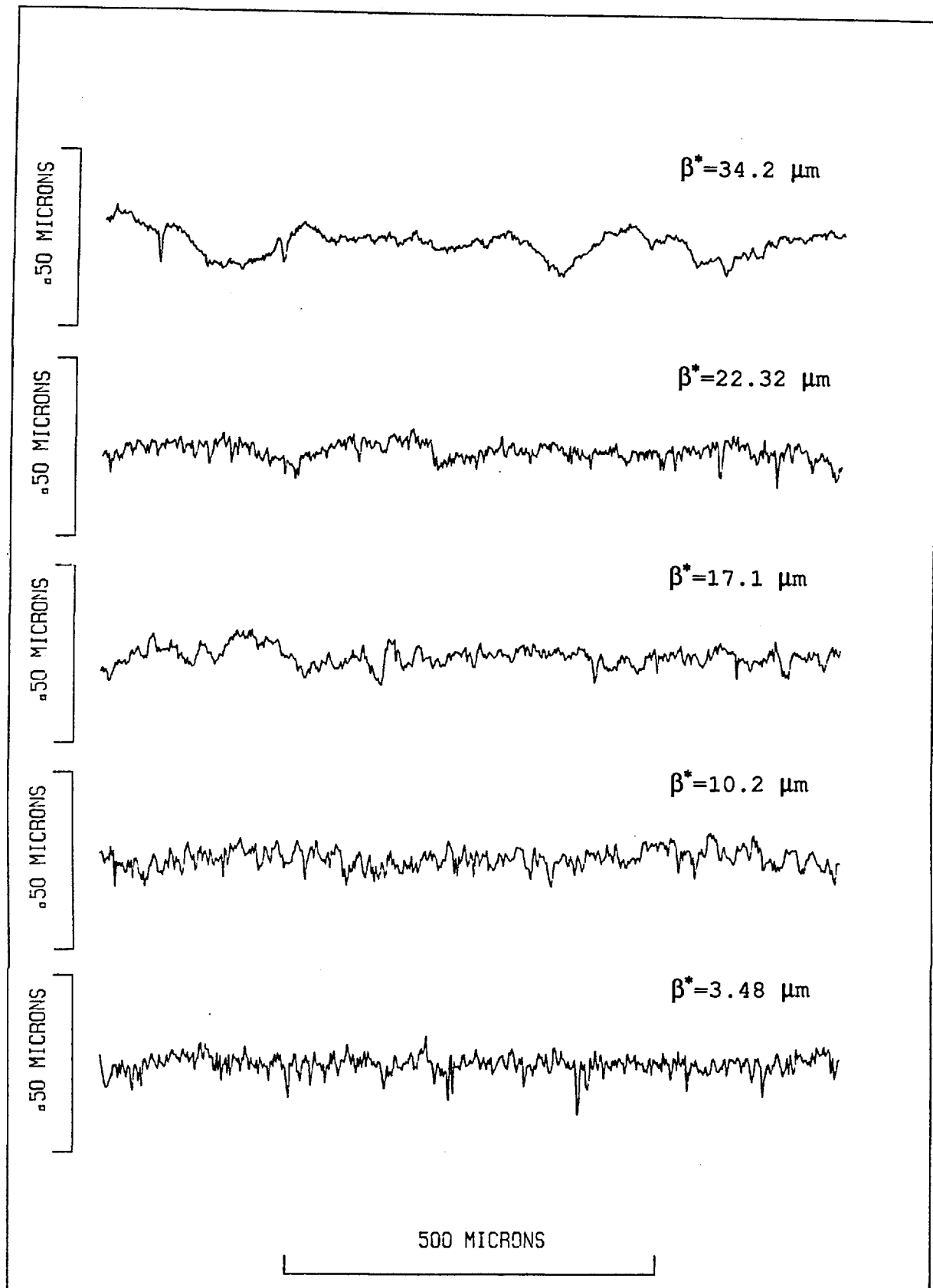


Figure 2.15 Surface profiles with a decreasing β^* .

For the fine structure,

$$\overline{p_k} = 0.48\sigma \quad (2.19)$$

$$\overline{p_c} = 20 \frac{\sigma}{\beta^2} \quad (2.20)$$

$$|\overline{\theta}| = 1.5 \frac{\sigma}{\beta^*} \quad (2.21)$$

and for the main structure,

$$\overline{p_k} = 0.83\sigma \quad (2.22)$$

$$\overline{p_c} = 0.45 \frac{\sigma}{\beta^2} \quad (2.23)$$

$$|\overline{\theta}| = 0.24 \frac{\sigma}{\beta^*} \quad (2.24)$$

It is instructive to know how well these relationships hold for the abraded surfaces being studied. The sampling distance $\tau=0.23\beta^*$ and $\tau=2.3\beta^*$ are used to calculate the mean peak height, mean peak curvature and mean absolute profile slope. The peak curvatures may be represented by the second difference differential at the required point. In this case a difference formula is:

$$p_c = z'' = \frac{2z_i - z_{i+1} - z_{i-1}}{\tau^2} \quad (2.25)$$

where z_i'' is the second differential at the i th point. To obtain the peaks, the i th point is first tested to ensure it is part of the correct population, if it fails the test it is ignored. Then, the mean peak height and peak curvature are calculated over the total number of peaks.

Whitehouse and Archard defined the profile slope as,

$$\theta = \frac{z_{i+1} - z_{i-1}}{2\tau} \quad (2.26)$$

Figure 2.16 shows the peak height against σ for $\tau=2.3\beta^*$ and $\tau=0.23\beta^*$ respectively. The results compare very well with the expression derived by Whitehouse and Archard. Similarly, the results of peak curvatures against σ/β^{*2} , shown in Figure 2.17 and absolute mean profile slope against σ/β^* , shown in Figure 2.18 also shows good agreement with Whitehouse and Archard's results. Therefore, the use of σ and β^* can be reasonably accepted to define the functional behaviour of ^a surface contact problem.

2.3.4 Surface Variability

This refers to the consistency of surface parameters obtained from each profile measured on different position of the surface. The variability of the surface parameters depends on the conditions of surface production. From a statistical point of view the degree of confidence that can be given to a measured parameter is of foremost important. Therefore, it is important to consider the variability of the surface parameters σ and β^* . Six profiles were recorded from each surface. The relative positions of these profiles were the same on each surface, with each profile taken 1mm apart. The maximum, minimum and mean values of σ and β^* for each surface were obtained. Figure 2.19 and 2.20 show the spread of σ and β^* relative to their mean value, indicating that scatter up to 100% is possible, particularly at small σ and β^* .

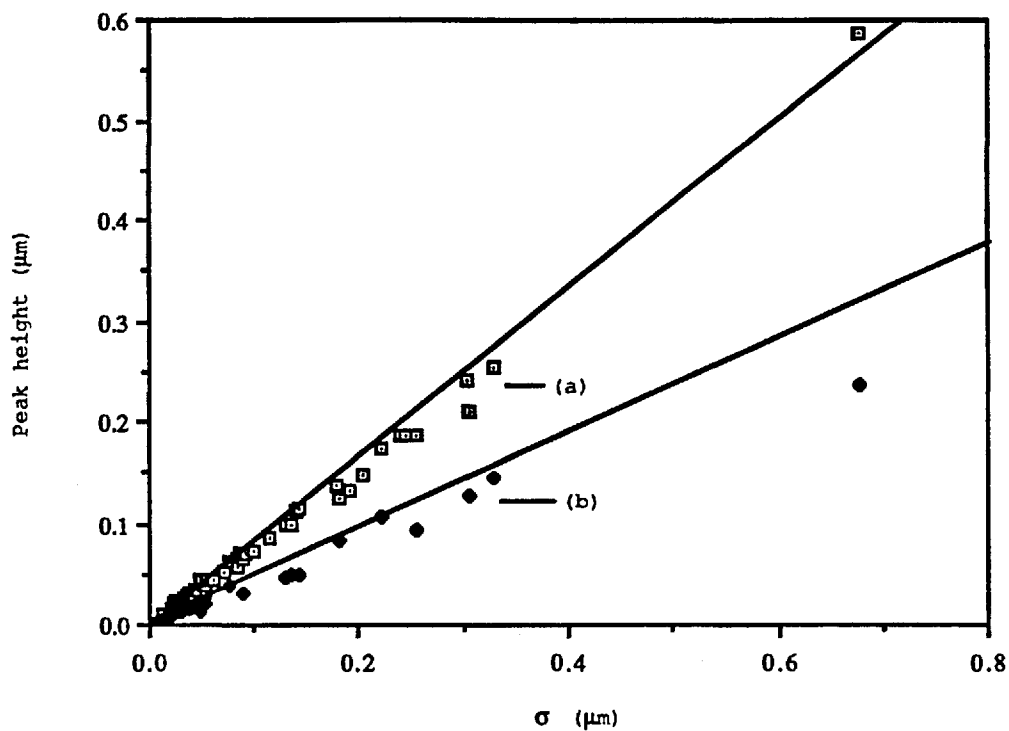


Figure 2.16 Variation of peak height with σ . (a) $\tau=2.3\beta^*$ and (b) $\tau=0.23\beta^*$. The straight lines give results derived by Whitehouse and Archard.

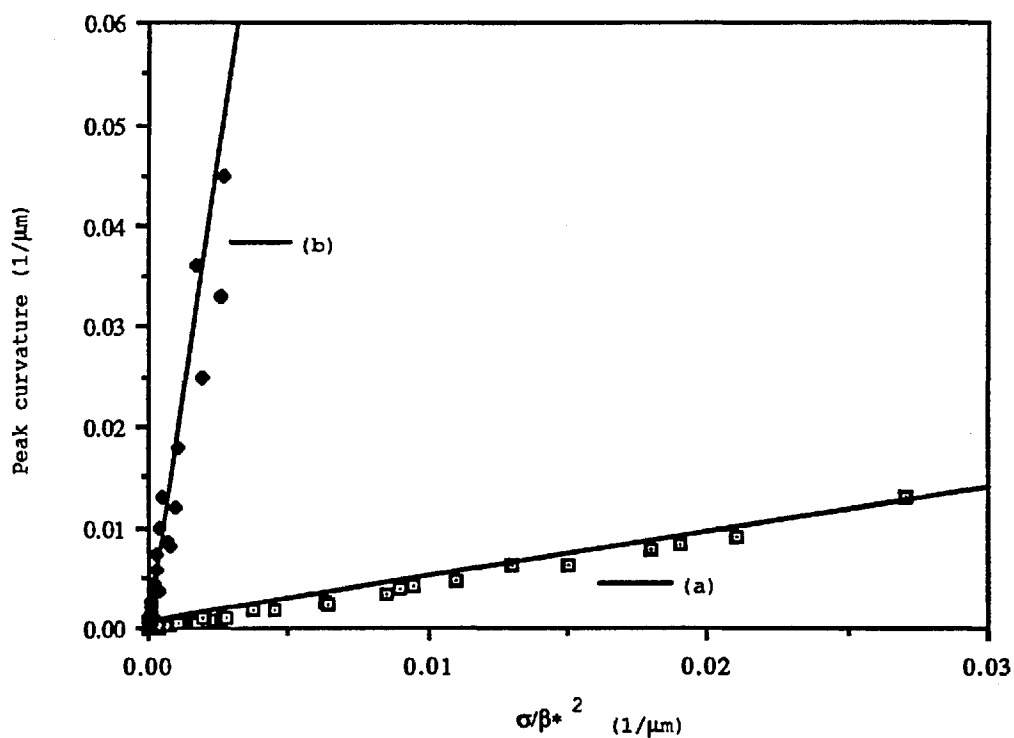


Figure 2.17 Variation of peak curvature with σ/β^{*2} . (a) $\tau=2.3\beta^*$ and (b) $\tau=0.23\beta^*$. The straight lines give results derived by Whitehouse and Archard.

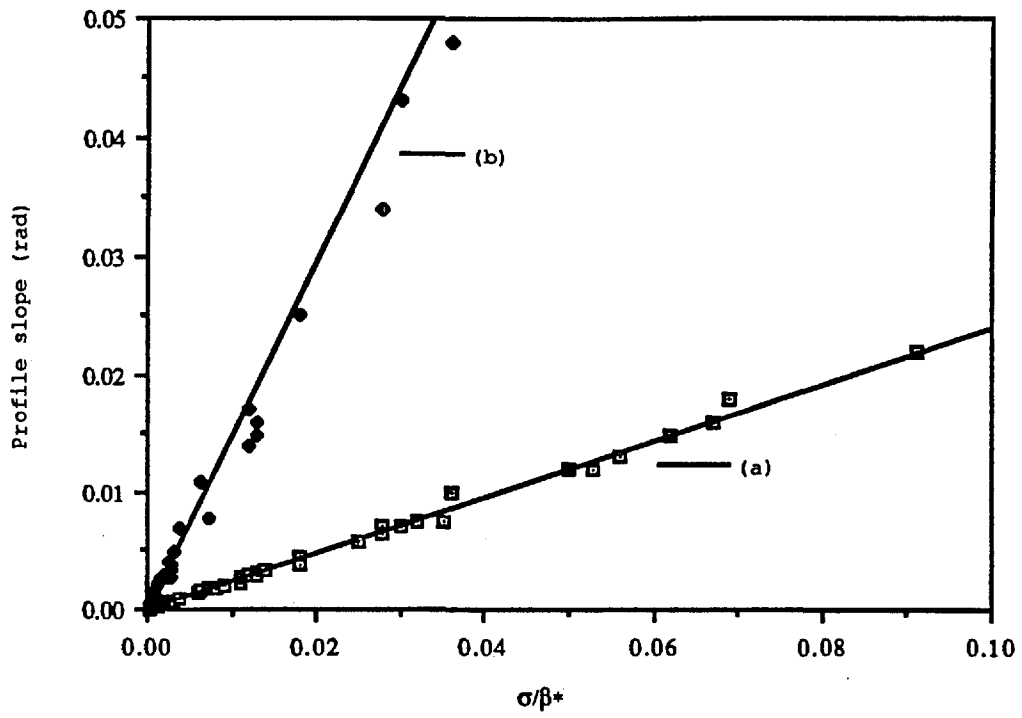


Figure 2.18 Variation of absolute mean profile slope with σ/β^* . (a) $\tau=2.3\beta^*$ and (b) $\tau=0.23\beta^*$. The straight lines give results derived by Whitehouse and Archard.

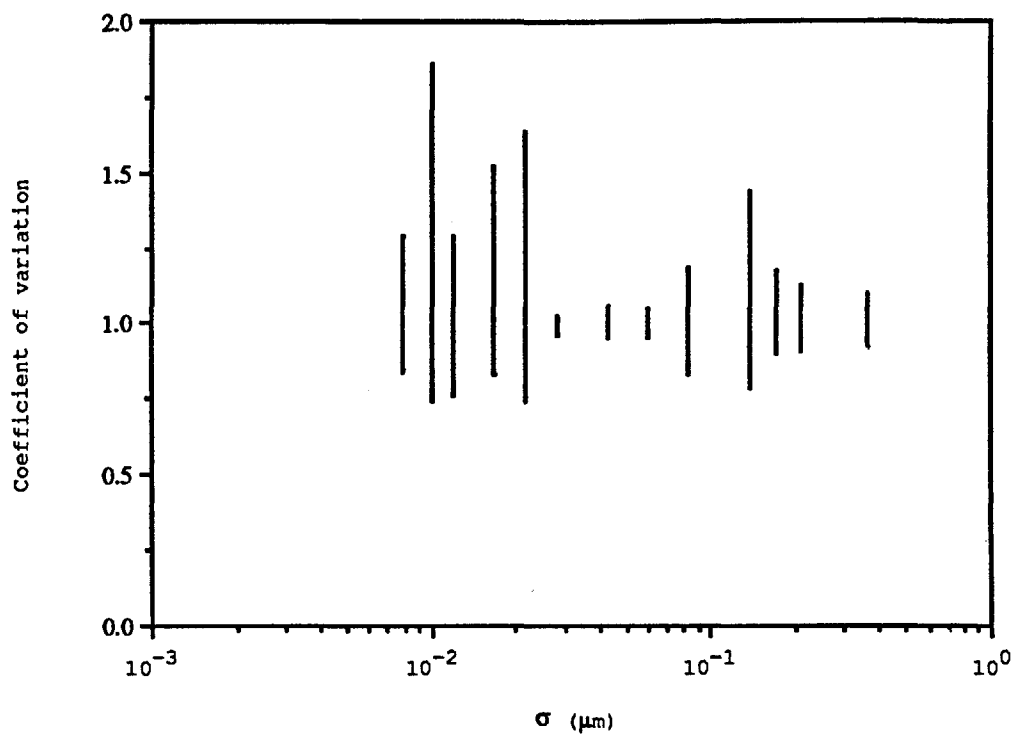


Figure 2.19 Scatter in roughness for 15 abraded surfaces.

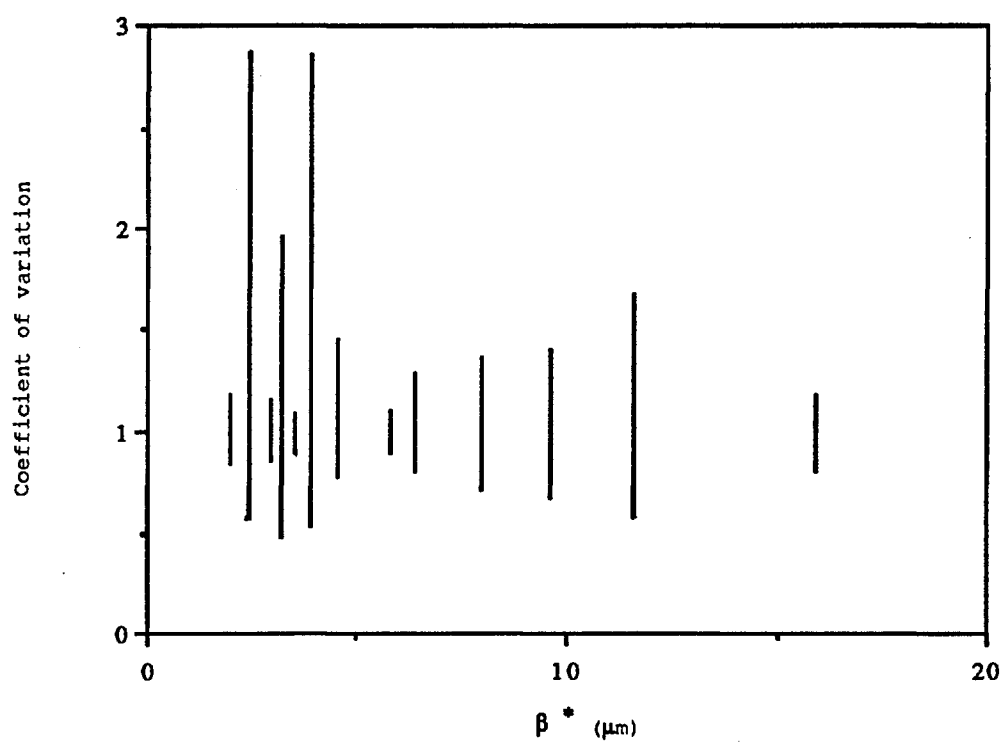


Figure 2.20 Scatter in correlation length for 15 abraded surfaces.

2.3.5 Sources of Error

This refers to errors due to the stylus instrument and quantization errors.

2.3.5.1 Effect of Stylus Size

The profile recorded by a stylus instrument is the locus of some points on the stylus. The stylus is of finite dimensions and the locus depends on the stylus tip geometry. It is suggested that the stylus may not reach the bottom of some valleys or follow all peaks accurately and hence will produce a distorted record of the surface. For steep-sided features this may well be the case. However, evidence suggests that results obtained using stylus instruments give reasonably accurate values for surface slope on common engineering surfaces (Huffington 1970).

Radhakrishnan (1970) investigated the effect of stylus size on real engineering surfaces by computing the path followed by different stylus size. The results indicate that for many surface parameters, quite large styli may be used before any serious change in roughness parameters occur. Whitehouse (1974) also investigated the effect of stylus size on getting the true measurements of a grounded surface. He concluded that the standard stylus (2 μm radius) does not appreciably reduce the true value of σ of the surface.

The roughness of the surfaces in the present work are relatively small and the profile slopes are much smaller than the tip flank angle (90° inclusive). Moreover measurements were made using a high resolution stylus of about 1 μm radius. Therefore, errors due to stylus size are considered to be negligible.

2.3.5.2 Effect of Stylus Load

The recorded profile always represents a deformed profile of the surface due to the

stylus load. Thomas calculated the elastic deformation of a steel surface to be $8.3 \times 10^{-4} \mu\text{m}$ for a stylus load of 100 mg; this figure is much smaller than the vertical resolution and therefore should not affect the measurement. In addition, the surface is everywhere deformed by approximately the same amount, thus the output of the instrument will be a reasonably true profile displaced downward by a constant small distance.

Although the stylus load is small the area of contact between the stylus and the specimen surface is also small, hence there is always the possibility of local plastic deformation. On softer materials, such as lead and aluminium, a visible track is left after the traverse of the stylus. The question is, 'Does it affect the measurement?'. In reply, Williamson (1967) used an aluminium specimen to show that although visible damage occurs the effect on the measurement is negligible, claiming that the furrow left behind is less than 1 micro-inch ($0.025 \mu\text{m}$) deep.

2.3.5.3 Effect of Sampling

Sampling too often results in highly correlated and redundant information, whereas sampling too rarely may cause aliasing to occur. Aliasing results in the higher frequencies in the profile appearing as a lower frequency in the digitised signal. In surface texture measurements, the finite dimensions of the stylus limits the choice of short sampling interval. For the standard stylus sampling below $2 \mu\text{m}$ would not result in additional information. On the other hand, the limited traverse length of 10 mm of the Talysurf 4 means that sampling intervals greater than $20 \mu\text{m}$ would result in fewer than 500 data points being collected with a corresponding loss of statistical significance.

2.3.5.4 Quantization

This refers to the accuracy of the conversion of the original signal into digital form. The output signal in three decimal digits is obviously more accurate than two decimal digits but noise on the signal can nullify this apparent advantage. Whitehouse (1978) considered the effect of quantization on the digital assessment of peak parameters, reporting that the number of peaks detected reduces as the number of quantization levels decreases. Quantization errors can also lead to erroneous values of curvature and slope. In practice, three decimal digits have been proved sufficient, with errors produced by other effects usually being larger than those from quantization. In the present system, measurements were normally recorded using between one and two thousand quantization intervals, hence errors due to quantization were small.

2.3.5.5 Other Sources of errors

Sources of error can also occur in the electronic part of the instrument, and through the dynamic response of the stylus.

All electrical components, such as transducers, amplifier, etc, have finite limits on non-linearity and drift are affected by temperature. The geometry of the stylus arm is also influenced by temperature. For a five degree Celsius change a stylus arm of 10 cm would suffer a change of length of 6 μm , i.e. greater than the nominal stylus dimension. For the present work, the time taken over a profile measurement is typically 1-2 minutes and it is very unlikely that a temperature change of 5 degrees would occur during that time. In general, these errors are small compared with the mechanical limitations of the instrument.

2.4 Surface Assessment

Most surface parameters, for example, σ and β^* are not intrinsic properties of a real surface. They depend on the scale of measurement and are particularly sensitive to the bandwidth selection. The decision of which scale of size to investigate depends on the physical problem involved. For contact applications, Thomas and Sayles (1978) introduced the concept of functional filtering. The procedure involves evaluating a mean line, called the low-pass output, this follows the general direction of the profile, and then subtracting the original profile from this reference mean line. The resultant profile with the low-pass output filtered out is called the high-pass output. The mean line may be obtained by graphical, electrical or digital methods.

By graphical methods, the mean line is generally achieved by the M or E-system. In the M-system, a series of mean lines are positioned to follow the general trends of the profile for a number of sampling lengths. Then each section is redrawn with its mean axis in line. The resulting profile is the original profile with the waviness filtered out.

In the E-system or envelope system, the reference line is defined by the loci of the centre of a circle rolled along the profile. It is suggested that a radius of 25 mm be used for roughness and 250 mm for waviness. The mean reference line or low pass output is defined as the contacting envelope displaced downwards by a distance such that the areas enclosed by the effective profile above and below it are equal. The main criticisms of the M or E-system are based upon their ambiguity and inaccuracy: the reference mean line must be left to the subjective judgement of the operator. This method obviously creates ambiguity and usually takes the analyst a considerable time to construct.

Electrical filtering is usually employed by instrument manufacturers by using a relatively simple capacitor-resistor network (2C-R). This technique has several

disadvantages, the roll-off characteristics are poor and some features longer than the nominal cut-off length are retained in the filtered profile. Furthermore, phase distortion can affect the local mean with which the profile is compared (Whitehouse 1967/68).

Digital filtering has advantages over graphical filtering and electrical filtering in its flexibility and accuracy. It allows sophisticated analysis to be carried out to control the size of features or to limit the analysis to the problem by a suitable choice of effective sampling interval. Moreover, M-system and E-system filtering can also be achieved by digital methods. Other digital filtering techniques have been suggested such as moving average filtering and cubic-spline filtering.

In the moving average filtering method, the low-pass output is calculated by averaging over a sample length as it is moved across the surface profile.

In the cubic-spline filtering method, the filter operates by splitting the profile into a series of cut-off lengths and then simply fitting a cubic spline curve through the mean of each cut-off length which is the low pass output.

In order to obtain an effective filtering method in assessing a surface before the relevant parameters are calculated, the digital filtering methods of the E-system, moving-average and cubic spline are studied and compared. The study is limited to the contact of a ball on a nominally flat rough surface, as this is directly relevant to the present work.

Any rough surface is known to consist of a range of asperity shapes and sizes, thus it is apparent that in contact, the spatially smaller asperities will persist and constitute the effective roughness of the contact, whereas the very broad wavelength features take no part in the contact but tend to dominate the numerical height characteristics and consequently dominate the numerical values of roughness. Therefore, the high-pass output, with the irrelevant longer wavelength features filtered out, is what is required for surface

assessment.

In the E-system, a radius of envelope is required to generate a mean line for waviness. In the case of a ball sliding on a nominally flat rough surface, the radius of the envelope may be taken as that of the ball, thus giving the effective surface roughness of the ball relative to the nominal profile. Figure 2.21 shows a typical profile superimposed with the low-pass output given by the E-system with an envelope radius 6.35 mm.

In the moving average and cubic-spline filtering methods, a cut-off length is required such that the mean value of a series of cut-off lengths can be calculated and joined by a smooth curve. It has been suggested that the nominal contact width should be used as the cut-off length because the wavelength features within this nominal contact are likely to be important. Figures 2.22 and 2.23 show typical profiles, together with their low-pass outputs, produced by moving average and cubic spline filters set at a cut-off length of 0.15 mm, which is the Hertzian contact width involved in most tests reported here.

2.4.1 Transmission Characteristics

It is instructive to know the efficiency of each of the three filtering methods in removing the unwanted longer wavelength features. Here, the efficiency or transmission characteristic of each filter is studied. A set of sinusoidal surfaces and random rough surfaces with different vertical height and spatial structures are subjected to the various filters. The cut-off length for the cubic-spline filter and moving-average filter are taken as 0.15 mm and the radius of envelope for E-system is 6.35 mm.

Figure 2.24 shows the transmission curve for sinusoidal profiles with a constant amplitude of $0.3\text{ }\mu\text{m}$. The cubic-spline filter and moving-average filter have a similar

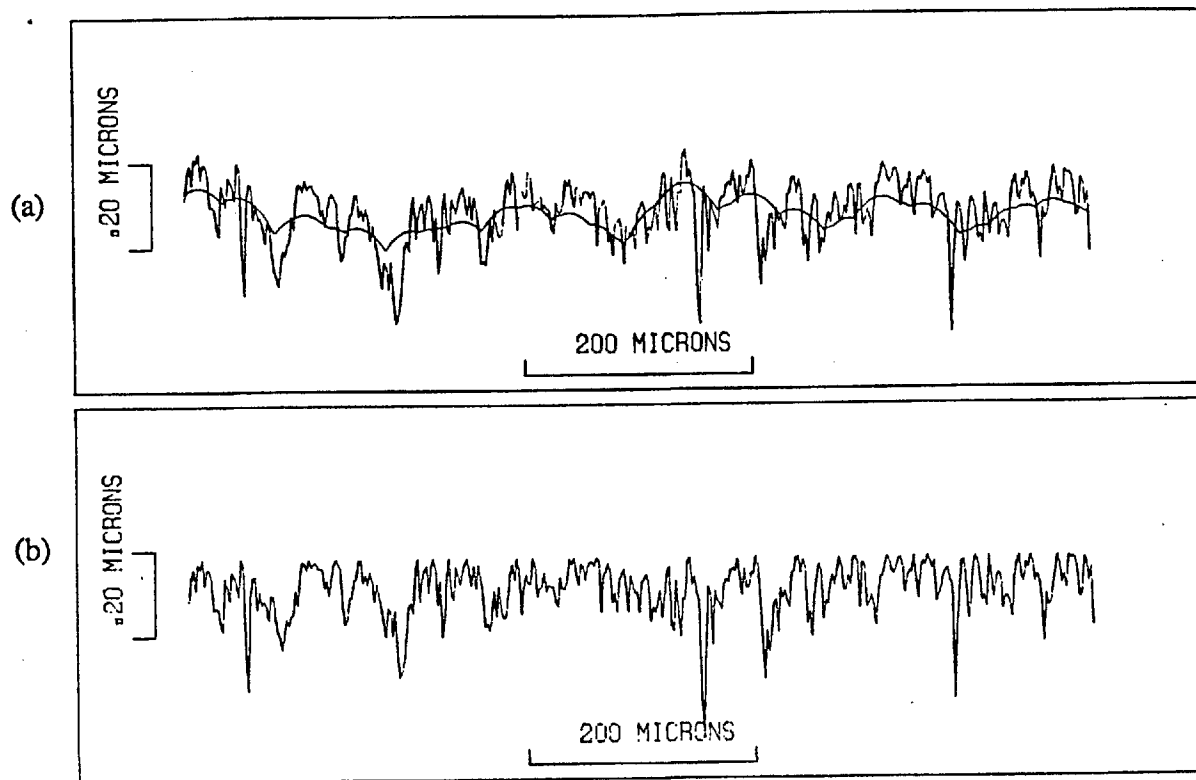


Figure 2.21 Surface profile with (a) low-pass output by the E-system filter, (b) high-pass output.

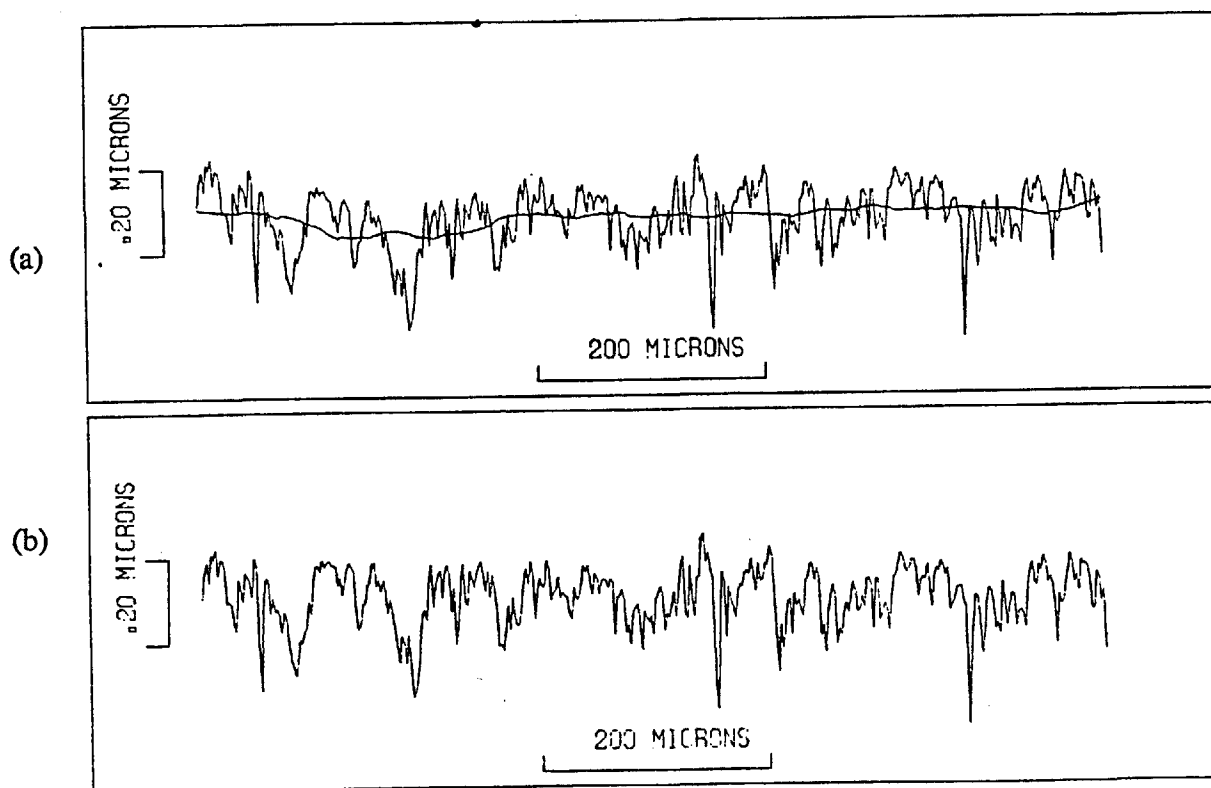


Figure 2.22 Surface profile with (a) low-pass output by the moving-average filter, (b) high-pass output.

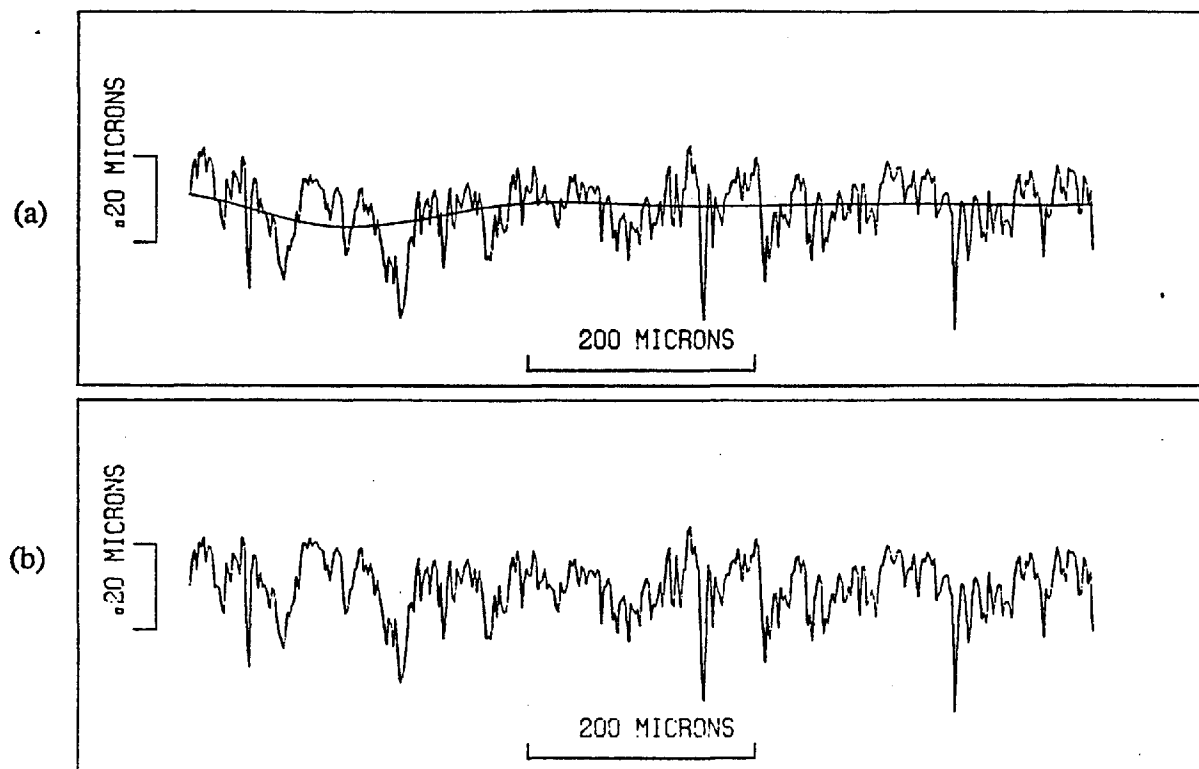


Figure 2.23 Surface profile with (a) low-pass output by the cubic-spline filter, (b) high-pass output.

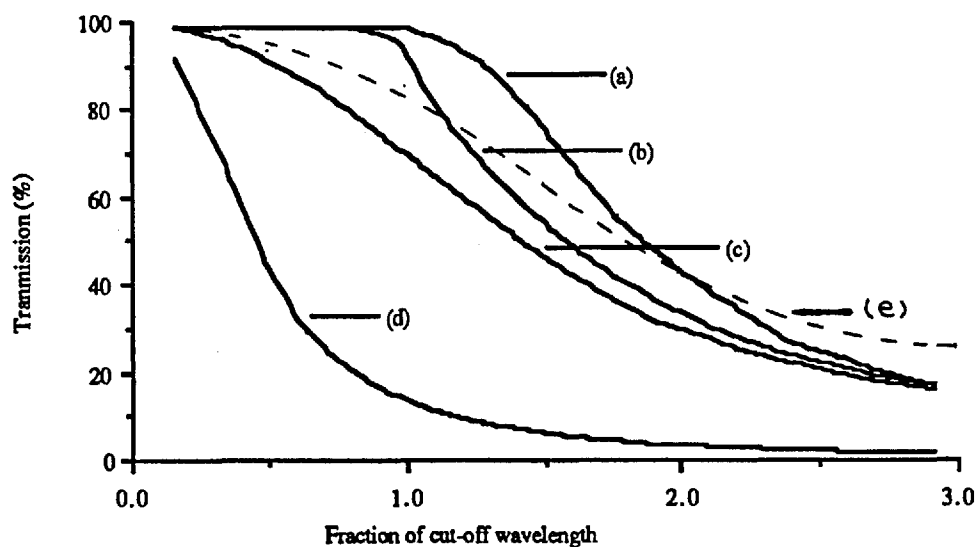


Figure 2.24 Comparison of transmission characteristics for various filters on sinusoidal profiles.

(a) Cubic-spline filter

(b) Moving-average filter

(c) E-system filter

(d) as (c) but amplitude of profile is reduced to $0.03 \mu\text{m}$

(e) 2-CR filter

transmission curve of unity up to the filter cut-off wavelength and then decreasing with increasing wavelength. The E-system filter has a transmission curve of unity for very small wavelengths and then it decreases less quickly as the wavelength increases. By reducing the amplitude of the sinusoidal profiles to 0.03 μm , the transmission curve of the cubic-spline and moving average filters remains unchanged but the transmission curve of the E-system filter reduces significantly, as shown in Figure 2.24(d).

The transmission behaviour of the E-system filter is as expected when the profile curvature become closer to that of the ball as the wavelength, λ increases or the amplitude, Δ decreases. For example, the radius of profile curvature, R , at the crest or the valley of a sinusoidal surface is given by $R = \lambda^2 / [\Delta \cdot (2\pi)^2]$. With $\Delta = 0.05 \mu\text{m}$ and $\lambda = 100 \mu\text{m}$, an approximate value of radius is $R = 5 \text{ mm}$ which is very close to that of the ball. Apparently, the reference line will follow closely along the surface resulting in a lower transmission high-pass output.

It should be noted that the reference line given by the E-system filter represents the real situation of a ball sliding on a nominally flat surface. The reference line has been shown to be dependent on the wavelength as well as the amplitude of a sinusoidal surface. However, the transmission curves of the cubic-spline and moving average filters appear little affected by changing the amplitude of the sinusoidal profile and they appear only dependent on the cut-off wavelength. Therefore, the high pass output of a cubic-spline filter or moving average filter is likely to retain the unwanted longer wavelength features, particularly on very smooth surfaces, which will result in the effective roughness and wavelength being likely overestimated. Thus in the case of a ball contact on a sinusoidal profile, the E-system filter is preferable to the use of cubic-spline or moving average filters.

For a random rough surface, the profile curvature has been shown to be related to the ratio σ/β^* (Whitehouse & Archard 1970). If σ is sufficiently small or β^* is sufficiently

large such that the profile curvature is close to that of the ball, then the low pass output should follow closely that of the surface profile. The effectiveness of each filter on a smooth surface is shown in Figure 2.25 to 2.27. The low-pass output of the E-system filter follows closely to the longer wavelength features of the surface, indicating that such wavelength features should be filtered out prior to obtaining the effective roughness. However, the cubic-spline filter and moving average filter appear to retain these longer wavelength features, thus giving higher σ and β^* values.

To examine filter characteristics, a set of 15 abraded surfaces of various roughness were subjected to each filter. The results are shown in Figure 2.28. For rough surfaces, the filtered σ and β^* given by various filters are very close. For smooth surfaces, the cubic-spline filter and moving average filter yield higher σ and β^* values. This is most likely due to the inefficiency of the filters at removing the unwanted longer wavelength features, particularly on smooth surfaces, for the reasons explained above. Therefore the E-system filter is chosen as preferable in assessing surface data in the present work.

The criticisms for the E-system are based upon the difficulties in programing by computer, or positioning of the geometrical profile by graphical method such that the areas enclosed by the effective profile above and below it are equal. It is also said that the mean line, composed of a succession of intersecting arcs, is mathematically discontinuous.

With the advent of small but powerful micro-computers, these problems can be overcome and a program was successfully written to compute the output of a moving ball passing along any recorded profile. A point of the loci at a particular position is determined by the first contact of the ball on the profile. The time taken to find the low-pass reference line by the E-system was found to be only a few seconds longer than that of the cubic-spline filter. Although the mean line is mathematically discontinuous, the σ and β^* for rough surfaces compare well with values obtained from the cubic-spline and moving average filters. So far, the only difficulty experienced is that the geometric profile is

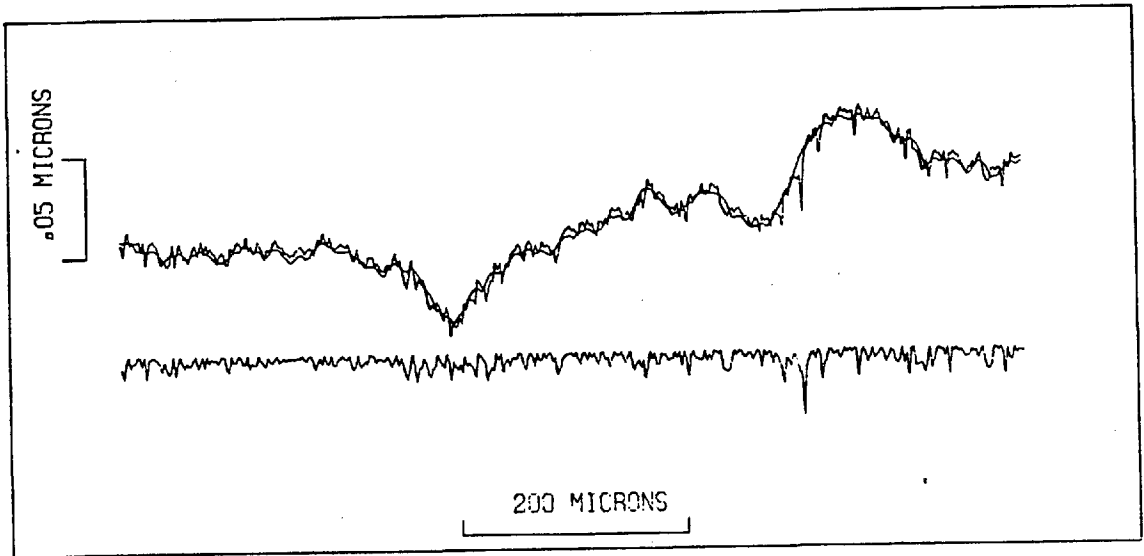


Figure 2.25 Smooth surface profile with the low-pass output via the E-system filter.

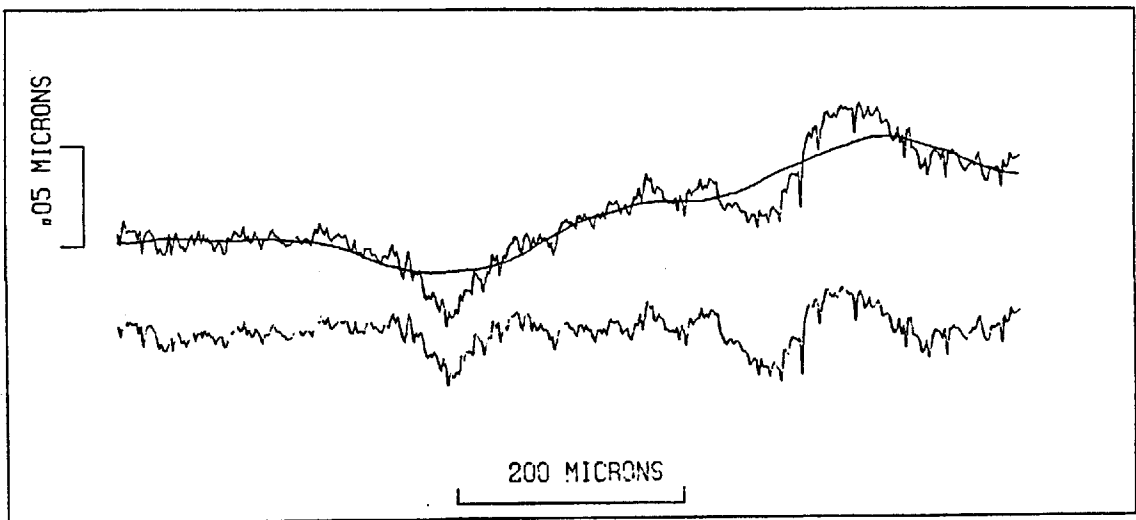


Figure 2.26 Smooth surface profile with the low-pass output via the moving-average filter.

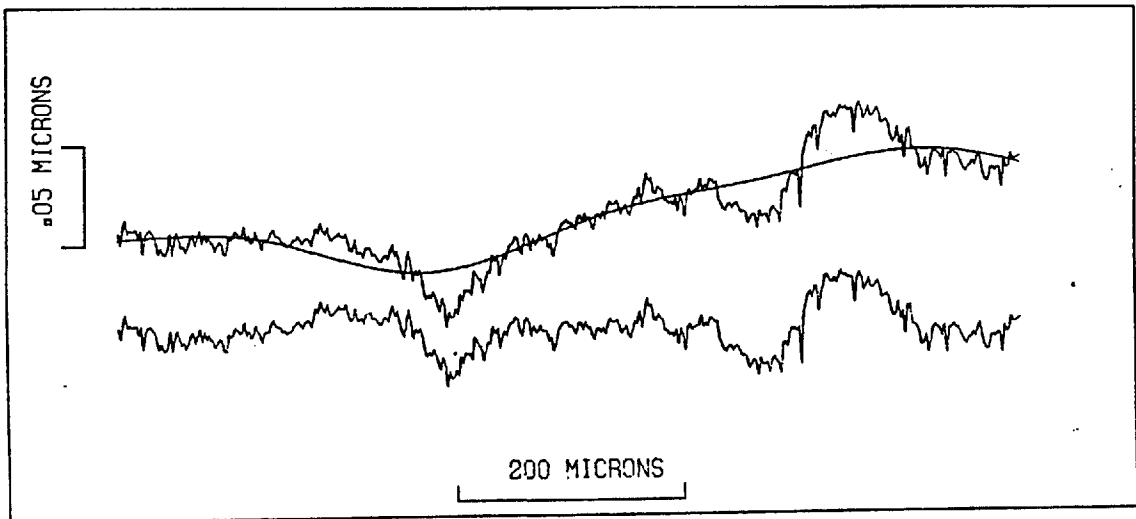


Figure 2.27 Smooth surface profile with the low-pass output via the cubic-spline filter.

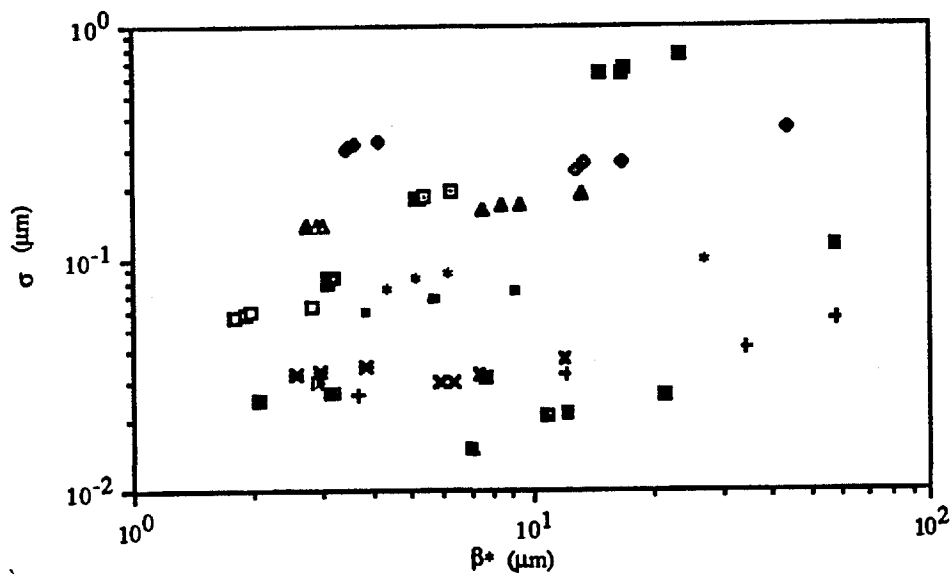


Figure 2.28 Comparison of the resultant σ and β^* values for each filter. The same symbol is used for each surface and appear in clusters of 4, with;

- right: original surface
- second right: cubic spline filter
- second left: moving average filter
- left: E-system

occasionally influenced by an asperity which is much higher than its neighbouring asperities, particularly on rough surfaces, thus giving a distorted low-pass curve. However, this can be overcome by computer programming which ignores any asperities which are unusually higher than their neighbours.

2.5 Conclusions

Abraded surfaces used in the friction tests described in Chapter 7 and 8 have been studied. Their surface heights distribution can be reasonably regarded as possessing a Gaussian distribution. The mean peak height, mean peak curvature and absolute profile slope can be related to the σ and β^* and agree well with the exponential model derived by Whitehouse and Archard using the assumptions that the surface heights and their first and second derivatives have a Gaussian distribution.

Methods of surface assessment have been discussed. For a ball contact on a nominally flat rough surface, the E-system filter is shown to be preferable to the use of the moving average or the cubic-spline filters to separate the waviness from roughness. It is shown that by applying a moving average or cubic-spline filter to the contact problem of a ball on a rough surface, the actual surface roughness is likely to be overestimated due to the filter's inefficiency at removing the longer wavelength features which are less significant to the contact.

CHAPTER 3

ROUGH SURFACE CONTACT : LITERATURE REVIEW

3.1 Introduction

The actual area of contact of microscopically rough surfaces and the changes that occur under load and relative motion are the most important parameters in understanding many tribological phenomena, such as stiffness of joints, wear, adhesion, friction force, frictional heating, thermal and electrical contact resistance and fluid leakage. It is understood that when two surfaces are brought together surface roughness causes contact to occur at discrete contact spots. The spatial distance between contact spots and the sizes of the contact spots are usually random because the surface roughness itself is random. The true contact area is the summation of the individual contact spot areas. Deformation occurs in the region of the contact spots, establishing stresses which oppose the applied load. The mode of surface deformation, whether it deforms elastically or plastically is of considerable importance in understanding surface mechanics.

There are two basic laws of friction which were first introduced by Leonardo da-Vinci (1452-1519) and rediscovered by Amonton in 1699 that the frictional resistance is proportional to the load and that it is independent of the nominal area of the sliding surfaces. On the other hand, friction between sliding surfaces is thought to arise from local adhesion occurring at areas of intimate contact associated with asperity summits. It was assumed that the friction force was the result of shearing these adhesive junctions. For these statements to be true and consistent with Amonton's laws, then the real area of

contact must be also proportional to load. Most early work on surface contact models was directed toward providing an explanation to Amonton's laws of friction by showing this load-real area proportionality existed for most rough surfaces.

Hertz (1881) was perhaps the first to treat analytically the case of elastic spheres in contact, and he showed that the size of the single contact area was a function of the load, radii of curvature, and elastic moduli of the solids. His work has stood the test of time and over the past one hundred years has provided the basis of much of our understanding of contact mechanics (Johnson 1982). For example, in Greenwood and Williamson's (1966) contact model, the contacts were assumed to occur on spherical asperity summits of constant radius. In this way the Hertz equations could be used to define the load-compliance and area-compliance relationships for each individual contact.

In the pioneering work of Bowden and Tabor (1950), they argued that plastic flow would occur at the asperities summits, and that the load was equal to the real area of contact A_r multiplied by the flow pressure of the softer material P_m . In other words, the real contact area can be determined by the load divided by the bulk hardness of the softer material regardless the shape of the asperities and of the size of the surfaces. This provided a simple explanation of Amonton's laws which was supported by electrical contact resistance measurements.

Archard (1957) suggested that although surface features may be plastically deformed at the first encounter, on subsequent encounter they will be elastic. Thus, in the process of running-in, local asperity contacts must resist the same load elastically for subsequent contacts. He then proved practically and theoretically that Amonton's law would still be obeyed even for elastic deformation. He used for his analysis a "scale of size" argument. By postulating a profile as the superposition of different scales of smooth capped asperities and he was able to employ Hertzian elastic contact mechanics to show that

it was possible to get a proportional relationship between real area of contact and load under elastic conditions. A general relationship for load and area of the form,

$$A = kF^n \quad (3.1)$$

can be applied where k and n are constant. For a smooth sphere, n is $2/3$. When one set of spherical asperities were superimposed $n=8/9$ and when two sizes are superimposed $n=26/27$ and so on. As the complexity of the model increases the real area of contact becomes more closely proportional to the load.

Some workers determine the area of contacts, the sizes and their distribution experimentally. The most commonly used methods are thermal and electrical conduction, optical methods, interferometry and profilometric approaches.

The realisation that surface asperity heights can often be regarded as possessing a Gaussian distribution has drawn attention to the use of random process theory as a means of deriving of the load-area relationship. Greenwood and Williamson (1966) were among the first to take into account the statistical nature of the asperity distribution on surfaces. Since then, stochastic approaches with more mathematically complex random process models, for plastic and elastic contact have been developed. The mode of asperity deformation can be predicted using the plasticity index (see Section 3.3).

Over the last few years, numerical techniques have been developed and used on rough surface contact problems. This approach provides a deterministic solution to rough surface contact stresses and areas. The technique highlights the deformable contact geometry of rough surfaces under loading. It provides useful information on the number of contacts, their sizes and distributions, and the spacings between contacts, which are thought to act as lubricant traps etc.

In this chapter, the approaches available for determining the real area of contact are reviewed and their merits discussed. The mode of deformation and the criteria for plastic deformation are also discussed.

3.2 Real Area of Contact

3.2.1 Experimental Approach

3.2.1.1 Thermal and Electrical Conduction

One of the earliest methods of obtaining the real area of surface contact was from measurements of electrical contact resistance, R (Bowden & Tabor 1950; Holm 1958), using the relationship $a \propto R$ and $n \propto 1/R^2$, where a and n are the radius and the number of contact spots respectively. By this method, errors are likely to occur owing to interfacial films of unknown and variable resistivity and current convergence patterns of unknown geometry etc. However, the method is useful in that it can be used for both static and sliding contacts.

The above arguments can be similarly applied to the thermal resistance between contact spots (Boeschoten & van der Held 1967; Mustacchi & Giuliani 1965). This method is less affected by the presence of intervening oxide layers, however, a correction must be made for the heat transferred across the contact interface via the air gaps.

3.2.1.2 Optical Methods

Bowden and Tabor (1950), Holm (1958), Dyson and Hirst (1954) all examined the

use of optical methods allowing the direct observation of the contact area while the load was still being applied. Uncertainties in the results were shown to exist due to the limitation on lateral magnification inherent in interferometric systems. Also the observed optical contact area may not be equal to the actual mechanical contact area. In extreme cases, e.g. in the contact of very smooth surfaces, an overestimation of up to 30-40% had been reported (Kragelskii 1965). These methods usually produce qualitative information only and analysis of the experimental data would require tedious digitizing from microphotographics.

3.2.1.3 Internal Reflection

This method is based upon the disruption of total internal reflection, (Kragelskii & Demkin 1960) when a rough surface is brought into contact with a triangular glass prism, if a parallel beam of light is directed at the prism face forming the contact at an angle such that there is total internal reflection. At the points of contact total internal reflection is destroyed and the contacts become visible in the reflected light. The contact area can thus be determined visually, which is usually achieved by photographs.

3.2.1.4 Nomarski Interferometry

Surface contours of equal slope are revealed in different colours by means of Nomarski interferometry contrast illumination (Uppal et al 1972). When a rough surface is viewed after it has been deformed by a flat, all the contact areas lying in one plane will be covered by a fringe of the same colour. The number and size of contacts can be determined by employing colour film and a projection microscope. This method is reported to have obtained a vertical resolution of $15\text{\AA}$.

3.2.1.5 Neutrographic Method (Johannet 1972)

A parallel beam of neutrons is directed at the pair of contacting surfaces. The interface between these surfaces is filled with a neutron-absorbing material except at the real contact areas where this material has been squeezed out. Thus, the non-contact area becomes opaque to the neutron beam and the stream of neutrons coming through the contact areas impinges on a converter, which is a metal plate that absorbs the neutrons and re-emits γ and β rays. These rays are exposed on a photographic medium to give a radiographic image of the contact interface.

3.2.1.6 Computer Simulation

This method provides a means of predicting contact based upon measured profiles or surface maps (Williamson 1967/68, Sayles 1978b). A surface profile is digitised, usually from a stylus instrument, through an analogue-to-digital conversion into a digital computer. A surface can be mapped by making multiple parallel traversals across it. Digital maps of pairs of different surfaces can be brought together to simulate contact inside the computer. Assuming the asperities in contact deform plastically with a total loss of volume, the contours of contact can be produced by adding height measurements of a number of pairs of real surfaces. The resulting contour maps can be analysed statistically to give contact parameters for various mean plane separations of the rough surfaces.

3.2.1.7 Relocation Profilometry

This method allows the plastic area of deformable bodies at various stages of load to be studied. (Williamson & Hunt 1968; Uppal & Probert 1973). Each stage is relocated on the relocation table and a profile is recorded along the same track. Thus, the same section of the rough surface is traversed each time which enables the effects of load on that particular section to be observed. The bearing area of the profile is found from the section of profile which has been flattened.

3.2.2 Indentation Models

When two bodies are brought into contact under heavy load, it is argued (Bowden & Tabor 1950) that the contact pressure will be of the order of the bulk hardness of the softer of the two solids in contact. The real area of contact will be equal to the load divided by the bulk hardness. One would expect that a soft rough surface pressed with great force, of the order of ^{its} Hardness against, a hard smooth one would become smooth. However, Moore (1948) pressed a smooth steel cylinder against a face-turned copper plate and found that the ridges in the copper remained clearly visible in the indentation profile even at the heaviest loads which he could use.

By profilometry and taper sections of the deformed specimen, the area of contact was shown roughly half the nominal area of the indentation. Thus, the ridges were able to withstand a local pressure twice that sufficient to produce flow in the bulk metal beneath them. Moore suggested that this was attributable to the work hardening of the asperities during crushing. However, asperity persistence, as it was called, also occurs on materials which have been so severely worked that no further significant work hardening can be reasonably expected. Moore's work stimulated several workers to investigate, both experimental and theoretical, the problems of asperity persistence between rough surfaces (Williamson & Hunt 1972; Childs 1973; Chivers et al 1974).

Williamson and Hunt (1972) extended Moore's work by using various materials and conditions. They used a hard smooth ball against a rough surfaces of indium, which is self-annealing and does not work harden. They concluded that indium asperities exhibited almost as much resistance to crushing as did Moore's work-hardenable copper ridges. Thus, concluding that work-hardening was not necessary for the persistence of asperities.

Williamson and Hunt further investigated the degree of asperity persistence on two classes of metals, namely homogeneous solids with uniform hardness, and solids with hardened surface layers. They showed that for homogeneous solids, the degree of contact which is the ratio of real contact area to the apparent area was $50.8 \pm 0.7\%$, and for solids with hardened surface layers, between 25 to 35%.

Williamson and Hunt's work was followed up by Pullen and Williamson (1972) who provided a theory for the plastic contact between solids based upon the assumptions that the total volume of metal is not changed by plastic deformation, and that the displaced material reappears as a uniform rise in the non-contacting surface. Using an energy-balance argument they derived the following relation between W , A_n and A_p

$$\frac{W}{A_n} = p \cdot \frac{\alpha}{1 - \alpha} \quad (3.2)$$

where W is the load, p is the mean pressure and α is the ratio of plastic area A_p to nominal area A_n . The equation predicts that for α to tend to unity, (i.e. contact area $\rightarrow$ nominal area) the load must tend to infinity so that the persistence of asperities can be partly explained on the basis of Pullen and Williamson's theory. They also performed indentation tests using a flat ended punch pressed on to a rough surface formed on the end of a rod of aluminium which was constrained within a steel collar. Relocation profilometry was used to measure the plastic deformation which compared well with their theory. They went on to consider the limit of A_p in the conditions of a Brinell test. By putting p equal to H at heavy load and

from equation (3.2). they obtained $\alpha=0.5$ which is the experimental value for homogeneous solids obtained by Williamson and Hunt.

Child (1973) used slip-line field theory (see Hill 1950) to predict the plastic contact area under loading. Theoretical results were obtained by modelling surface roughness as a uniform array of wedge-shaped ridges. He then performed indentation tests using a flat ended punch on three machined surfaces with a single ridge, arrays of ridges, and pyramids respectively. The experimental results generally compared well with the theoretical prediction. However, the maximum value of A_r/A_n he obtained was 0.8 which contradicts that obtained by Williamson and Hunt.

Chivers et al (1974) used a similar slip-line field solution to investigate the real plastic area under loading. He used a flat punch pressed on to cold worked copper constrained in a thick walled cylinder. One specimen was machined to produce a uniform array of wedges and the other specimen abraded to produce a random surface texture. Their experimental and theoretical predictions were similar to those obtained by Childs.

Experimental results reported by Wanheim (1973) and Demkin and Izmailov (1975) also closely followed the theory by Childs. In a later paper by Childs (1977), he summarised the results of previous work and presented the results as $P_r/2k$ against A_r/A_n , where P_r is the mean normal stress acting over A_r and k is the metal's shear flow stress. The experimental results of Pullen and Williamson (1973) and Uppal and Probert (1972) on bead blasted surfaces agree with the theory only at low values of A_r/A_n . At higher values, $P_r/2k$ increases with A_r/A_n more rapidly than expected. Childs attempted to explain why the results obtained by Uppal and Probert's (1973) and Pullen and Williamson deviated from his own by postulating two possibilities which were: the theoretical model was either geometrically over simple or both Uppal & Probert and Pullen & Williamson failure to consider elastic distortions of the punches and rough surfaces in their experiments

resulting in the underestimation of measured A_r/A_n and, in turn, the overestimation of contact pressures. These possibilities were studied experimentally as described below.

Childs investigated the influence of the nature of the surface finish on asperity resistance. He used round ended cylindrical punches pressed on work-hardened mild steel blocks which were machined by sand-blasting, abrasion and grinding. He found that the nature of the surface finish only influences an asperity's resistance to deformation to a minor extent.

Childs also investigated the effects of elastic deformations of the punched and rough surfaces, and work-hardening effects. Using elasticity theory (Timoshenko & Goodier 1951), for a circular flat ended rigid punch pressed on a semi-infinite elastic solid, he showed that the elastic compliance was in the order of the measured plastic compliance. The area of contact measured using relocation techniques is based on the permanent plastic deformation, thus it ignores the effect of elastic conformity of the punch between the rough surface, and hence the area obtained, is likely underestimated. The pressure P_r calculated using the load divided by the area is in turn overestimated. In fact, in his earlier paper, Childs (1973) used a flat punch pressed against a single ridge surface, an experimental condition similar to that of Uppal and Probert, and he found the real area was not limited to the ridge top but also to the rest of the work piece surface. The punch would have touched the surrounding surface by elastic compression of the ridge.

Childs considered the influence of work-hardening on the dependence of $P_r/2k$ on A_r/A_n . He concluded that the final shear flow stress of the material due to strain hardening should be used in order to conform to the results due to theoretical prediction, and if the initial shear flow stress at zero plastic strain were used, the value of $P_r/2k$ would have been overestimated. In conclusion, Childs claimed that Uppal & Probert's and Pullen & Williamson's results could be reconciled with his theory by considering the elastic

distortion of their punch and of the hinter-land beneath their rough surface coupled with a suitable choice of shear flow stress.

Pullen and Williamson's energy balance approach and Child's slip line field theory approach both ignore the elastic behaviour of the metal. However, the elasticity of real materials plays an important part in the plastic indentation process. Generally, at the onset of plastic deformation, the plastic zone is first contained by material which remains elastic and the material displaced by the indenter is accommodated by an elastic expansion of the surrounding solid. As the indentation becomes more severe, the plastic zone expands until it breaks out to the free surface and the displaced material is free to escape by plastic flow to the sides of the indenter. With a rigid-plastic solid the material displaced by the indenter appears in the piled-up shoulders, but with an elastic-plastic solid this is not the case. Most of the displaced material can be accommodated by radial expansion of the elastic hinterland (Johnson 1985). Therefore, Pullen and Williamson's assumption of the displaced material reappearing as a uniform rise in the non-contacting surface is only partially true.

3.2.3 Stochastic Approach

Greenwood and Williamson (1966) assumed that peak heights were distributed in a Gaussian manner and that the summits have a constant radius. With this model they derived expressions for the expected area of contact, expected total load and expected number of contact spots as a function of separation. They concluded that a near direct proportionality was found for load-v-area and load-v-number of contact spots. The average contact size was also shown to be almost independent of the load.

Tsukizore and Hisakado (1965,1968) used a model of the contact between a smooth plane and an isotropic rough surface to derive expressions for the compliance (load-v-penetrating depth), the average clearance and the number and average size of the

contact spots as a function of load, geometry of the surface and properties of the material. The rough surface was modelled as a Gaussian distribution of intersecting cones which deformed plastically. Theoretical results were compared with a number of experiments. Later, Thomas and Probert (1970) compared experimental results with the theoretical results obtain by Tsukizre and Hisakado. The general trends of the theoretical results were shown to agree well with experiment, although some of the comparisons were less conclusive. This was particularly true for the compliance and the number of contact spots-v-applied load. These differences appeared to be due to a combination of difficult experimental measurements and a highly idealised mathematical model which ignored the effects of elastic contact, asperity interaction and strain hardening together with the usual simplifying assumptions regarding the topography.

Mikic and Roca (1974) assumed a Gaussian ordinate height distribution and that the distribution of surface slopes was independent of height. With this model, they obtained the plastic contact area A_p based upon the geometrical overlap of two surfaces and the elastic contact area A_{el} using Hertzian theory. They also obtained the following relationships for any given separation,

$$\frac{A_{el}}{A_p} = \frac{1}{2} \quad (3.3)$$

and

$$\frac{A_{el}}{A_a} = \frac{P}{E' \tan \theta} \quad (3.4)$$

where P is the applied pressure, A_a is the apparent area of contact and $\tan \theta$ is the mean absolute profile slope.

Other workers have approached the problem somewhat differently. Using random process theory, following the work of Longuet-Higgins (1957a,1957b) and Nayak (1971), Bush et al (1975) modelled the summits of an isotropic rough surface as circular paraboloids. They obtained the density of extreme, distribution of peak heights and the distribution of the summit radii in terms of moments of the power spectral density function, m_0 , m_2 and m_4 . At large separation, their model achieved an asymptotic proportionality between load and area, and also produced the same result as the analysis of Mikic and Roca (1974), namely that the elastic contact area is half the undeformed overlap area. The constant of proportionality between load and area was derived as:

$$\frac{F}{A} = \left(\frac{m_2}{\pi}\right)^{\frac{1}{2}} E \quad (3.5)$$

where m_2 is the second moment of the profile power spectral density function, which can be shown to be the variance of the profile slope distribution. (Thomas 1982)

In a separate paper, Bush et al (1978) extended the analysis on strongly anisotropic rough surfaces. The constant of proportionality was similar but involved extra terms requiring elliptical integrals.

$$\frac{F}{A} = \frac{\pi (\bar{\lambda}^2 + \bar{k}^2)^{\frac{1}{2}} m_2^{\frac{1}{2}}}{2^{\frac{1}{2}} \bar{k} (\bar{k})} \left(\frac{m_2}{\pi}\right)^{\frac{1}{2}} E \quad (3.6)$$

By a similar approach Suratkar et al (1976) used the spectral moments of a profile m_0 , m_2 and m_4 , to determine the probability density of land widths as a function of height. Assuming the contact spots are circular, he obtained the total area of contact per unit length A_h at a separation h

$$A_h = \frac{m_0^2}{h^2 m_2} \left(\frac{m_4}{m_2} \right)^{\frac{1}{2}} P_p(h) \quad (3.7)$$

where $P_p(h)$ is the probability density function of peaks above height h .

Stochastic techniques can be used to obtain general results on the amount of contact area, number and distribution of contact spots etc. This type of information is important, however the nature of the models limits the results to average values and yields little information on the behaviour of individual asperities. Such models require the real surface to be modelled as a population of idealised asperity shapes, e.g. spherical or parabolic or conical asperity caps, and therefore the contact spots are either circular or elliptical. Moreover, these stochastic models have a number of shortcomings which can be summarised as follows:

- (1) They require a number of assumptions regarding the probability distribution of asperity heights, slopes, curvatures etc.
- (2) They do not predict the deformed shape of the surfaces, particularly for the areas not in contact.
- (3) Knowledge is required about the size range of asperities that are important to the problem.
- (4) They are limited to discrete contact "spots" which are often assumed to deform independently of one another.

The last assumption may be valid for nominally flat surfaces where the individual contact spots are well separated. However, for concentrated contacts it is usually not the

case (e.g. a small ball against a nominally flat rough surface). Using numerical techniques, the interaction of deformations from all the contact points can be considered.

3.2.4 Numerical Approach

Numerical methods are essential for those contact problems which cannot be solved analytically such as the case of conforming contacts where the separation cannot be obtained approximately by a second degree polynomial. Modern computing facilities have also relaxed many of the old problems encountered by numerical methods, which were usually associated with memory size and speed, due to the a large number of surface elements needed to adequately model the shape of asperities over a reasonable area of surface.

Numerical techniques have been applied to study the indentation problem (Akyuz & Merwin 1968; Hardy et al 1971; Dumas & Baronet 1971; Lee et al 1972; Follansbee & Sinclair 1984) using the stress-strain relation of Reuss (see Hill 1950) for an elastic-perfectly-plastic material. The studies were limited to a spherical or conical indenter on a flat smooth surface. However, only in recent years have numerical techniques found an application in study of rough surface contact problems.

Francis (1982) obtained an elastic contact solution of one-dimension plane strain surface elements having a triangular pressure distribution. The model was tested using a sinusoidal surface. The results showed the early flattening of the waviness present on the deformation of the roughness with increasing applied load. Francis (1983) extended the study to a two-dimensional model to investigate the validity of the one dimensional plane strain assumption. He concluded that the one dimensional model was sufficiently accurate if the ratio of the distance between ^{the} _{peak} ^a _{in} ^{was} _{perpendicular direction} _a greater than 1.5.

Chiu and Liu (1970) used a similar approach to study the contact problems of rough surfaces having Gaussian height distributions. The rough surfaces were provided by computer simulation. Results for load-v-compliance and load-v-area were compared with some of the stochastic models discussed previously.

Webster and Sayles (1986a) presented a similar model based upon one-dimensional elements having a constant pressure. The model was used to investigate the contact of damaged bearing surfaces and study the load-v-area relationship of a sample set of digitised rough surface profiles (Webster et al 1985; Webster & Sayles 1986).

The technique has also been applied to study the sub-surface stresses created around debris indentations on rolling element bearings (Webster et al 1985). West et al (1987a) used the same model to study the variation of surface roughness parameters σ and β^* on the proportionality of load-v-area relationships. He concluded that for constant σ , an increase in β^* creates a corresponding increase in the real contact area and a subsequent reduction in local asperity stresses. While keeping β^* constant, decreasing σ is seen to have a similar effect. Numerical results of load-v-area proportionality also showed good agreement with random process models for strongly anisotropic surfaces due to Bush and his colleagues (1978).

In Webster's elastic contact model, it has been shown that the local asperities' pressures can be many times greater than nominal pressures calculated assuming smooth surfaces even at moderate loads. In this situation it would be expected that the asperities would have collapsed plastically. West et al (1987a) extended the model to include plasticity by using the Johnson 'cavity model' (see section 4.3). The approach is to restrict the elemental surface pressures to a predetermined value which was $1.6H$ where H is the hardness of the material. West and Sayles (1987b) further extended the analysis to three-dimensional elastic-plastic surface contact. The model was used to study the contact on

several typical engineering surfaces.

In comparing a stochastic approach with numerical techniques the latter appear to have several advantages,

- (1) No probabilistic assumptions such as asperity heights, slopes, curvatures etc need to be made and an asperity model is not required.
- (2) The matrix methods used take full account of the interaction of deformations from all the contact points.
- (3) It is possible to predict the elastic strains at all points and thus calculate the deformed geometry for both bodies.
- (4) Individual asperity pressures are calculated and not the expected values.

However the difficulty of choosing the sampling interval and sample length still remains. This choice obviously depends on the spatial range of the topography of interest. If the sampling interval is too large, this will result in the loss of information on the small scale asperities and the numerically calculated contact area will tend to be greater. If the sampling interval is too small, this will lead to a longer computational time which may not justify the accuracy achieved. However, the smallest sampling interval on real rough surface is generally limited by the finite stylus size of the measuring instrument.

3.3 Mode of Deformation

It is of considerable importance to know whether a surface in contact deforms

elastically or plastically. The change in topographic features of a surface under loading is mostly attributed to the plastic crushing of asperities. Many tribological phenomenon such as surface sliding damage (Hollander 1974), scuffing (Hirano 1978), subsurface crack initiation (Suh 1977), wear (Ritchie 1978) etc, are related to plastic deformation. For this reason the boundary between elastic and plastic deformation of rough surfaces has been considered by several workers in terms of a plasticity index.

Blok (1952) investigated the limits of elastic deformation of sliding features, based upon a sinusoidal roughness surface model. For the complete elastic compression such that no void space is left out of contact, the average nominal stress, σ_{av} , is given by,

$$\sigma_{av} = \frac{\pi}{2} \frac{h_{max}}{\lambda} E' \quad (3.8)$$

where h_{max} is the peak to valley height, λ the wavelength, E' is the reduced modulus, which if both surfaces are the same material ^{is} equal to $E/(1-\nu)^2$ and ν is the Poisson's ratio. Blok used the average normal stress to compare with the hardness H of the softer of the two solids to define the limit of elastic contact. Thus if

$$\frac{h_{max}}{\lambda} < \frac{2H}{\pi E'} \quad (3.9)$$

no plastic flow would occur. By considering typical values for the average slope of surface features and material hardness Blok stated that complete levelling of surface features will almost always involve plastic deformation.

Greenwood and Williamson (1966), in their model of rough surface contact described earlier, went on to consider the condition for the onset of plastic flow and derived their plasticity index ψ_{GW} , as a means of determining the mode of deformation where ψ_{GW}

is given by,

$$\psi_{GW} = \frac{E'}{H} \sqrt{\frac{\sigma^*}{R}} \quad (3.10)$$

where σ^* is the rms value of peak height distribution and R is the peak radius. They proposed $A_p/A_a < 0.02$ as the condition for an essentially elastic deformation mode and $A_p/A_a > 0.02$ as a measure of the onset of plastic flow. They concluded that the nature of rough surface contact is almost entirely dependent on the value of their plasticity index. For $\psi_{GW} > 1$, the mode of contact is predominantly plastic and for $\psi_{GW} < 0.6$, the mode of contact is predominantly elastic. Significantly, the applied load was shown to have little influence on whether the contact is elastic or plastic.

Greenwood and Williamson's model of assuming a constant peak radius is obviously not realistic. This restriction was removed by Whitehouse and Archard (1970) who extended the model to allow peaks to have a distribution of curvatures. In addition, they assumed surface profiles have an exponential autocorrelation function, an assumption which is justified for many engineering surfaces. In particular, they obtained an expression for the mean peak height and mean peak curvature in terms of σ and β^* . When substituted into Greenwood and Williamson's plasticity index, they obtained.

$$\psi_{WA} = 0.69 \frac{E'}{H} \frac{\sigma}{\beta^*} \quad (3.11)$$

In the subsequent analysis by Onions and Archard (1973), they suggested that the higher peaks, those which were more likely to make contact, had sharper radii than the lower ones. Thus, the value of ψ_{WA} is likely underestimated and they suggested the following more realistic expression for the plasticity index that allows more plastic deformation,

$$\psi_{OA} = \frac{E' \sigma}{H \beta^*} \quad (3.12)$$

Mikic and Roca (1974) used a similar approach to Greenwood and Williamson (1966), but they defined the boundary as 90% or greater proportion of asperity contacts being in the plastic state. Using equation (3.10), they derived the expression,

$$\psi_{MR} = \frac{H}{E' \tan \theta} \quad (3.13)$$

to determine the mode of deformation. Deformation would be predominantly elastic for $\psi_{MR} > 3$ and predominantly plastic for $\psi_{MR} < 1/3$.

Gupta and Cook (1972) assumed a Gaussian distribution of peak heights and a log-normal distribution of peak curvatures. They defined a topographic index ζ essentially as a measure of the ratio of nominal to real contact area of two surfaces in contact. Thus,

$$\zeta = \frac{1}{\sqrt{2D_s}} \cdot \frac{1}{R_m} \cdot \frac{E'}{H} \quad (3.14)$$

where D_s is the average number of peaks per unit area (or the smaller of the two densities when the surface are different.)

$$R_m = \frac{R_1 R_2}{R_1 + R_2} \quad (3.15)$$

where R_1 and R_2 are the average radii of asperities of the two surfaces.

The onset of plastic flow has also been considered by Bush et al (1976), using the correction between summit heights and curvatures given by random process theory. The mean summit heights and curvatures were expressed in terms of the spectral moment parameters m_0 , m_2 and m_4 and using Greenwood & Williamson's plasticity index, the expression they obtained was,

$$\psi_B = \sqrt{\frac{8}{3}} \left(\frac{E'}{H} \right) \left(\frac{m_0 m_4 c}{\pi} \right)^{\frac{1}{4}} \quad (3.16)$$

where $c=1-0.8968/\alpha$ and $\alpha=m_0 m_4/m_2^2$.

It has been suggested (Halling & Nuri 1975) that the hardness is not a constant but varies due to the work hardening of materials. Francis (1977) considered the transition region between elastic and ideal plastic contact by including the work-hardening effect. Using Nayak's surface statistics, he defined what amounts to two indices ψ_{F1} and ψ_{F2} ,

$$\psi_{F1} = \frac{\sigma'}{\sqrt{k}} \frac{E_1}{P_m} \quad (3.17)$$

and

$$\psi_{F2} = \frac{\sigma'}{\sqrt{k}} \frac{E'}{P_m} \quad (3.18)$$

where k and E' are given by

$$k = \sqrt{1.5} \frac{\sigma'^2}{\sigma \cdot \sigma''} \quad (3.19)$$

and

$$\frac{1}{E'} = \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \quad (3.20)$$

k is a parameter similar to Nayak's bandwidth parameter α , and used to describe the spectral nature of the surface. σ'' is the standard deviation of the second differential of the gap between the two surfaces and σ' is the rms slope. σ is the rms value of the gap. P_m is the uniaxial yield stress.

In general ψ_{F1} is the most important parameter; the larger the value of ψ_{F1} the greater is the deviation of the contact mechanics from being totally elastic and this trend increases with decreasing strain hardening. The elastic model is valid for any contact for which $\psi_{F2} < 4$ and the complex iterative plastic-elastic model must be used for $\psi_{F2} > 14$. Between 4 and 14 the behaviour depends on specific values of the parameters.

From equations (3.8) to (3.18) the plasticity index derived from various surface models share the common feature of a non-dimensional combination of surface statistics and material constants which determine whether the contact will be essentially elastic or mixed elastic-plastic. The topographic parameters basically involve either first differentials ($\sigma'', 1/R$), peak density (D_p) or spectral moment related parameters such as k .

The plasticity index was first derived based on the rms value of the peak height distribution and a peak radius. This was expressed in terms of σ and β^* by using the

exponential autocorrelation model (Whitehouse & Archard 1970), or in terms of the spectral moments m_0 , m_2 and m_4 by using the random process approach, (Nayak 1971), to encounter a range of asperity shapes. However, the exponential model has created much controversy between theoreticians and experimentalists, mainly because of its discontinuity at the origin, which prevents the second and fourth spectral moments being defined uniquely. The random process approach of Nayak requires a well defined limit to the detail on surfaces. Although stylus measurements always reveal such a limit by showing a smooth autocorrelation function at the origin; the reason for this is now well known and due to finite radius of the stylus acting as a low-pass filter. In no way does it imply that surfaces have a natural short-wavelength limit. Whitehouse (1970) argued that the exponential model, apart from the region close to the origin, has in many cases given good agreement with measured data, and this has been confirmed by many others.

CHAPTER 4

ROUGH SURFACE CONTACT : NUMERICAL CONTACT MODEL

4.1 Introduction

In this chapter, a numerical contact model of a smooth ball on a directionally structured anisotropic rough surface is presented. The contact geometry of the model coincides with that of the friction tests describes in Chapter 7 and 8 in which a smooth ball is slid on a directionally structured anisotropic and nominally flat rough surfaces. The model will be used to obtain contact information such as the degree of plasticity and the real area of contact over a range of rough surface specimens. Results from this contact analysis will be used to correlate results from friction tests.

A general approach using the numerical method is to divide the contacting surfaces into small elements of appropriate size. Over each element, it is necessary to choose the form of pressure, often a constant or a linear variation being used. From the shape of the surface element and the assumed elemental pressure distribution, the matrix of influence coefficients can be obtained which express the displacement u_i at a general point i due to a unit pressure element p_j centred at point j . The total displacement at a given point is merely the sum of the displacements due to all the elemental pressures. In this way a set of linear simultaneous equations that relate displacements u_i to pressures p_j can be formed. These equations are solved by satisfying the boundary conditions due to the elastic displacements of corresponding points on the two surfaces,

$$u_1 + u_2 - \delta = 0 \quad \text{within contact}$$

and

$$u_1 + u_2 - \delta = 0 \quad \text{outside contact} \quad (4.1)$$

From the shape of the contacting bodies and the displacement of one body towards the other, the elastic displacements u_i can be obtained for those points in contact and the unknown pressures found. However, it is usually not known that a point is in contact in advance. This problem can be solved by two different methods, namely the variational method (Kalker & van Randen 1972) or the Matrix Inversion method (Bentall & Johnson 1967; Paul & Hashemi 1981).

In variational methods, the true contact area and pressure distribution are those which minimise the total strain energy. The strain energy can be expressed in terms of the unknown forces and a matrix of influence coefficients. This minimum strain energy solution has been applied to frictionless non-Hertzian contact problems by Kalker and van Randen (1972).

In the Matrix Inversion method, a guess is made of the shape of the contact surface, and its size is chosen to be sufficiently large to enclose the true area. The first set of surface displacement elements can be obtained and the unknown pressures solved. For this first guess it will be found that some pressures p_j are negative, which implies that a tensile traction is required at some mesh points to maintain contact over the whole of the assumed area. For the second iteration these mesh points are excluded from the assumed contact area and the process continues until the set of p_j 's are all positive for those points in contact. This approach is used in the numerical contact model described as follow.

4.2 Constraints of the numerical contact model

Referring to Figure 4.1 and in addition to the usual linear elastic theory assumptions, the assumptions of the model are:

- (1) Strains are small for the linear elastic theory equations to be applicable.
- (2) The contact area is contained within the geometric overlap region.
- (3) Inside the elastic contact region, the total displacement, δ , is equal to the sum of the displacement of each body, u_1 and u_2 , i.e. $u_1(x)+u_2(x)=\delta(x)$ for $H > p(x) > 0$ where H is the Hardness of the softer material.
- (4) Inside the plastic contact region, the pressure is equal to H .
- (5) Outside the contact region,
 $u_1(x)+u_2(x)>\delta(x)$ and $p(x)=0$
- (6) Relative horizontal strains and displacements are ignored. i.e. frictionless contact.
- (7) The surface slopes are reasonably small so that forces can be assumed to act normally to the surfaces.

4.3 The Plasticity Model

The plasticity model used is based on the experimental observations of Samuels and Mulhearn (1956) and Mulhearn (1959) that the subsurface displacements produced by any blunt indenter are approximately radial from the point of first contact, with roughly hemispherical contours of equal strain. The plastic zone is contained within the interior of the body and the material displaced by plastic flow can be considered to be accommodated by the elastic expansion of subsurface core (see Johnson 'cavity' model of mixed elastic-plastic deformation (Johnson 1985)). In the contact model, the contact pressure is allowed to increase until it is equal to the hardness of the softer material.

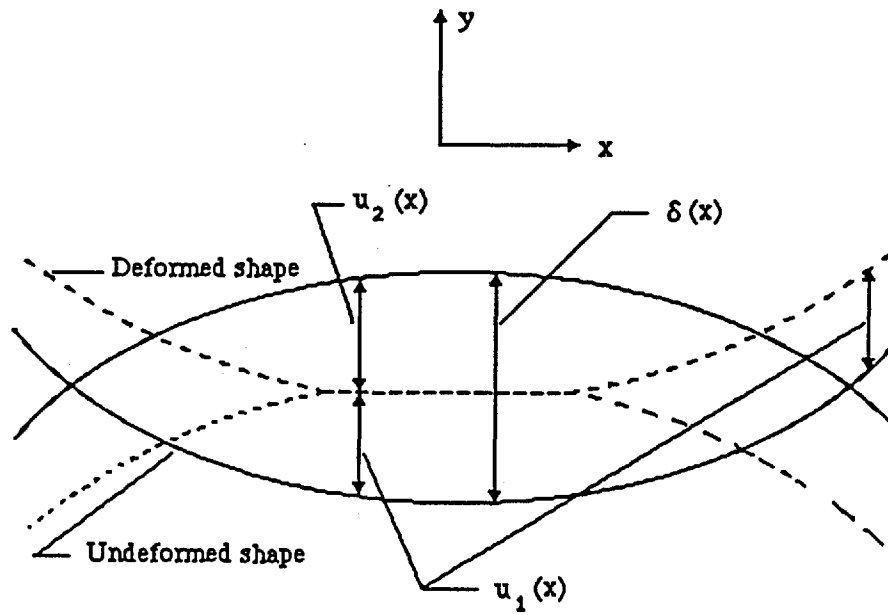


Figure 4.1 Diagram showing the geometric overlap and deformed shape of an ideal smooth contact.

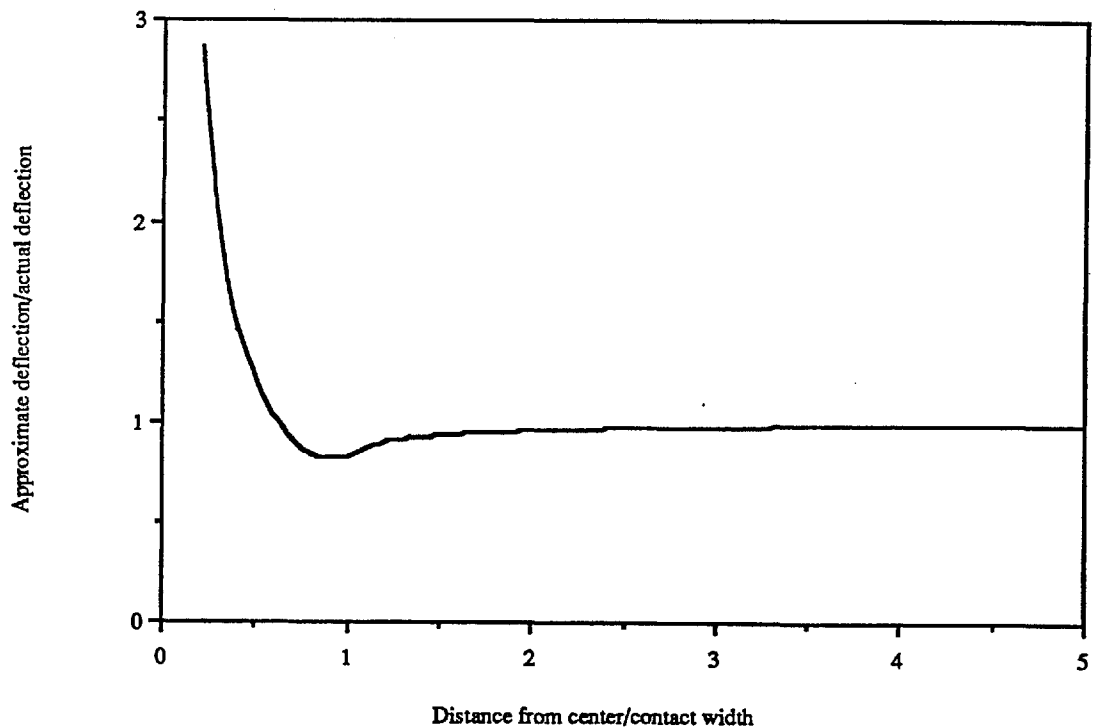


Figure 4.2 The ratio of surface displacement given by equation (4.2) to equation (4.4) against distance between the surface element and the pressure.

4.4 Contact Equations

The classical approach to finding the stresses and displacements in an elastic half-space due to surface forces is due to Boussinesq (1885), and Cerrati (1882) who made use of the theory of potential. The approach is presented by Love (1952). The Boussinesq theory gives the normal surface deflection u_z of a surface point due to a concentrated point force P_L acting at a distance r normally to the surface i.e.

$$u_z = \frac{1-\nu^2 P_L}{\pi E r} \quad (4.2)$$

The surface displacement produced by a normal pressure distributed over an area S of the surface is obtained by superposition, in polar coordinates (s, ϕ) .

$$u_z = \frac{1-\nu^2}{\pi E} \int \int_S p(s, \phi) ds d\phi \quad (4.3)$$

The surface data is obtained by sampling on a rectangular grid over a finite interval. It would thus be convenient to use a rectangular area of sides $2a \times 2b$ over which the normal pressure could be assumed uniform.

The effect of a uniform pressure acting on a rectangular area of $2a \times 2b$ has been analysed by Love (1929) using equation (4.3). The deflection of a general point (x, y) on the surface due to a uniform pressure p over a rectangle centred at the origin is:

$$\begin{aligned}
\frac{\pi E}{1 - \nu^2} \frac{u_z}{p} = & (x + a) \ln \left[\frac{(y + b) + \{(y + b)^2 + (x + a)^2\}^{\frac{1}{2}}}{(y - b) + \{(y - b)^2 + (x + a)^2\}^{\frac{1}{2}}} \right] \\
& + (y + b) \ln \left[\frac{(x + a) + \{(y + b)^2 + (x + a)^2\}^{\frac{1}{2}}}{(x - a) + \{(y + b)^2 + (x - a)^2\}^{\frac{1}{2}}} \right] \\
& + (x - a) \ln \left[\frac{(y - b) + \{(y - b)^2 + (x - a)^2\}^{\frac{1}{2}}}{(y + b) + \{(y + b)^2 + (x - a)^2\}^{\frac{1}{2}}} \right] \\
& + (y - b) \ln \left[\frac{(x - a) + \{(y - b)^2 + (x - a)^2\}^{\frac{1}{2}}}{(x + a) + \{(y - b)^2 + (x + a)^2\}^{\frac{1}{2}}} \right] \quad (4.4)
\end{aligned}$$

Equation (4.4) gives the exact solution of a surface deflection due to the uniform pressure acting on a rectangular area. If the distance between the elemental pressure and the surface displacement is large, the surface displacement due to the elemental pressure can be regarded as due to the concentrated load acting at the centre. Then equation (4.2) can be used as a good approximation to relate the surface deflection and the pressure.

It is advantageous to use equation (4.2) as far as computing time is concerned. For example, if the number of contact elements is 100, equation (4.2) or (4.4) is used 10000 times to obtain the influence coefficient matrix. In the numerical solution, an iterative method is used in which each iteration will use equation (4.2) or (4.4) 10000 times. Thus, if equation (4.2) can be used as much as possible, then the computing timing will be reduced considerably.

To test under what circumstances equation (4.2) can be used in preference to (4.4), it is necessary to know the distance from which the surface displacement given by equation (4.2) can be considered as close to that given by equation (4.4). Figure 4.2 shows the variation of surface displacement with distance according to equations (4.2) and (4.4). Taking 95% of the actual displacement as a good approximation, then equation (4.2) can be used to calculate the influence coefficient matrix if the distance between the pressure and the deflection is greater than $2a$.

The total deflection at any point can be calculated by summing the influence of each pressure element acting on area $2a \times 2b$. Therefore, we have a set of linear equations

$$[u_i] = [C_{ij}][p_j] \quad (4.5)$$

The values of the u_i 's are obtained by giving the bodies a known bulk overlap. The sum of the strains in the two bodies at any contacting point is equal to the geometric overlap δ . Thus, the strain in each body can be expressed in terms of the geometric overlap using the relationship:

$$\frac{u_{z1}E_1}{1-\nu_1^2} = \frac{u_{z2}E_2}{1-\nu_2^2} \quad (4.6)$$

which gives

$$u_{z1} = \frac{\delta}{1 + \frac{E_1(1-\nu_2^2)}{E_2(1-\nu_1^2)}} \quad (4.7)$$

By assuming some of the points to be in contact, the strains at these points of contact can be obtained. The influence coefficients can be calculated by using equation

(4.2) or (4.4) depending on the distance between the pressure element and the surface point. The solution of the set of simultaneous equations is obtained by the Gaussian Back Elimination method.

Once the solution for the elemental pressures has been found any elemental pressures which are negative are removed from the contact (Assumption 5). Also, any asperity with a calculated pressure greater than the flow pressure would collapse and the pressure should be reduced to the flow pressure. There are two methods to reduce the pressure. The first method is to modify the shape of the original undeformed surfaces until the pressures are equal to the flow pressure of the material. The second method is to put the pressures equal to the flow pressure and re-calculate the pressure distribution until all pressures satisfy the contact conditions. In the numerical solution employed here, the second method is used, as it gives a direct method of obtaining the plastic area and no assumption of the asperity shape is needed.

The deformations are found everywhere to check that no contact occurs outside the assumed contact regions. If contact does occur outside these regions, the extra points are added to the assumed contact region and the system of linear equations is then solved again for the new assumed contact points.

This process is repeated until convergence on a particular set of contact points is achieved. The total contact area, the plastic area and the normal load can then be calculated by integrating the contact elements and their pressures.

4.5 Implementation of the Model

The contact program is implemented on an Acorn A443 Workstation micro-computer. This fast 32 bit machine with 4 megabytes of RAM, although not as powerful

and fast as a main-frame computer, was chosen so that it can form part of a stand alone surface data acquisition and analysis system.

4.6 Numerical Solution

For the contact of a ball on a directionally structured anisotropic rough surfaces, the number of pressure and deflection elements that can be used are limited by computer memory. The computer chosen can analyse up to about 250 contact elements. If each contact element is $2 \times 2 \mu\text{m}^2$, then the area that can be analysed is $1000 \mu\text{m}^2$ which is equivalent to a nominal circular contact of about $36 \mu\text{m}$ diameter. Under these conditions the maximum number of contact elements across a section is then 18. This is obviously not sufficient for the investigation.

In order to increase the number of points analysed, the pressure for each contact along the lay is assumed Hertzian. The surface elements are set lying at the centre of contact, as shown in Figure 4.3. Thus the pressure p_y along the lay for an element j is,

$$p_y = p_j \left(1 - \frac{y^2}{w_j^2}\right)^{\frac{1}{2}} \quad (4.8)$$

where w_j is the contact length from the centre.

Referring to Figure 4.4, showing a uniform pressure element p_y acting on an elemental area $2a \times 2b$, the vertical surface displacements at point i due to an applied pressure p_y is given by equation (4.2) or (4.4), thus we have

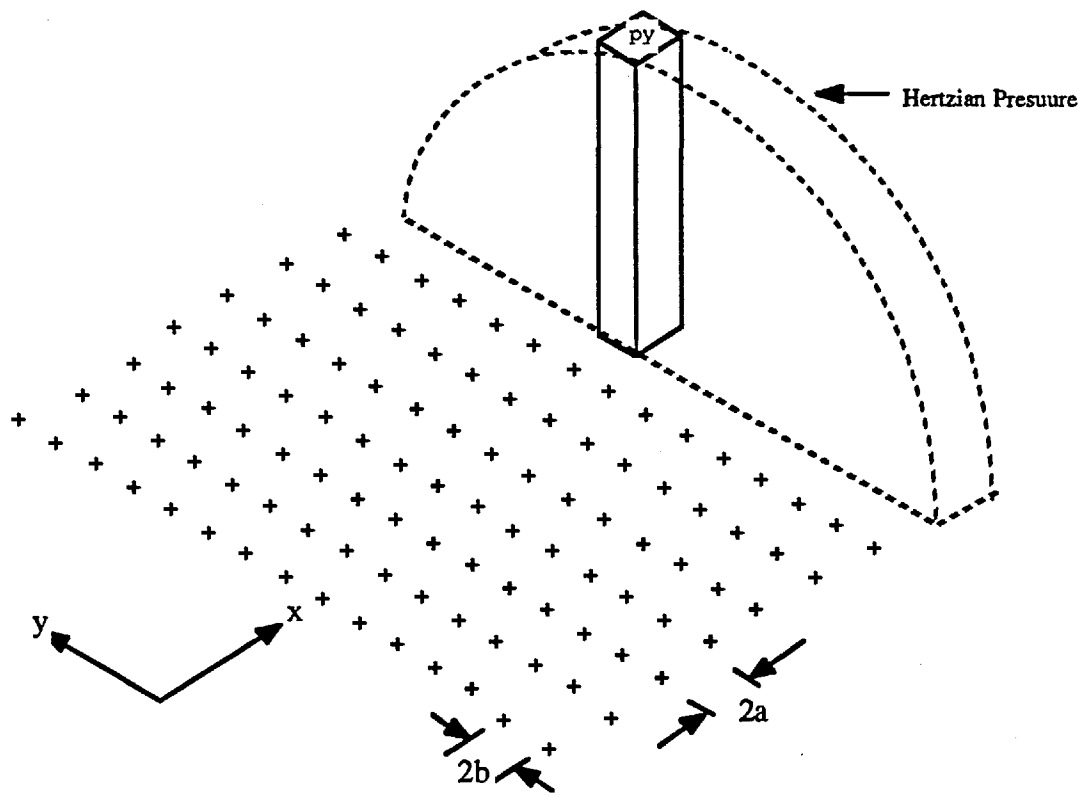


Figure 4.3 Schematic representation of surface and pressure elements.

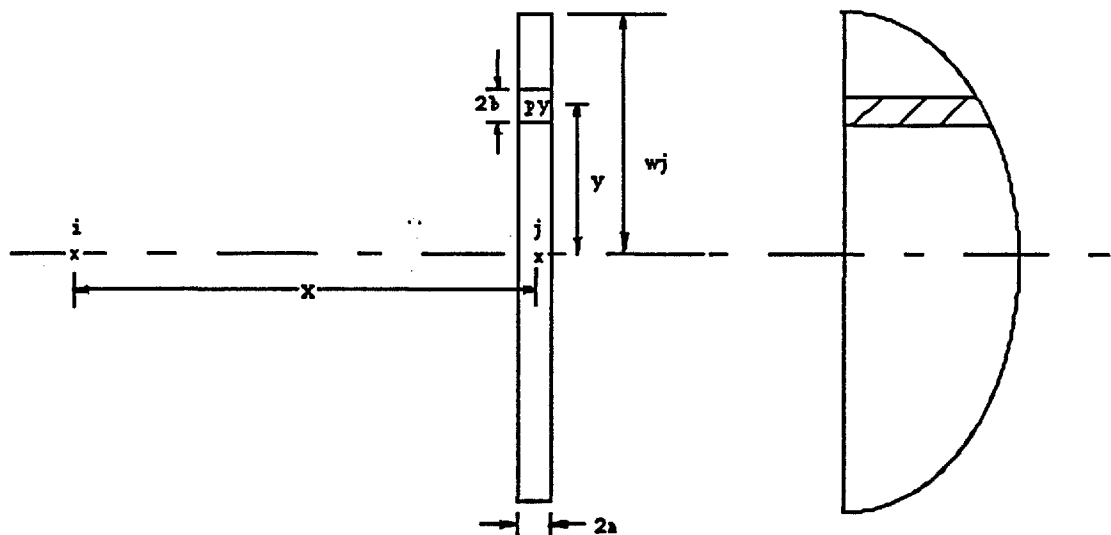


Figure 4.4 Diagram showing the surface displacement i and pressure element p_y subdivided by Hertzian pressure at j .

$$u_i = p_y \frac{(1 - \nu^2)}{\pi E} F_{xy} \quad (4.9)$$

where

$$\begin{aligned} F_{xy} = & (x + a) \ln \left[\frac{(y + b) + \{(y + b)^2 + (x + a)^2\}^{\frac{1}{2}}}{(y - b) + \{(y - b)^2 + (x + a)^2\}^{\frac{1}{2}}} \right] \\ & + (y + b) \ln \left[\frac{(x + a) + \{(y + b)^2 + (x + a)^2\}^{\frac{1}{2}}}{(x - a) + \{(y + b)^2 + (x - a)^2\}^{\frac{1}{2}}} \right] \\ & + (x - a) \ln \left[\frac{(y - b) + \{(y - b)^2 + (x - a)^2\}^{\frac{1}{2}}}{(y + b) + \{(y + b)^2 + (x - a)^2\}^{\frac{1}{2}}} \right] \\ & + (y - b) \ln \left[\frac{(x - a) + \{(y - b)^2 + (x - a)^2\}^{\frac{1}{2}}}{(x + a) + \{(y - b)^2 + (x + a)^2\}^{\frac{1}{2}}} \right] \end{aligned} \quad (4.10)$$

or

$$F_{xy} = \frac{2a \cdot 2b}{\sqrt{x^2 + y^2}} \quad (4.11)$$

For each element j, the pressure is sub-divided into $2N_y+1$ elements. The elemental width $2b$ is given by,

$$2b = \frac{2w_j}{2N_y + 1} \quad (4.12)$$

The distance between elements i and j is given by,

$$x = |i - j| \cdot 2a \quad (4.13)$$

Writing $y=k.2b$, the surface displacement at point i due to Hertzian pressure at j becomes,

$$u_i = \frac{1-\nu^2}{\pi E} \sum_k p_j \left(1 - \frac{4k^2}{(2N_y + 1)^2}\right)^{\frac{1}{2}} F_{xy} \quad (4.14)$$

where $k=0,\pm1,\pm2,\dots,\pm N_y$

If the number of contact is N_c , then a set of simultaneous equations can be set up by superposition. Thus,

$$[u_i] = [C_{ij}] [p_j] \quad (4.15)$$

where $i=1,2,3,\dots,N_c$ and

$$C_{ij} = \frac{1-\nu^2}{\pi E} \sum_j \sum_k p_j \left(1 - \frac{4k^2}{(2N_y + 1)^2}\right)^{\frac{1}{2}} F_{xy} \quad (4.16)$$

The set of simultaneous equations (4.15) are solved by a Matrix Inversion method. An initial guess of the contacting points can be obtained by considering the regions of geometrical overlap between the two bodies. For each point in contact, the strain in each body is calculated using equation (4.7).

If the contact length of each element, in the lay direction, is known, then the

influence coefficient can be calculated and the contact pressure found. However, the contact length of each element is not known but its value can be obtained by the information given at the boundary of contact at which the surface deflection u_i given by equation (4.7) is equal to the surface deflection u_z induced by the contact pressures. If $u_z > u_i$, then the contact length is overestimated and the contact length should be reduced. Similarly, if $u_z < u_i$, the contact length should be increased. However, there is no direct method relating contact lengths to surface deflections or pressures, and therefore no direct method of defining an appropriate change to the contact lengths. An effective way to determine the contact lengths will be discussed later.

The initial contact lengths are obtained by the geometrical overlap which allows a calculation of the first set of p_j 's. If there are negative elemental pressures, these points are excluded and the problem solved again to obtain a new set of C_{ij} 's and p_j 's. This process is repeated until the values of p_j 's are everywhere positive.

The next step is to compare the values of elastic pressure p_j to the material hardness H . If any elemental pressure is greater than H , the point of displacement is shifted from the centre of the contact element where plastic flow will begin to a point where the contact pressure begins to decrease. An index array is used to store the plastic contact width p_{cw} . The plastic contact width is incremented according to the difference of the current p_j 's and H . In the case of plastic deformation, the pressure profile is no longer Hertzian and equation (4.14) has to be modified to include the effect of plastic deformation. Consider the Hertzian pressure profile being truncated to give the pressure profile under plastic deformation shown in Figure 4.5. When $y < y_h$, the pressure will be equal to the indentation hardness H . On the other hand, when $y > y_h$ the pressure will be,

$$p = p_0 \left(1 - \frac{y^2}{w_j^2}\right)^{\frac{1}{2}} \quad (4.17)$$

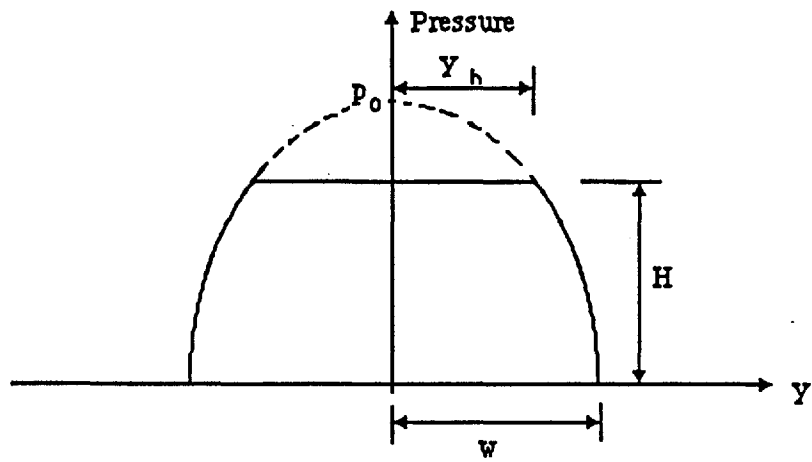


Figure 4.5 Diagram showing the elastic pressure being truncated to give the plastic pressure profile.

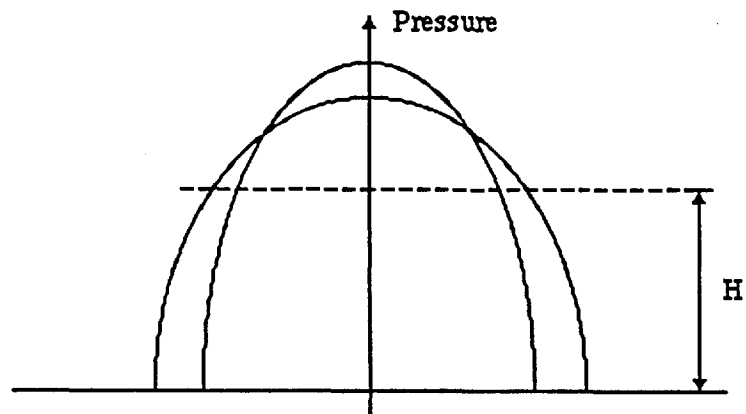


Figure 4.6 Diagram showing two possible forms of Hertzian pressure having the same asperity load.

At $y=y_h$, $p=H$ thus eliminating p_o from equation (4.17). Therefore,

$$p = H \cdot \sqrt{(1 - \frac{y^2}{w_j^2}) / (1 - \frac{y_h^2}{w_j^2})} \quad \text{where } y > y_h \quad (4.18)$$

Writing y and y_h in terms of discrete points, i.e.

$$y = k \cdot 2b \quad \text{where } k=0,\pm1,\pm2,\dots,\pm N_k \quad (4.19)$$

and

$$y_h = k_h \cdot 2b \quad \text{where } k_h \text{ is the point at which } y=y_h \quad (4.20)$$

Substitute equations (4.12) ,(4.19) and (4.20) into equation (4.18),

$$p = H \cdot \alpha_k \quad (4.21)$$

where

$$\alpha_k = \sqrt{(1 - \mu^2 k^2) / (1 - \mu^2 k_h^2)} \quad (4.22)$$

and

$$\mu = \frac{2}{2N_y + 1} \quad (4.23)$$

Thus, the elastic-plastic contact equations can be generalised as follow,

$$u_i = \frac{1 - \nu^2}{\pi E} \sum_k p_j \alpha_k F_{xy} \quad (4.24)$$

For elastic deformation, α_k is calculated by putting y_h equal to zero, which is equation (4.14). For plastic deformation, $p_j = H$. If $k < k_h$, α_k is equal to 1 and if $k > k_h$, α_k is given by equation (4.22). A set of C_{ij} 's are again obtained using equation (4.24). The solution for the p_j 's is continued until the plasticity model are satisfied.

In what follows the method of determining contact lengths is described. The contact lengths are always less than their geometrical overlap because of the pressure acting at the neighbouring points. The amount of deformation depends upon the magnitude and the spatial distance of elemental pressures. At some stages of the iterative process, a change in contact length of a particular contact will re-distribute the contact pressures which in turn will affect the displacements. In order to obtain an effective way to make changes to the previously assumed contact lengths, it is essential to know how the u_i 's and p_j 's vary with respect to the contact length w_j . A way of doing this is to differentiate equation (4.2) or (4.4) with respect to w_j . As far as the contact at the boundary is concerned, equation (4.2) can be used as a good approximation. By differentiating equation (4.2) with respect to w_j , see Appendix 1,

$$\begin{aligned} \frac{du_i}{dw_j} = & \frac{2 a (1 - \nu^2)}{\pi E} \sum_j p_j \sum_k \alpha_k \frac{\mu x^2}{(x^2 + y^2)^{\frac{3}{2}}} \\ & + \frac{2 a (1 - \nu^2)}{\pi E} \sum_j \frac{dp_j}{dw_j} \sum_k \alpha_k \frac{\mu w_j}{(x^2 + y^2)^{\frac{1}{2}}} \end{aligned} \quad (4.25)$$

At equilibrium, the load capacity on each asperity is constant. Therefore, the change in contact length and pressure with respect to w_j will be in such a way that the load remains unchanged. The load L on each asperity with pressure p_j is given by,

$$L = 2 \cdot a \cdot \int_{w_j}^{w_j} p_j \left(1 - \frac{y^2}{w_j^2}\right)^{\frac{1}{2}} / \left(1 - \frac{y_h^2}{w_j^2}\right)^{\frac{1}{2}} dy \quad (4.26)$$

writing

$$\theta = \sin^{-1} \frac{y_h}{w_j} \quad (4.27)$$

and solving equation (4.26)

$$\begin{aligned} L &= 4 \cdot a \cdot p_j \cdot \left(\frac{w_j \cdot \left(\frac{\pi}{2} - \theta - \frac{\sin 2\theta}{2} \right)}{\cos \theta} + y_h \right) \\ &= 4 \cdot a \cdot p_j \cdot (w_j f + y_h) \end{aligned} \quad (4.28)$$

where

$$f = \frac{\pi/2 - \theta - \sin 2\theta/2}{\cos \theta} \quad (4.29)$$

Differentiating equation (4.28) with respect to w_j and putting $dL=0$, we get

$$\frac{dp_j}{dw_j} = \frac{-p_j}{w_j f + y_h} \left[\frac{y_h}{w_j} \left(2 - \frac{f \sin \theta}{\cos^2 \theta} \right) + f \right] \quad (4.30)$$

Substituting equation (4.30) into equation (4.25)

$$\begin{aligned} \frac{du_i}{dw_j} = & \frac{2a(1-v^2)}{\pi E} \left[\sum_j p_j \sum_k \frac{\alpha_k \mu x^2}{(x^2 + y^2)^{\frac{3}{2}}} \right] + \\ & \frac{2a(1-v^2)}{\pi E} \sum_j \frac{-p_j}{w_j f + y_h} \left[\frac{y_h}{w_j} \left(2 - \frac{f \sin \theta}{\cos^2 \theta} \right) + f \right] \sum_k \frac{\mu w_j}{(x^2 + y^2)^{\frac{1}{2}}} \end{aligned} \quad (4.31)$$

Thus, for small increments of surface displacement, we have a set of simultaneous equations

$$[\Delta u_i] = [A_{ij}][\Delta w_j] \quad (4.32)$$

where

$$\begin{aligned} A_{ij} = & \frac{2a(1-v^2)}{\pi E} \left[\sum_j p_j \sum_k \frac{\alpha_k \mu x^2}{(x^2 + y^2)^{\frac{3}{2}}} \right] + \\ & \frac{2a(1-v^2)}{\pi E} \sum_j \frac{-p_j}{w_j f + y_h} \left[\frac{y_h}{w_j} \left(2 - \frac{f \sin \theta}{\cos^2 \theta} \right) + f \right] \sum_k \frac{\mu w_j}{(x^2 + y^2)^{\frac{1}{2}}} \end{aligned} \quad (4.33)$$

At the boundary of contact the displacement u_i should be equal to the total surface deflection u_z induced by the contact pressure. If $u_z > u_i$, then the contact length is overestimated initially and the contact length should be reduced. Similarly, if $u_z \leq u_i$, the contact length should be increased. Thus,

$$\Delta u_i = u_i - u_z \quad (4.34)$$

The terms A_{ij} can be obtained by the current values of p_j 's and w_j 's. A set of Δw_j 's are obtained by solving the set of simultaneous equations (4.32). A new set of contact lengths are established by adding the current values of contact lengths to the Δw_j 's. The process of getting the w_j 's and Δw_j 's are repeated until all the values of Δw_j are smaller than an acceptable value, say $2\mu\text{m}$.

Finally the surface points out of contact are checked everywhere such that the surface displacements are greater than the geometrical overlap. Otherwise, the iterative process is repeated to obtain p_j 's and w_j 's with the points included in the contacts.

It may be noted the numerical contact solution involves many unknowns such as the number of contacts, the contact lengths and the plastic lengths, each affecting the other. Initially, the geometrical overlap length w_g is used to obtain the pressure distribution. The actual contact length w_a is always less than the geometrical overlapped resulting in the underestimation of pressure. Figure 4.6 shows two possible forms of Hertzian pressure having the same asperity load but with different contact lengths. If both the maximum Hertzian pressures are higher than the indentation hardness, the asperity at the centre will undergo plastic deformation. If the boundary between the plastic and elastic deformation is taken as though the top pressure has been truncated at a level corresponding to the hardness of the softer material, the plastic contact width based upon the geometrical overlap is likely to be overestimated. Should this happen, the numerical program may crash or have problems with convergence.

The problem can be avoided by decreasing the maximum pressure in stages. The intermediate hardness can be taken as two or three times the actual hardness such that the pressures higher than the intermediate hardness are reduced until all contact assumptions are satisfied.

The complete solution for a given load and geometry is obtained as follows and a flow chart showing the major features of the solution is shown in Figure 4.7.

Step 1 Set up constants

Values δ , E_1 , E_2 , ν_1 , ν_2 , ball diameter, contact position and total load are input by the operator through the VDU.

Step 2 Set up geometry

Read data from files of either previously recorded surfaces profiles, or generated from appropriate body shapes stored in variable arrays. Then calculate sphere coordinates in relation to the profile.

Step 3 Rigid body movement

The upper body is given a rigid body displacement δ towards the lower body until contact is detected.

Step 4 Set up Matrices for first iteration

Values of u_i and C_{ij} are calculated for all overlapped points using equations (4.7) and (4.16) respectively. w_j 's are calculated from the geometrical overlap. An index array is used to store the positions of the contacting elements, which is then used to calculate the values of x by equation 4.13 and check for recontact. The surface displacement points are set at the centre of contact.

Step 5 Solve for pressures

Solve for values of p_j 's for the current displacement, using Gaussian Elimination

Step 6 Check for negative pressures

The solution for p_j can give some negative (tensile) pressures which therefore

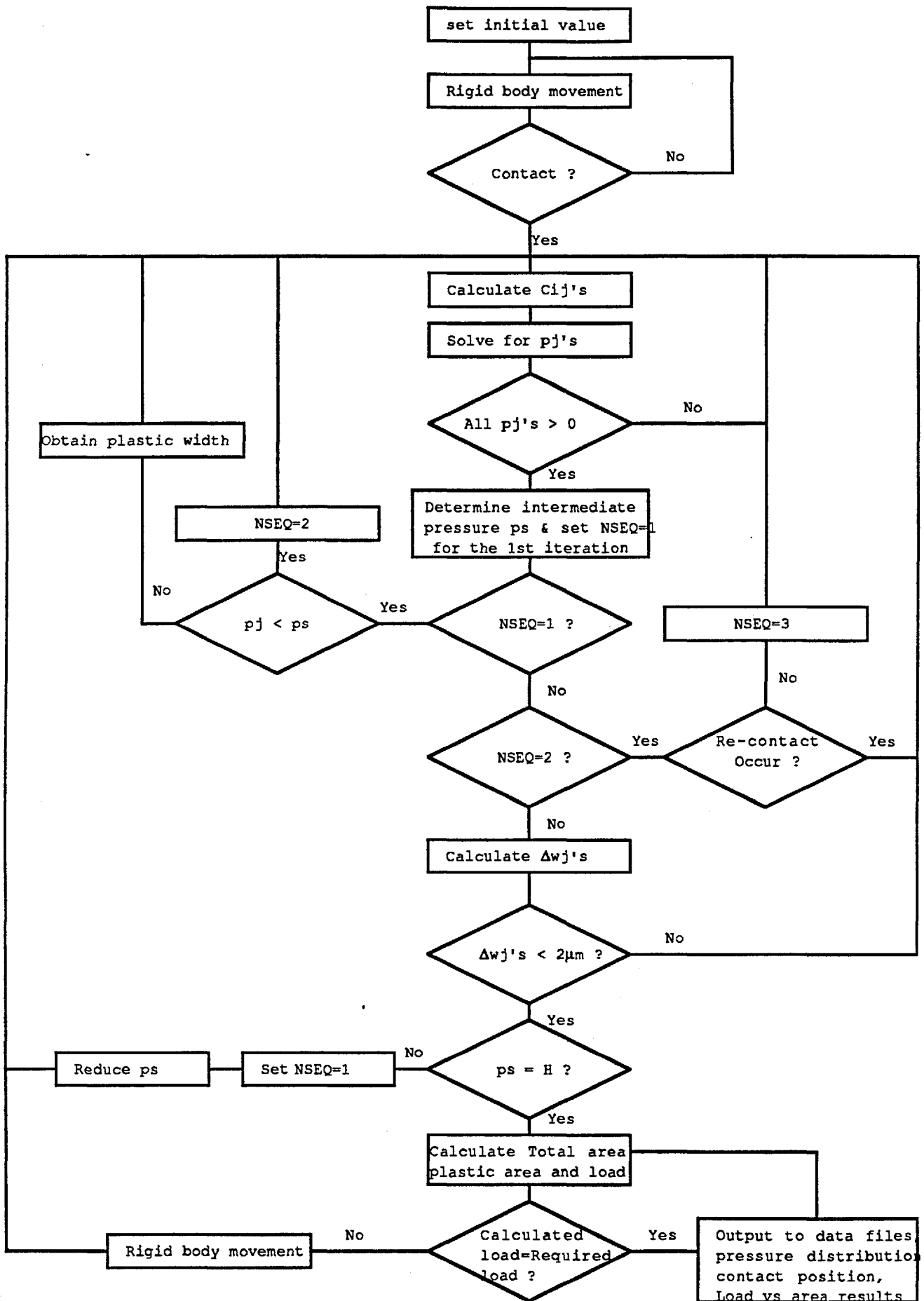


Figure 4.7 Flow chart for the numerical elastic-plastic contact programs.

violates assumption 3. This is due to the fact that the initial guess at δ_i , based upon the geometric overlaps, contains points which will not be part of the true solution and hence underestimates the true surface strains. A new set of u_i 's are then calculated, but excluding points corresponding to $p_j < 0$. A second array is used to store the position of the current contacting elements and is used to calculate the new C_{ij} values. Steps 4,5 and 6 are repeated until all p_j 's > 0 .

Step 7 Set up intermediate hardness

It is necessary to reduce the plastic pressure in stages to avoid problems with convergence. The intermediate stages are set at $p_s = 1.5H, 2.0H, 2.5H, 3.0H$ and so on. Therefore, the number of stages depends on the maximum pressure calculated. Any asperity contact with $p_j > p_s$ is subjected to an increase in its plastic contact width until all p_j 's $< p_s$.

Step 8 Calculate plastic contact width, checking re-contact or calculate contact length

The calculation of plastic contact width, number of contacts and contact length are obtained individually while keeping the others as constants. A sequence is set up to reduce first the plastic pressure to the intermediate hardness. Then any p_j 's < 0 or contact occurs are checked for re-contact outside geometrical overlapped contact boundary. Finally, the contact lengths are calculated using equation (4.32) until all the Δw_j 's $< 2 \mu\text{m}$. An index NSEQ is set for the next operation after each set of calculations is finished.

Step 9 Reduce intermediate hardness

If the intermediate hardness p_s is greater than the indentation hardness H_s , the intermediate hardness is reduced according to step 7 for the next operation. Steps from 4 are repeated again until the intermediate pressure is reduced to the required hardness.

Step 10 Output data

Numerical integration of the elemental pressure, contact width and plastic contact

width are performed to calculate the load, total contact area and plastic area respectively. If the resulting pressure integral is less than the required total load, the process is repeated from step 4. Two separate output data files are created to store load versus area results, pressures, and initial and final geometries.

Figure 4.8 shows a test case of the contact on a smooth surface. The theoretical pressure distribution and normal displacement are plotted on the same axis. The numerical result corresponds almost exactly with the Hertz analytical result. Figure 4.9 shows a 3-dimensional view of the pressure distribution and the deformed geometry of a ball on a sinusoidal profile. Figure 4.10 shows the pressure distribution and normal displacement at a section through the centre. The results show the expected behaviour, i.e. a number of contact regions in which the peak pressure is flattened by plastic deformation and decreases away from the ball centre. Figure 4.11 shows a 3-dimensional view of the pressure distribution and the deformed geometry of a smooth ball on a rough surface. Figure 4.12 shows the pressure distribution and the deformed profile at a section through the centre of Figure 4.11. Such results indicate that the numerical contact model described represents a very useful tool for investigating the problems of rough surface contact.

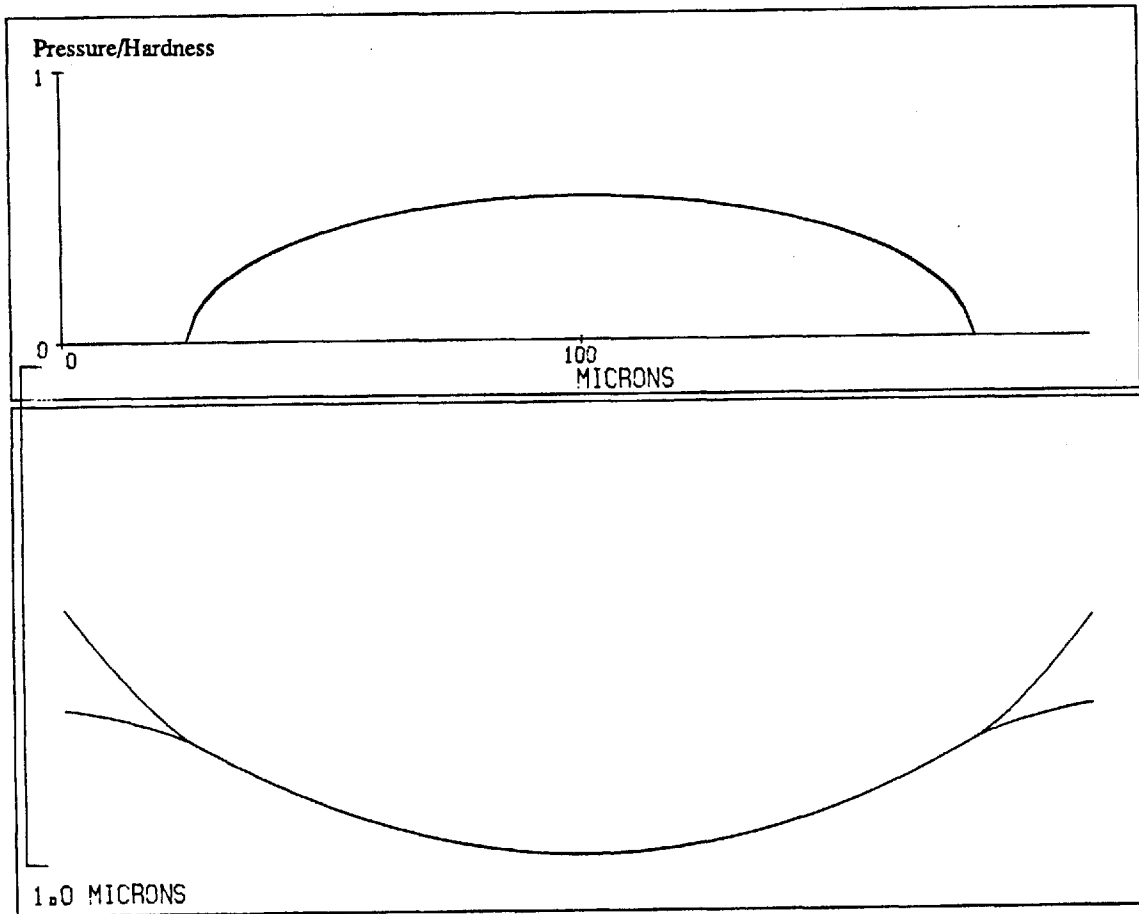


Figure 4.8 Smooth case results from the numerical contact model. A Hertzian pressure distribution has been superimposed and is seen to correspond almost exactly to the numerical results.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

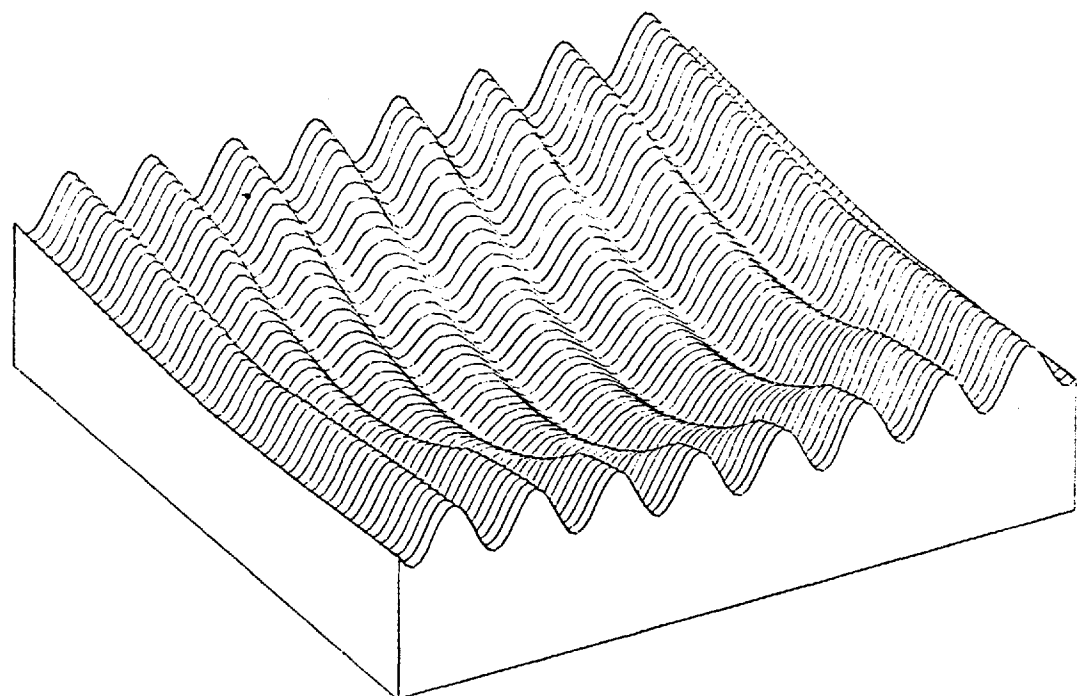
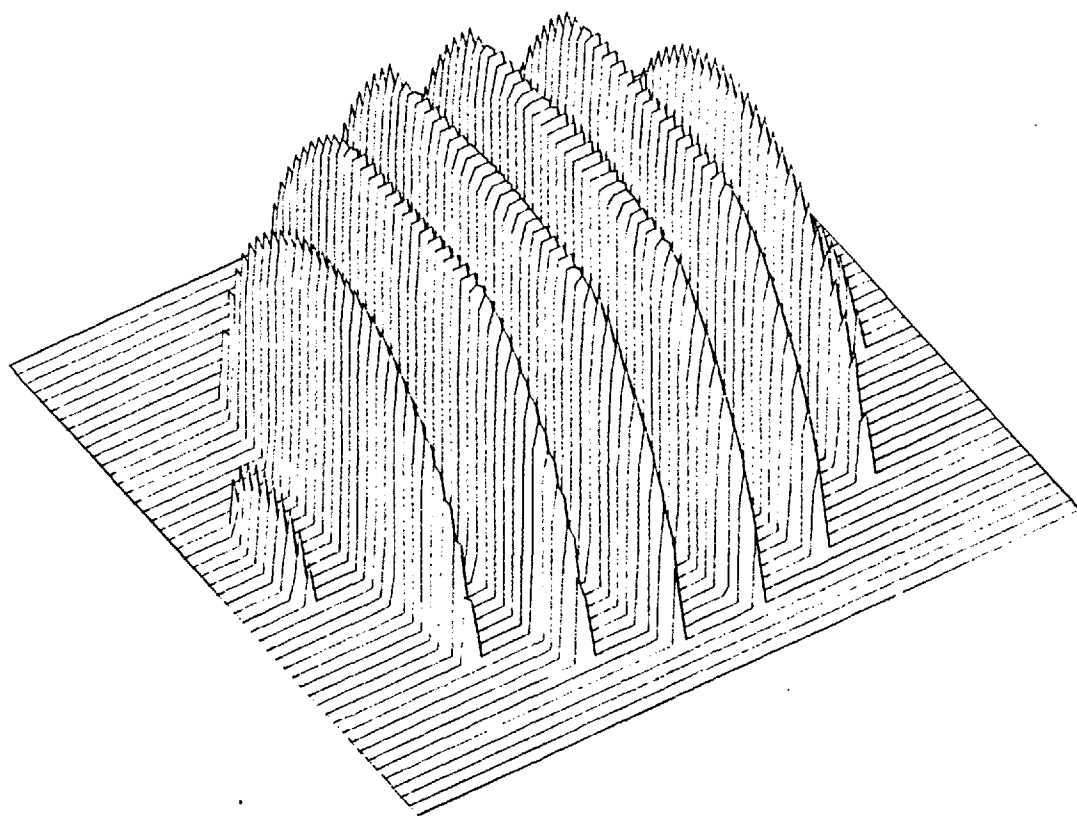


Figure 4.9 3-dimensional view of the pressure distribution and deformed geometry of a ball on a sinusoidal surface using the numerical contact model.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$
 $\Delta=0.08\text{ }\mu\text{m}$, $\lambda=24\text{ }\mu\text{m}$, Ball diameter=12.7 mm

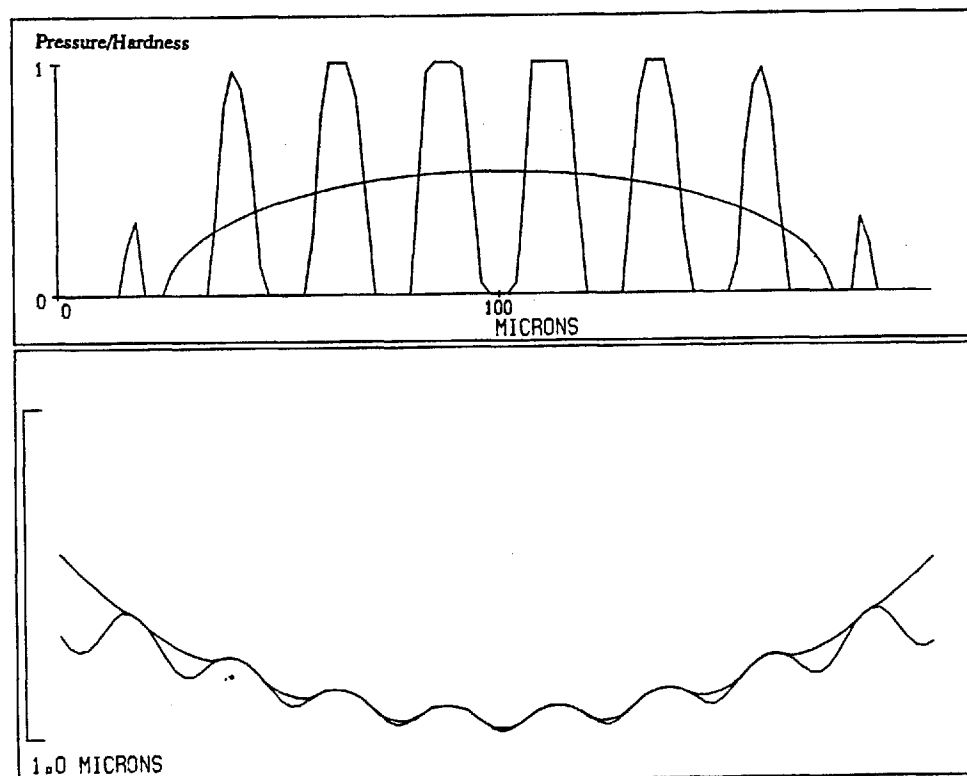


Figure 4.10 Pressure distribution and contact geometry at a section through the centre of contact shown in Figure 4.9.

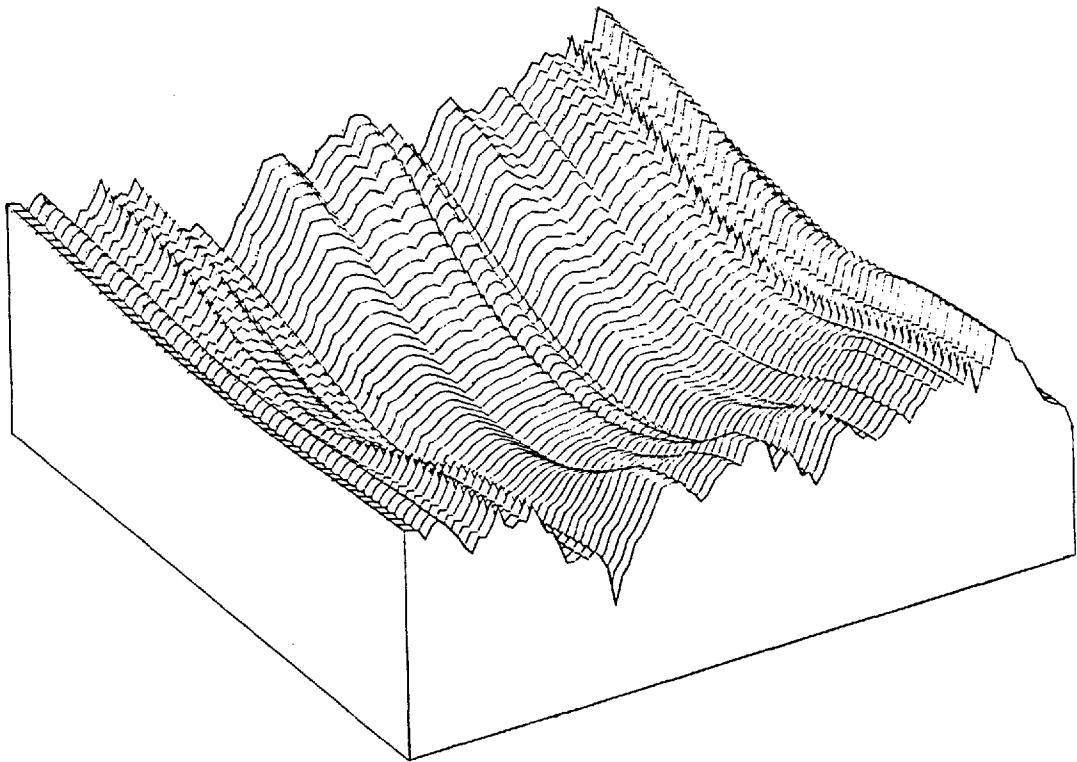
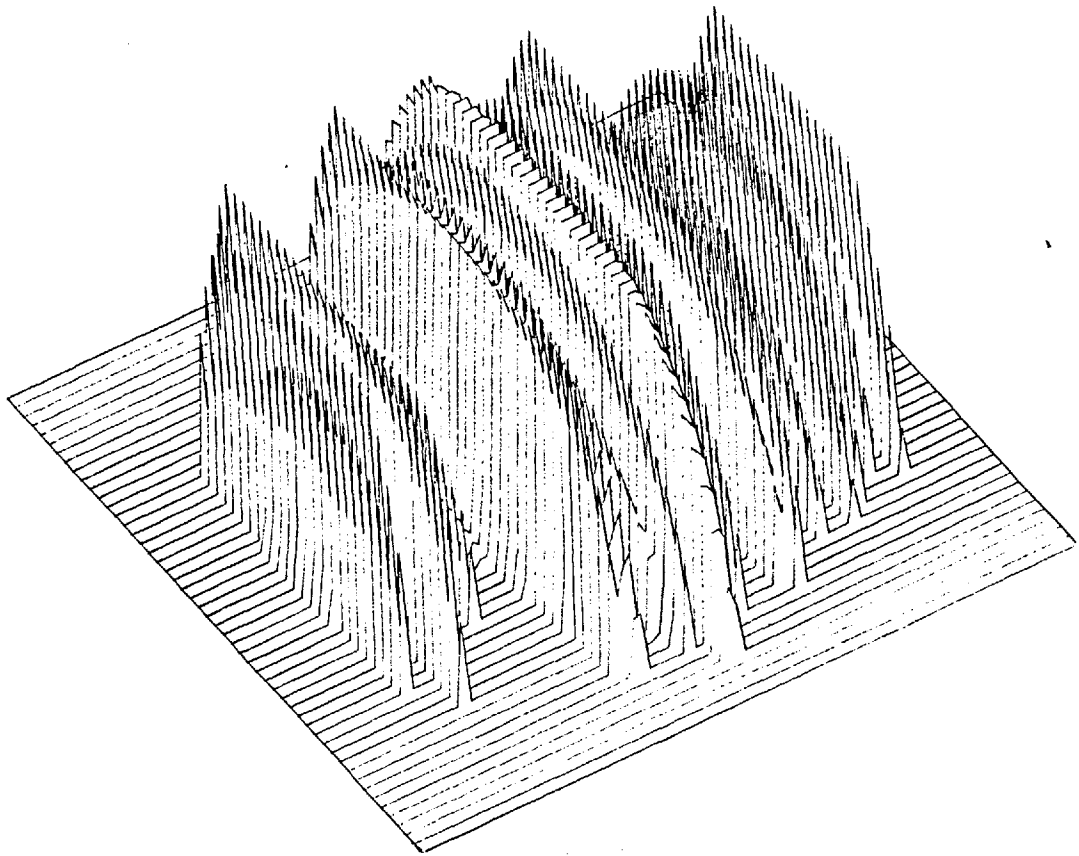


Figure 4.11 3-dimensional view of the pressure distribution and deformed geometry of a ball on a rough surface using the numerical contact model.

$p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$
 $\sigma=0.07$ μm , $\beta^*=7.4$ μm , Ball diameter=12.7 mm

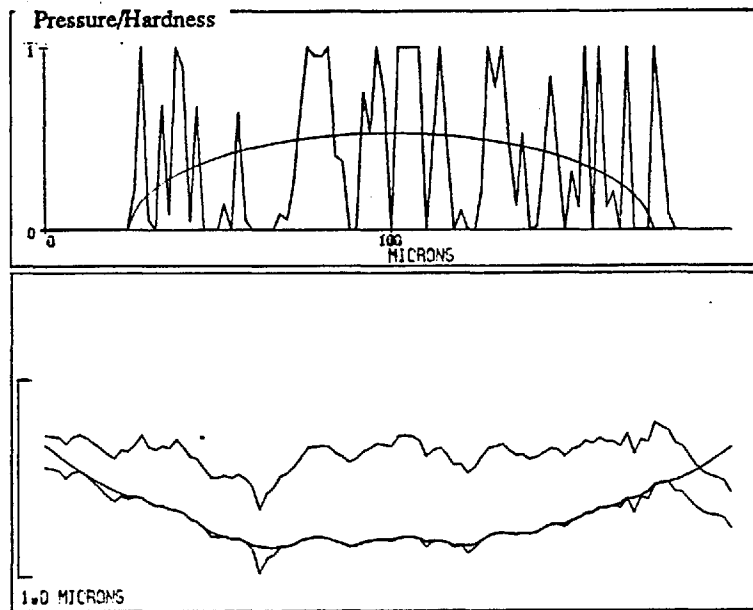


Figure 4.12 Pressure distribution and contact geometry at a section through the centre of contact shown in Figure 4.11.

CHAPTER 5

ROUGH SURFACE CONTACT ANALYSIS

5.1 Introduction

The tribological behaviour of a surface can often be related to the surface roughness. If the roughness is high, the area of intimate surface contact is usually much less than the apparent contact area and the contact stress will be inevitably high. The mode of contact with most metals will probably be plastic. On the other extreme, if the roughness is very low, the area of contact, as will be shown later, can be relatively large and approach a value equal to the nominal area such that no spacing between contact spots is available to trap lubricants or break any adhesion junction formed in the interfaces during sliding. Experience suggests that a surface offering good tribological performance and a long life in sliding usually has an even distribution of contact plateaus and spacings such as found in a cylinder bore.

For rough surface contacts, there is a need to know how the contact area, spacings and contact stresses are distributed: does the contact consist of one relatively large spot, or a large number of evenly distributed smaller spots of equal size, or (more likely) a randomly distributed number of spots of widely different sizes? As the load increases, is the increase in real area due to an increase in their number or their sizes ?

In the past, stochastic techniques have been used to study rough surface contact problems, such as the proportionality of the load-area relationship (Greenwood &

Williamson 1966; Bush et al 1975 etc), the distribution of contact areas and their size (Greenwood 1967; Sayles & Thomas 1978b; Nayak 1971, etc). The stochastic approaches usually assume that the discrete contact "spots" deform independent on one another and that the contacts tend to occur on the asperity peaks only. For nominally flat surfaces under light loads the individual contact spots may well be remote enough for these assumptions to be valid, however for concentrated contacts such as the contact of a sphere on a rough surface, it is not usually the case.

As the roughness of a surface decreases, the contacts can be expected to occur on valleys as well as on peaks due to elastic conformity of the longer wavelength features. The development of numerical techniques for elastic rough surface (Webster & Sayles 1986a) contact has enabled us to consider the interaction of deformation from all the contact points. Therefore, in this chapter, the influence of surface roughness on the distribution of real area, contact spot, spacings and contact stresses etc are studied using the numerical rough surfaces contact model described in Chapter 4.

The study is performed with a further objective in mind. The contact results will be used to investigate the influences of surface roughness on the sliding friction described in chapter 8 in which the contact of a ball on a directionally anisotropic rough surface is examined.

5.2 Contact Parameters

The material parameters for the upper body are those for an En31 bearing steel ball with a diameter of 12.7 mm ($E=207$ GPa, $H=800$ Hv, $\nu=0.3$). The lower body used is En58b stainless steel ($E=207$ GPa, $H=160$ Hv, $\nu=0.3$). The two bodies are set to approach in $0.15\text{ }\mu\text{m}$ increments until the final load is just above 10N which is equivalent to the

nominal pressure $p_n=0.56$ Gpa calculated for a circular Hertzian contact.

5.3 Surface Production and Characterisation

The surfaces were prepared using silicon carbide abrasive papers of different grit sizes and states of wear, in a way similar to that described by Hirst and Hollander (1974), to produce different values of σ and β^* . Measurements were made using a Talysurf 4 with a fine-tipped stylus, the recorded profiles being analysed on an LS1-11.23 based micro-computer. The nominal circular contact width of a ball on a smooth surface at a load of 10N is about 150 μm . Therefore, each profile consisted of 100 digitised points at a spacing of 2 μm , thus giving a profile length of 200 μm which is enough to cover the nominal width.

Each profile was subjected to an E-system high pass filter at a radius of 6.35 mm and the values of σ and β^* were calculated. A set of 11 abraded surfaces with different spatial structure were prepared, the magnification of which was changed in computer programing to obtain different values of σ , and then subjected to contact analysis. A total of 110 different profiles are made available for contact analysis by this method. The surfaces used in the analysis are summarised in Table 5.1. Each group of profiles is obtained from a measured surface profile. It can be seen that for a surface with smaller wavelength structure, the reduction in magnification does not change the β^* significantly. On the other hand, for a structure with longer wavelength structure, β^* decreases with magnification. This is due to the characteristics of the E-system filter which were discussed in Chapter 2.

Surfaces	$\sigma(\mu\text{m})$	$\beta^*(\mu\text{m})$	Surfaces	$\sigma(\mu\text{m})$	$\beta^*(\mu\text{m})$
A01	0.0014	2.79	F01	0.004	4.21
A02	0.004	4.95	F02	0.008	4.63
A03	0.008	7.17	F03	0.141	4.87
A04	0.012	9.14	F04	0.020	5.05
A05	0.024	13.5	F05	0.031	5.34
A06	0.041	18.4	F06	0.043	5.55
A07	0.061	22.5	F07	0.055	5.83
A08	0.084	24.0	F08	0.068	6.12
A09	0.013	24.4	F09	0.096	6.76
A10	0.184	24.6	F10	0.130	7.53
A11	0.250	24.8			
A12	0.275	24.8	G01	0.003	2.62
			G02	0.006	3.05
B01	0.001	2.40	G03	0.010	3.38
B02	0.003	3.90	G04	0.014	3.67
B03	0.005	9.50	G05	0.022	3.85
B04	0.008	11.3	G06	0.030	3.96
B05	0.015	13.7	G07	0.038	4.25
B06	0.024	15.5	G08	0.046	4.72
B07	0.036	17.3	G09	0.067	5.96
B08	0.05	21.0	G10	0.093	7.80
B09	0.08	25.6	G11	0.190	12.1
B10	0.11	26.8	G12	0.280	15.4
B11	0.16	27.6			
B12	0.21	28.0	H01	0.090	17.2
B13	0.25	28.1	H02	0.133	17.6
B14	0.31	28.2	H03	0.156	17.8
B15	0.36	28.2	H04	0.189	17.9
B16	0.46	28.2	H05	0.223	18.0
B17	0.57	28.1	H06	0.276	18.2
C01	0.006	3.01	I01	0.005	4.60
C02	0.015	3.40	I02	0.014	6.29
C03	0.023	3.76	I03	0.023	7.13
C04	0.032	4.00	I04	0.033	7.34
C05	0.052	4.70	I05	0.051	7.38
C06	0.071	5.10	I06	0.070	7.37
C07	0.091	5.40	I07	0.089	7.33
C08	0.111	5.64	I08	0.108	7.34
C09	0.150	5.97	I09	0.146	7.61
C10	0.190	6.25	I10	0.1855	7.87
D01	0.006	2.50	J01	0.008	1.93
D02	0.015	3.26	J02	0.176	3.66
D03	0.023	4.30	J03	0.0275	4.03
D04	0.033	4.90	J04	0.0376	4.25
D05	0.051	5.66	J05	0.0585	4.56
D06	0.070	6.55	J06	0.079	4.73
D07	0.084	6.80	J07	0.101	4.84
D08	0.108	7.32	J08	0.122	4.91
D09	0.145	7.44	J09	0.165	4.97
D10	0.182	7.54	J10	0.208	5.01
E01	0.005	2.0	K01	0.003	1.90
E02	0.118	2.80	K02	0.009	2.50
E03	0.0197	3.8	K03	0.017	3.00
E04	0.028	5.2	K04	0.027	4.50
E05	0.0464	5.8	K05	0.0455	6.80
E06	0.065	6.2	K06	0.0455	6.8
E07	0.084	7.00	K07	0.065	7.80
E08	0.103	7.8	K08	0.084	8.80
E09	0.14	10.0	K09	0.103	8.90
E10	0.177	11.0	K10	0.142	10.0
			K11	0.18	12.0

Table 5.1 Surfaces used in the numerical contact analysis.

5.4 Numerical Contact Results

5.4.1 Effect of σ

Figures 5.1 to 5.3 show the contact stresses and surface deformation at a load of $p_n=0.56$ Gpa for three surfaces I02, I06 and I09 respectively. For surface I09, the contact is predominantly plastic. For surface I06, the contact is predominantly elastic. For surface I02, the contact is also predominantly elastic but the contact is in a state of virtual complete conformity and there is little or no part of the nominal contact area remaining out of contact. It can be seen from this set of Figures that for a constant β^* , as the roughness decreases, the area of contact increases and the state of contact will change from predominantly plastic to predominantly elastic and eventually reach total conformity.

Figure 5.4 shows the variation of the real area, which includes the plastic area, and the plastic area alone, with σ . The results shown are from surfaces I02 to I09 of which the values of β^* can be considered as constant. As σ increases the real area decreases and the proportion of plastic area increases. At large σ , the real area and plastic area appear to converge to a common value indicating that the area is predominantly plastic.

There is no simple relationship defining the real area or plastic area with σ . An attempt to fit a curve of the form $y=kx^n+c$ has been fruitless. At a lower load of $p_n=0.354$ Gpa, the general trend of the curve appears similar.

5.4.2 Effect of β^*

Figure 5.5 to 5.7 show the contact stresses and surface deformation at a load of $p_n=0.56$ Gpa for three surfaces J07, H01 and B10 respectively. Again there is a tendency

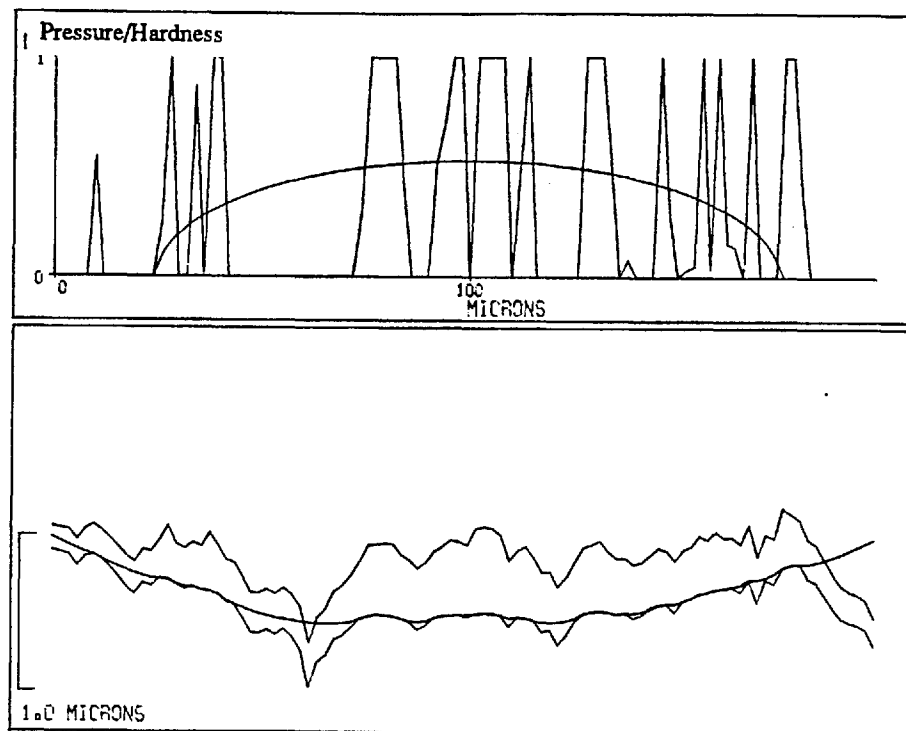


Figure 5.1 Numerically calculated distribution and deformed geometry of surface I09. $\sigma=0.146 \mu\text{m}$, $\beta^*=7.61 \mu\text{m}$.
 $p_n=0.56 \text{ Gpa}$, $H=160 \text{ Hv}$, $E_1=E_2=207 \text{ Gpa}$, $\nu_1=\nu_2=0.3$

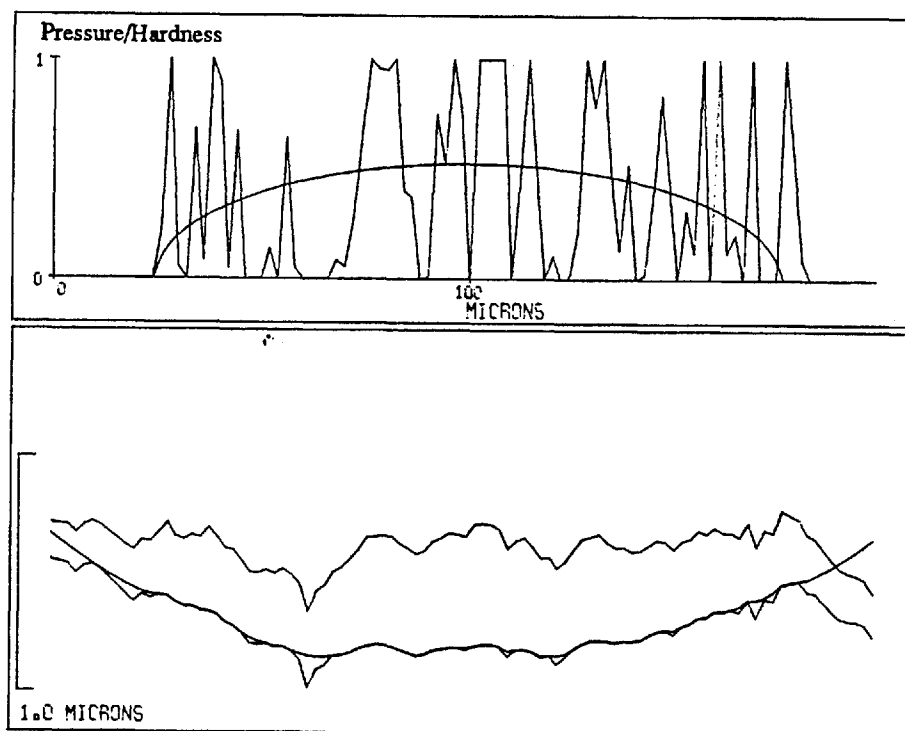


Figure 5.2 Numerically calculated distribution and deformed geometry of surface I06. $\sigma=0.07 \mu\text{m}$, $\beta^*=7.37 \mu\text{m}$.
 $p_n=0.56 \text{ Gpa}$, $H=160 \text{ Hv}$, $E_1=E_2=207 \text{ Gpa}$, $\nu_1=\nu_2=0.3$

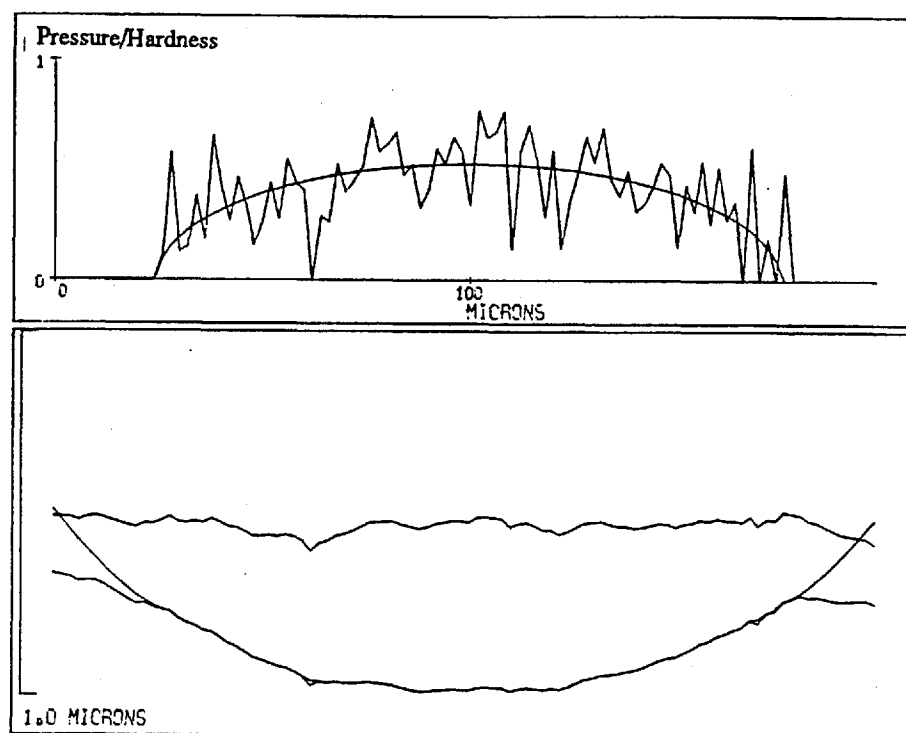


Figure 5.3 Numerically calculated distribution and deformed geometry of surface I02. $\sigma=0.014 \mu\text{m}$, $\beta^*=6.29 \mu\text{m}$. $p_n=0.56 \text{ Gpa}$, $H=160 \text{ Hv}$, $E_1=E_2=207 \text{ Gpa}$, $\nu_1=\nu_2=0.3$

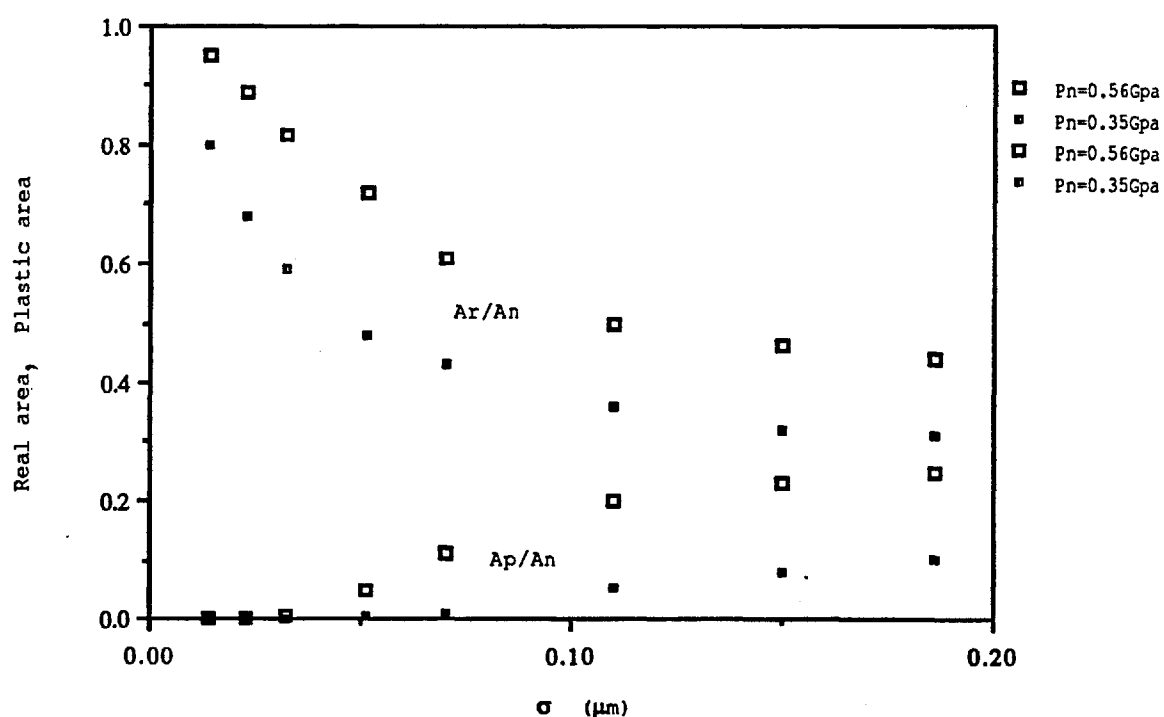


Figure 5.4 Variation of real and plastic area with σ at a constant β^* .

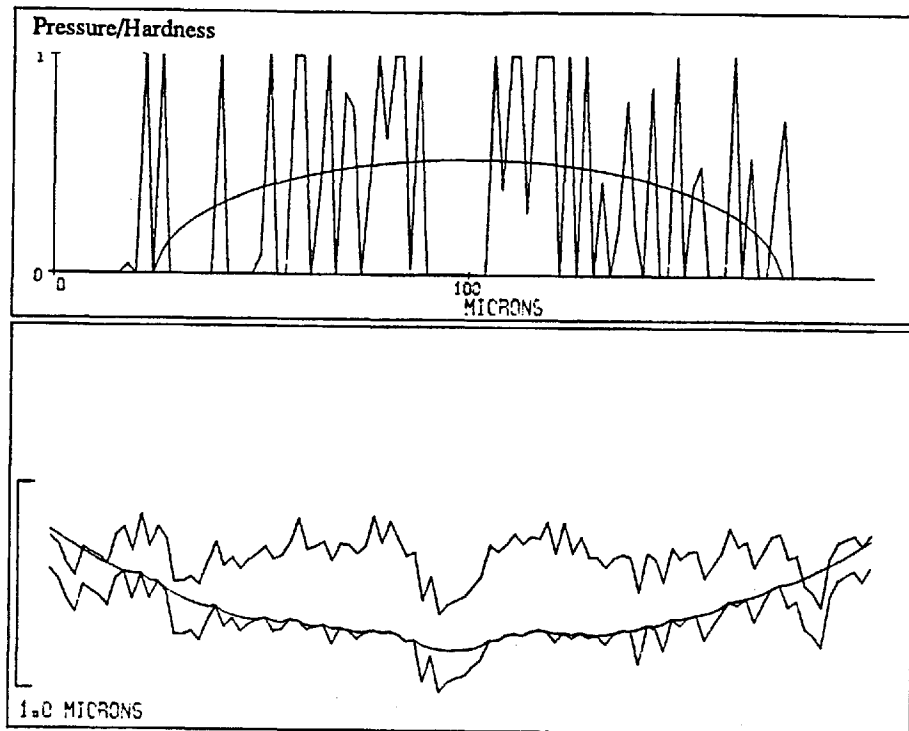


Figure 5.5 Numerically calculated distribution and deformed geometry of surface J07. $\sigma=0.1 \mu\text{m}$, $\beta^*=4.84 \mu\text{m}$. $p_n=0.56 \text{ Gpa}$, $H=160 \text{ Hv}$, $E_1=E_2=207 \text{ Gpa}$, $\nu_1=\nu_2=0.3$

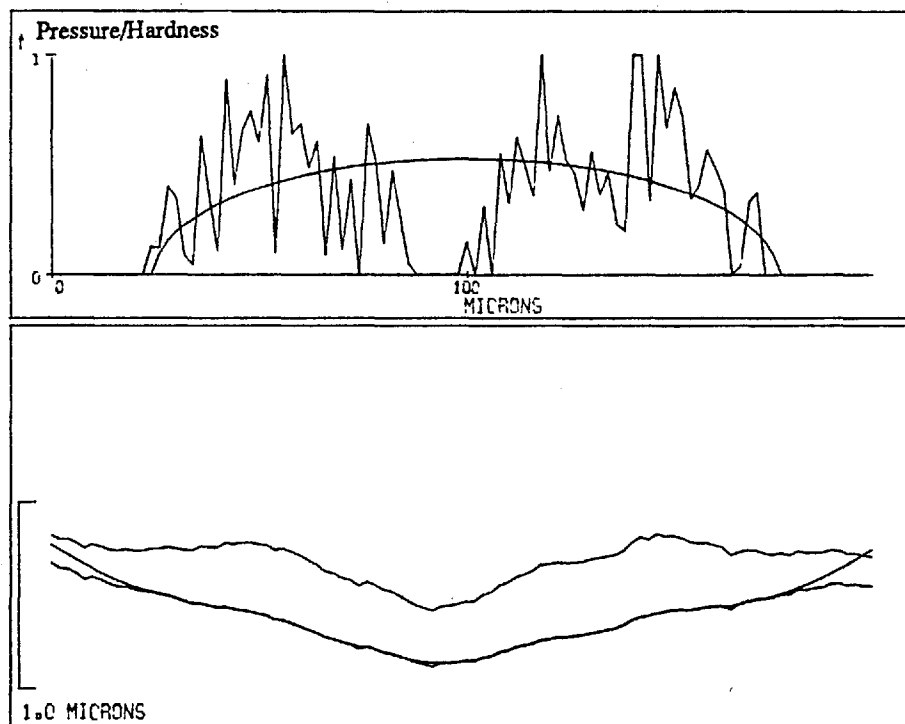


Figure 5.6 Numerically calculated distribution and deformed geometry of surface H01. $\sigma=0.09 \mu\text{m}$, $\beta^*=17.2 \mu\text{m}$. $p_n=0.56 \text{ Gpa}$, $H=160 \text{ Hv}$, $E_1=E_2=207 \text{ Gpa}$, $\nu_1=\nu_2=0.3$

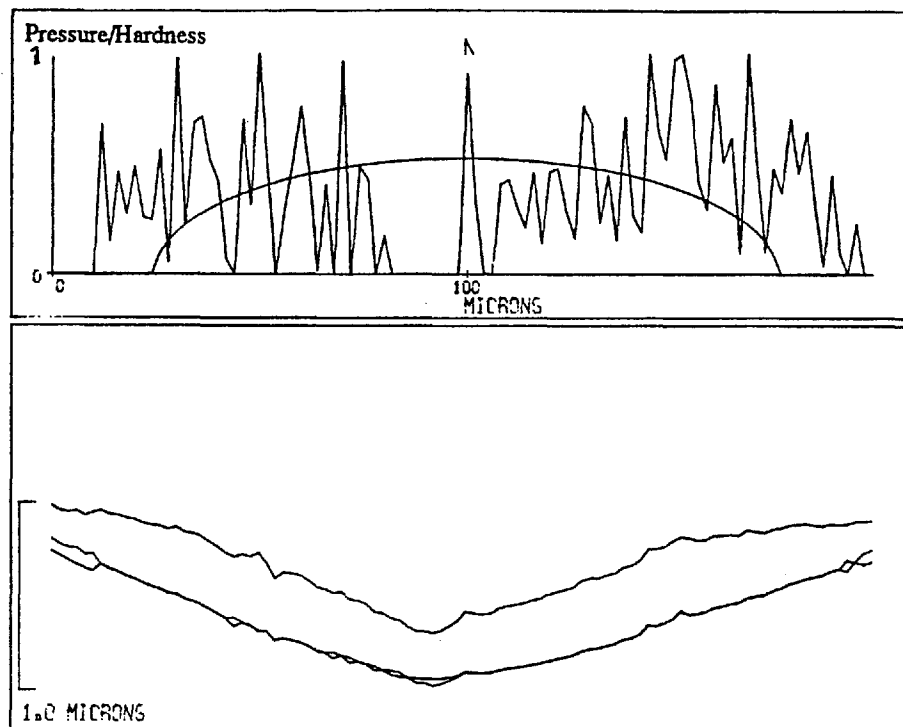


Figure 5.7 Numerically calculated distribution and deformed geometry of surface B10. $\sigma=0.11 \mu\text{m}$, $\beta^*=26.8 \mu\text{m}$. $p_n=0.56 \text{ Gpa}$, $H=160 \text{ Hv}$, $E_1=E_2=207 \text{ Gpa}$, $\nu_1=\nu_2=0.3$

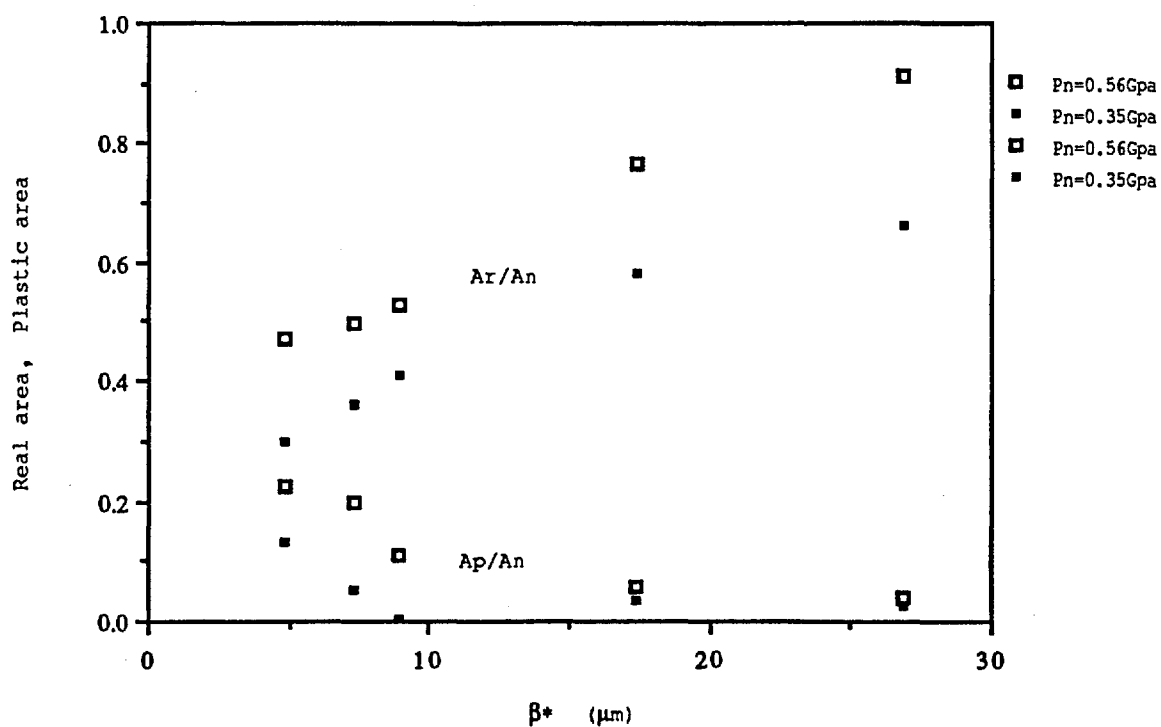


Figure 5.8 Variation of real area and plastic area with β^* at a contact σ .

to three types of contact. As β^* increases, the contact changes from predominantly plastic to predominantly elastic. At large β^* , the contact tends to a state of total conformity.

Figure 5.8 shows the variation of real area and plastic area with β^* . The results were obtained from surfaces J07, I08, K09, H01 and B10 where $\sigma \sim 0.1 \mu\text{m}$, and the contact is performed at a load of $p_n = 0.56 \text{ GPa}$. The real area increases and the plastic area reduces with β^* . At small β^* , the contact is predominantly plastic but as β^* increases, the contact becomes predominantly elastic and tends to a state of total conformity ($A_r/A_n = 1$).

There is again no simple relationship for the real area or plastic area with β^* . An attempt to fit a curve of the form $y = kx^n + c$ has been fruitless. At a lower load of $p_n = 0.354 \text{ GPa}$, the general trend of the curve appears similar.

5.4.3 Effect of σ/β^*

From the previous results, it is to be expected that the contact area decreases as σ/β^* increases. In the characterisation of rough surfaces, it is found that many surface contact conditions can be related to the ratio σ/β^* . Thus it is of interest to know if the contact area is dependent on σ/β^* only. Figure 5.9 shows the real and plastic area against σ/β^* of all 110 surface profiles at a load of $p_n = 0.56 \text{ GPa}$. The results immediately show that both the real area and plastic area seem to depend on σ/β^* , and not on the individual values of σ or β^* . At large σ/β^* , the contact is predominantly plastic and the real area is less dependent on σ/β^* . At a very small σ/β^* , there exists a small but finite value of σ/β^* below which the real area is equal to the nominal area.

Figure 5.10 shows the distribution of real area on a σ - β^* map. A general linear trend is evident for each value of A_r/A_n except at small β^* . This may be explained by the

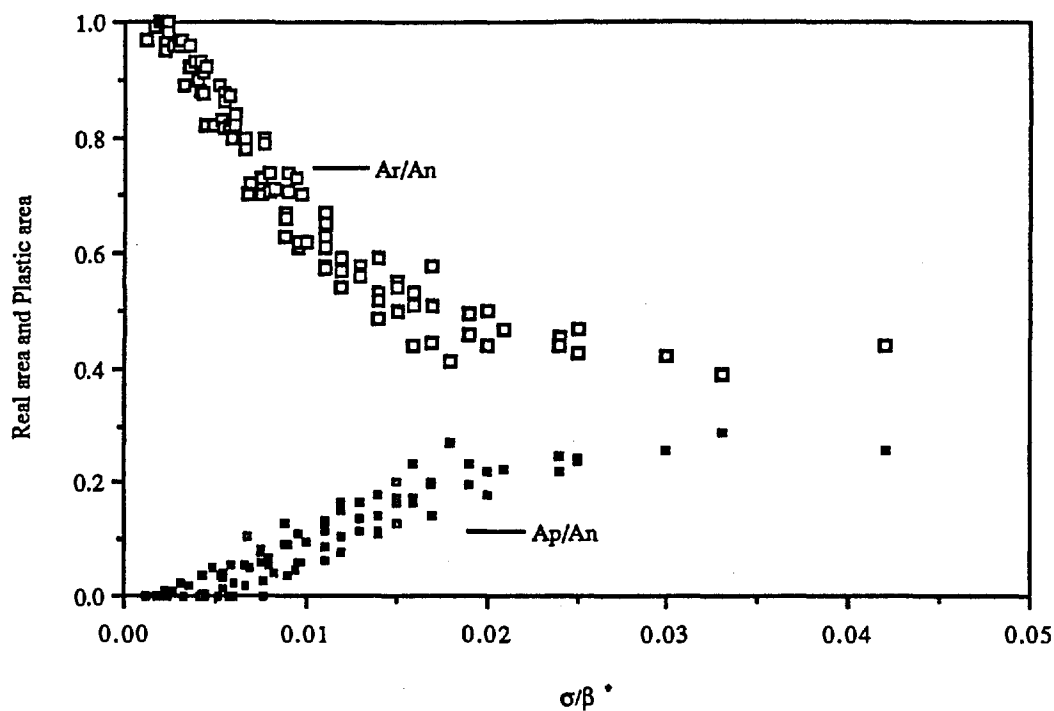


Figure 5.9 Variation of real area and plastic area with σ/β^* for 110 surfaces with different combination of σ and β^* .

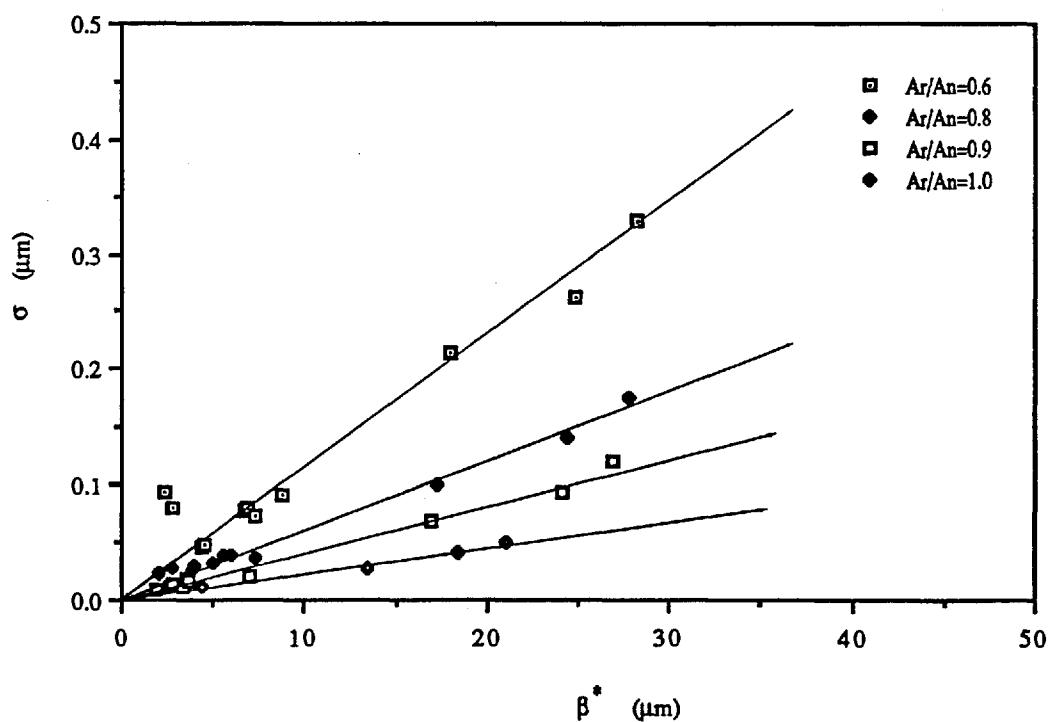


Figure 5.10 Points of constant A_r/A_n are plotted on a σ - β^* map. A straight line is drawn through the points for each value of A_r/A_n .

effect of the finite sampling techniques inherent in the numerical methods employed. Figure 5.11 shows the effect of sampling interval τ on the real area of two surfaces with $\beta^*=5\text{ }\mu\text{m}$ and $30\text{ }\mu\text{m}$ respectively. The area for different τ is normalised by the value obtained at $\tau=2\text{ }\mu\text{m}$. For a surface with a small β^* , the rate of increase in real area with τ is greater than that of the surface with a higher β^* .

The numerical results were obtained at a $\tau=2\text{ }\mu\text{m}$. In reality, surface undulation exists even below $\tau=2\text{ }\mu\text{m}$, therefore overestimation in real area will be inevitable.

By ignoring the points at small β^* , a straight line can be drawn for a particular value of A_r/A_n , i.e. the distribution of real area is largely dependent on σ/β^* . Similarly, a plot of the distribution of plastic area on a σ - β^* map is shown in Figure 5.12. A straight line approximation can also be used for a particular value of A_p/A_n .

5.4.4 Effect of load

It is generally believed that the real area is directly proportional to load for light loads. In this section, the proportionality of ^{the} load-area relationship is investigated. Figure 5.13 and 5.14 show the variation of real area and plastic area with load for different σ . The results are obtained from surfaces I01 to I10. In all cases proportionality is seen to be valid over a limited range of pressure and this is usually the low pressure region. However in some cases proportionality extends over a wide range of loads.

Figure 5.15 and 5.16 show the variation of real area and plastic area with load for different β^* . The results are obtained from surfaces J07, I08, H01 and B10. Again, in general proportionality is limited to the light loads.

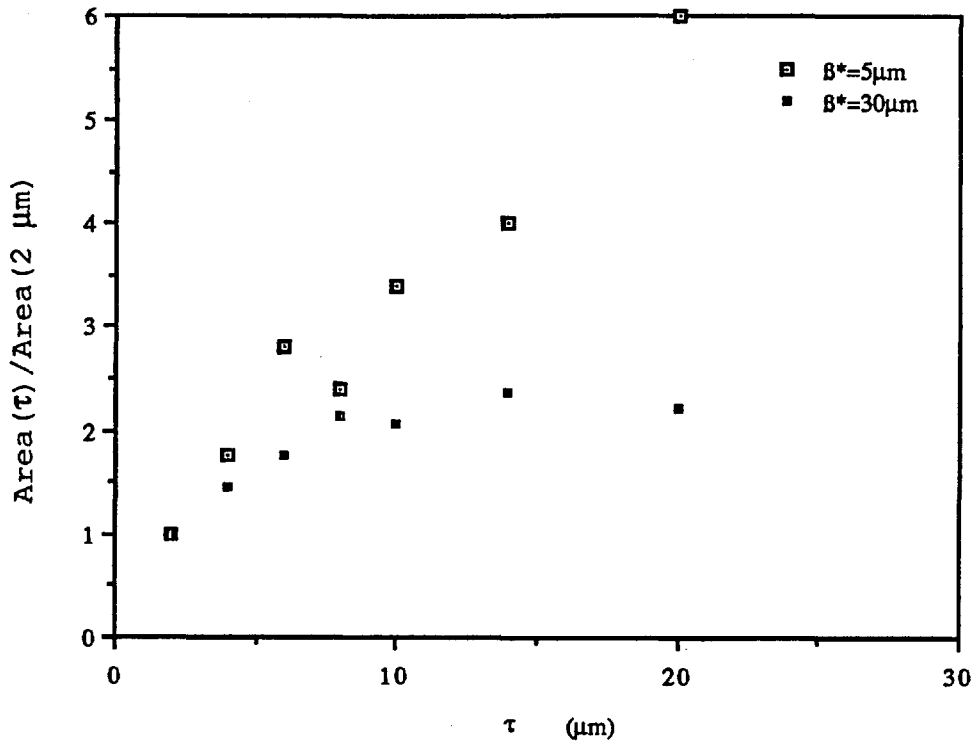


Figure 5.11 Variation of real area with sampling interval using the numerical contact analysis on two surfaces with different β^* .

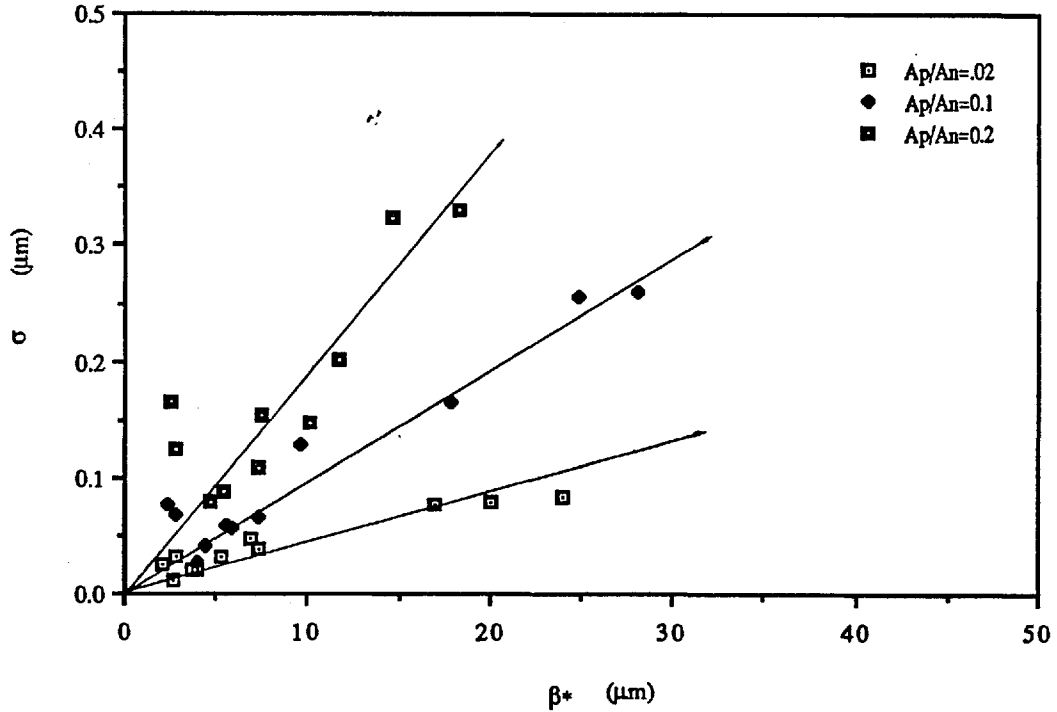


Figure 5.12 Points of constant A_p/A_n are plotted on a σ - β^* map. A straight line is drawn through the points for each value of A_p/A_n .

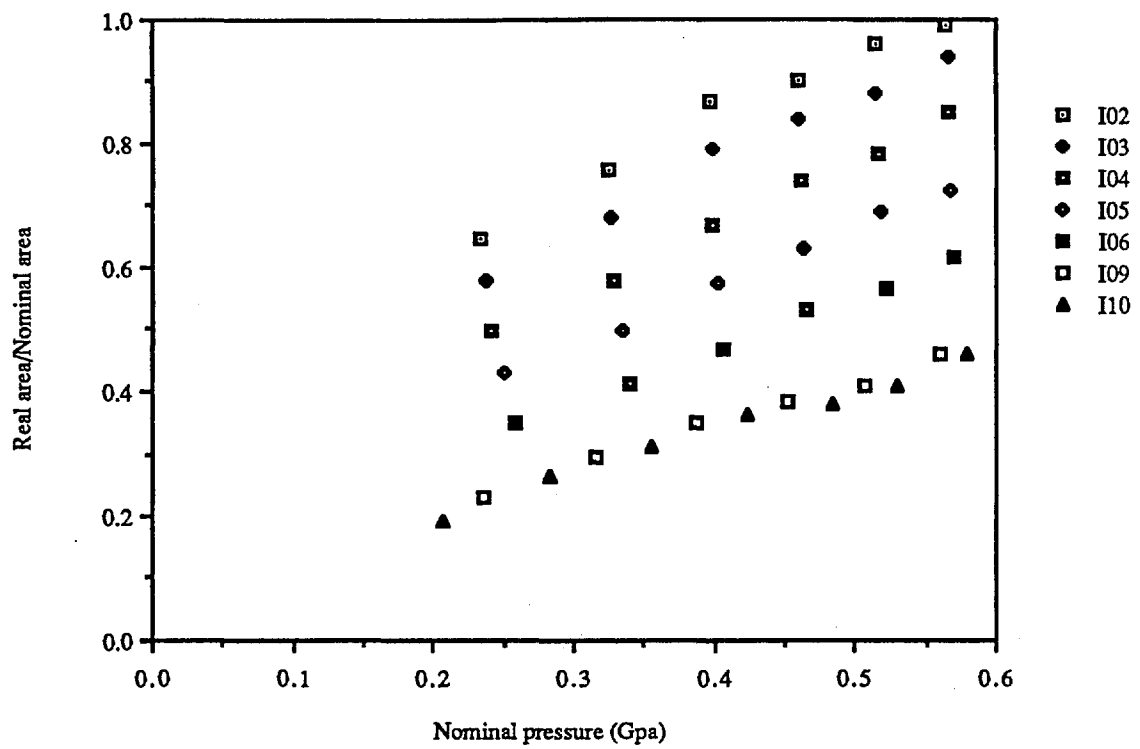


Figure 5.13 Variation of real area with load for different σ .

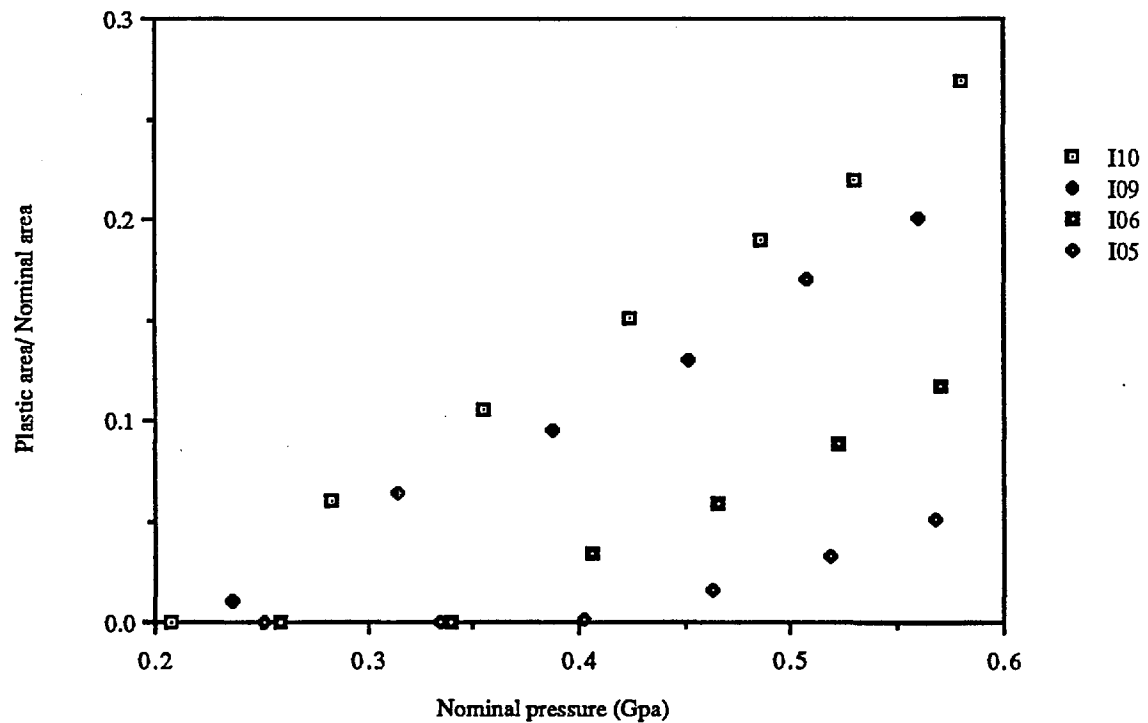


Figure 5.14 Variation of plastic area with load for different σ .

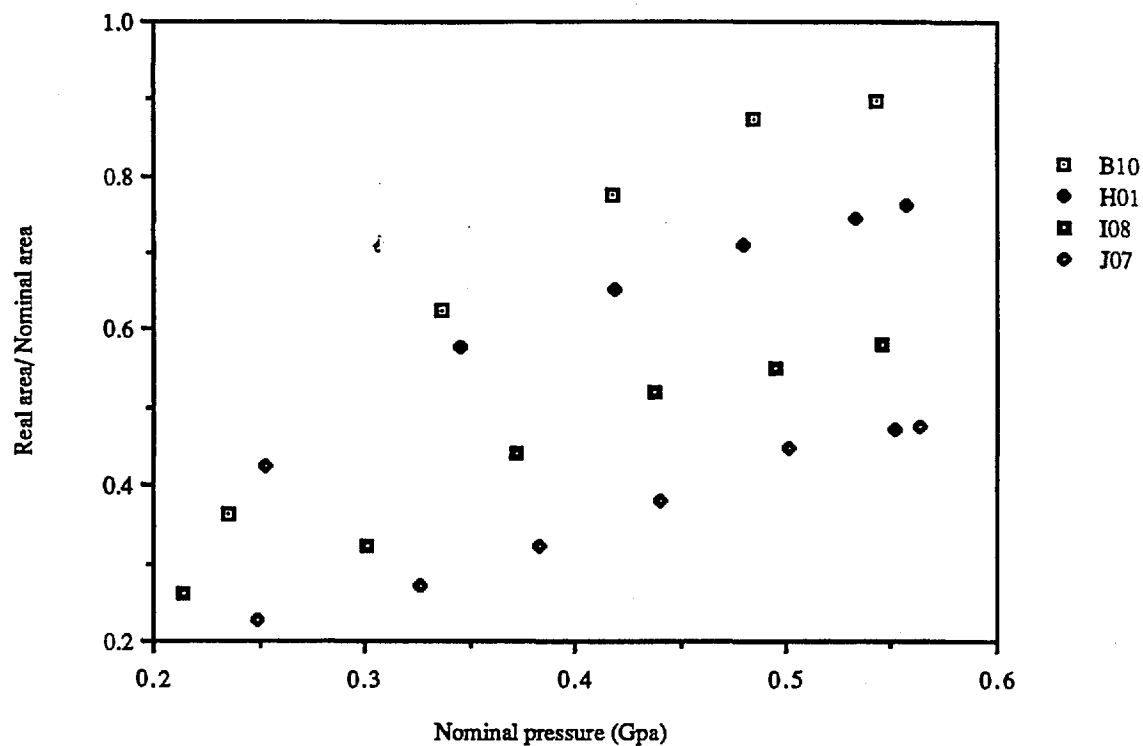


Figure 5.15 Variation of real area with load for different β^* .

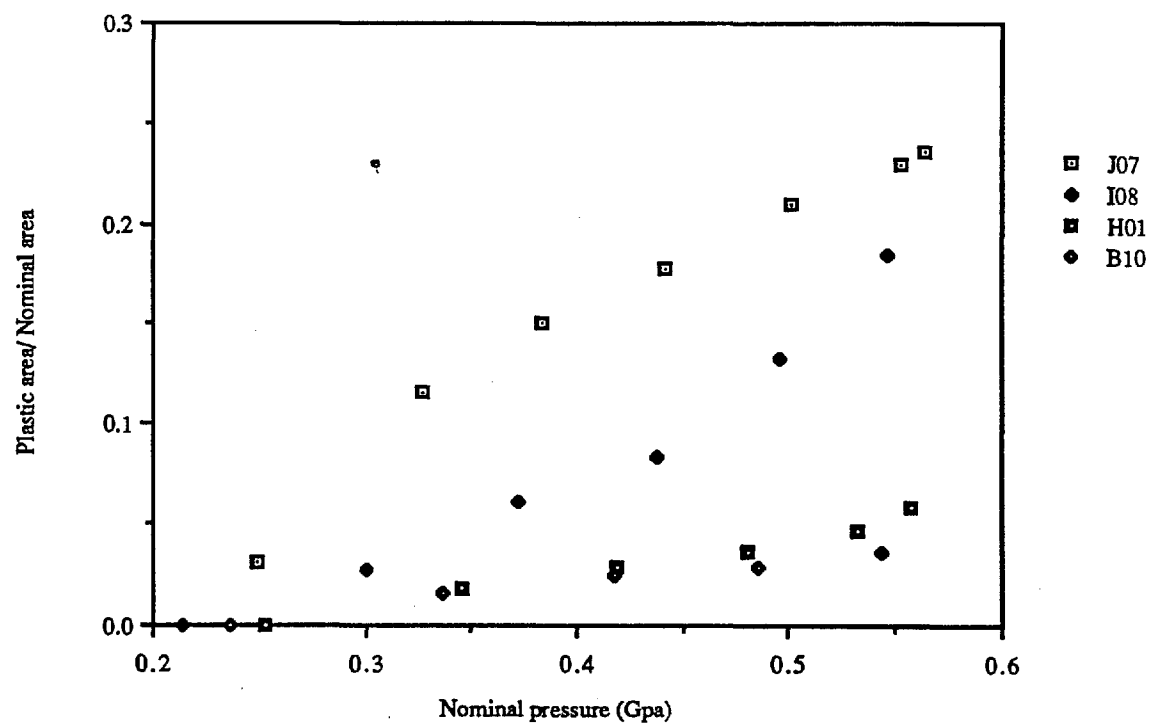


Figure 5.16 Variation of plastic area with load for different β^* .

5.4.5 Distribution of Contact Areas, Pressures and Spacings

In this section, the distribution of contact areas, contact pressures and spacings for different σ/β^* are examined. In order to have enough sample data for the distribution curves to be representative, contact results obtained from 7 to 8 surfaces with the same ratio of σ/β^* at a load of $p_n=0.56$ Gpa are used.

Figure 5.17 to 5.24 show the distribution of contact pressures for different σ/β^* . The contact pressure is the average pressure in the lay direction. For large σ/β^* , the contact is predominantly plastic and for small σ/β^* , the contact is predominantly elastic. For $\sigma/\beta^* \leq .005$, the pressure distribution tends to a Gaussian form as shown in Figures 5.22 to 5.24. For each distribution, the mean pressure is calculated and the results are shown in Figure 5.25. (The mean pressure is the mean value over the local contact spot pressures and is not the nominal pressure). The mean pressure is seemingly proportional to σ/β^* .

Figure 5.26 to Figure 5.33 show the distribution of contact areas for different σ/β^* . The log-normal distribution curve is superimposed for comparison. For large σ/β^* , the mode of distribution tends to left. This means that the contact consists of a relatively large number of smaller contact spots. For an intermediate value of σ/β^* , for example $0.006 < \sigma/\beta^* < 0.013$, the distribution is closely log-normal. As σ/β^* becomes smaller, the mode of distribution tends to right significantly. This indicates the contact is dominated by some larger contact spots. This can be interpreted as the individual contact spots merging together to form a larger spot as the surface becomes smoother.

Figure 5.34 shows the cumulative area distribution for different values of σ/β^* . The graph is scaled such that a straight line is produced when the distribution is log-normal. It can be seen that the distribution is closely log-normal for large σ/β^* . As σ/β^* becomes very small, the distribution does not follow a log-normal form. This may be due to the

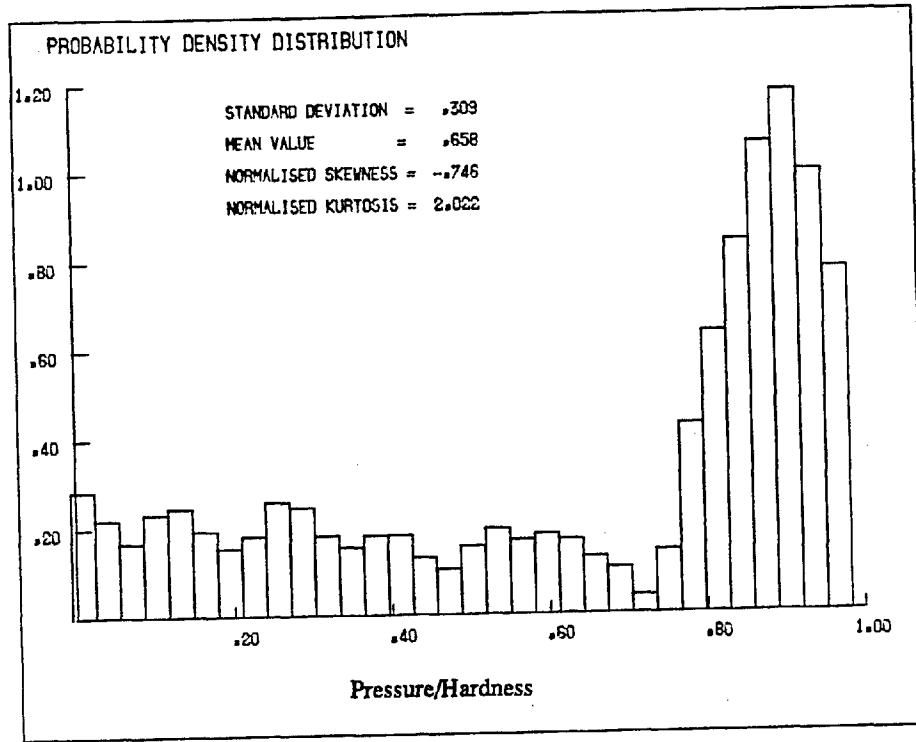


Figure 5.17 Pressure distribution for $\sigma/\beta^*=0.022$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

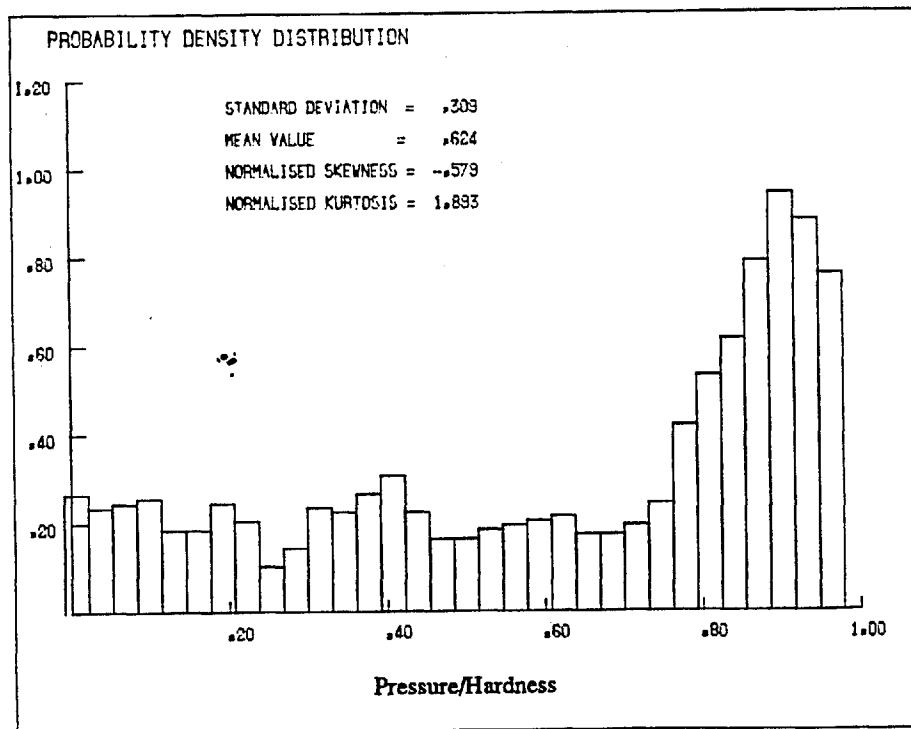


Figure 5.18 Pressure distribution for $\sigma/\beta^*=0.017$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

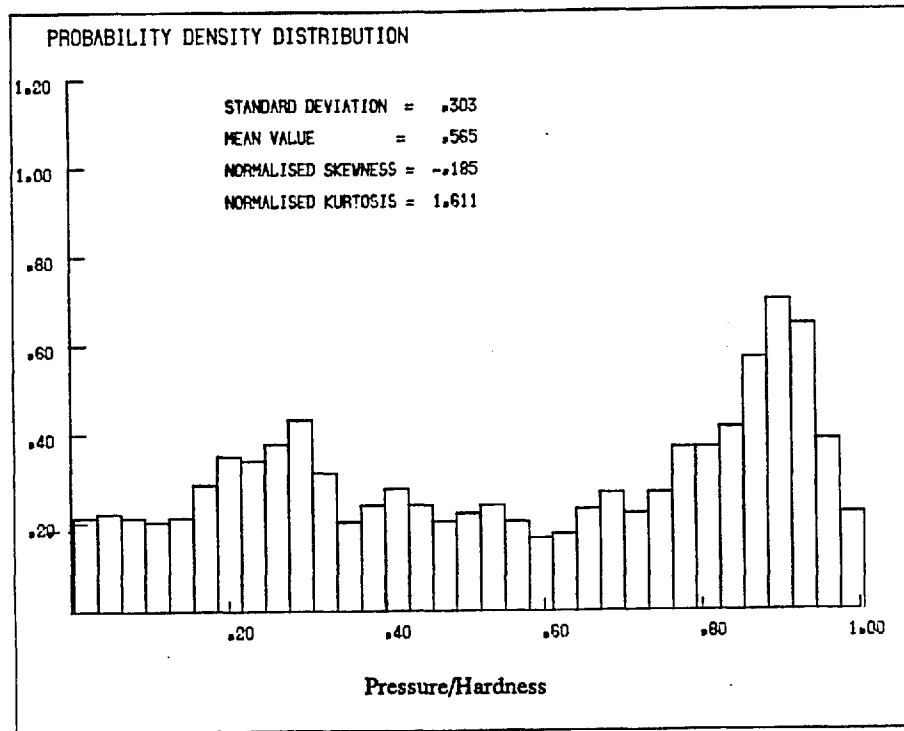


Figure 5.19 Pressure distribution for $\sigma/\beta^*=0.015$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

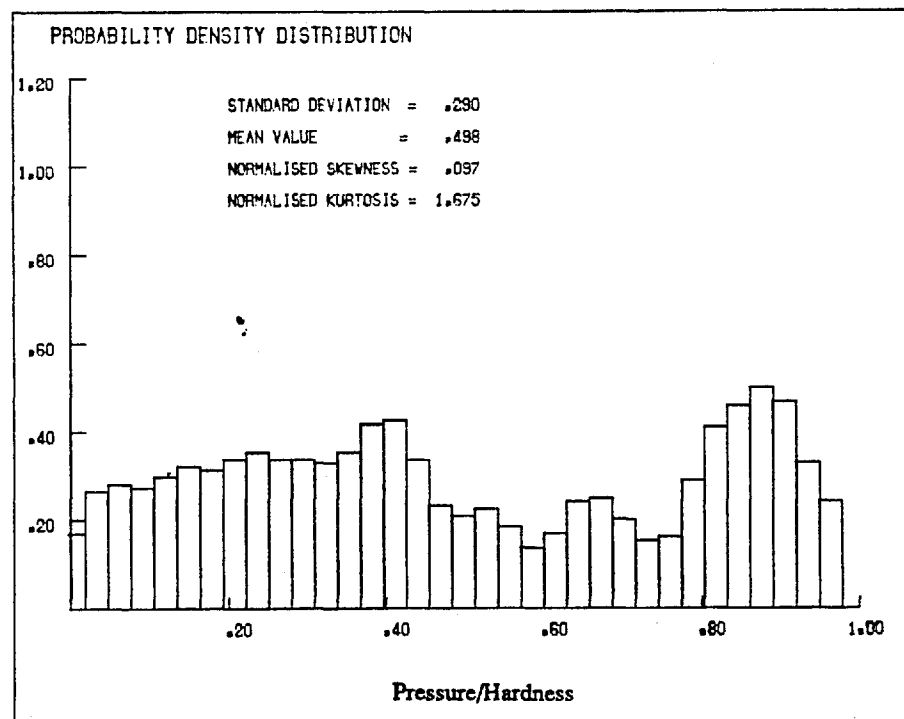


Figure 5.20 Pressure distribution for $\sigma/\beta^*=0.0126$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

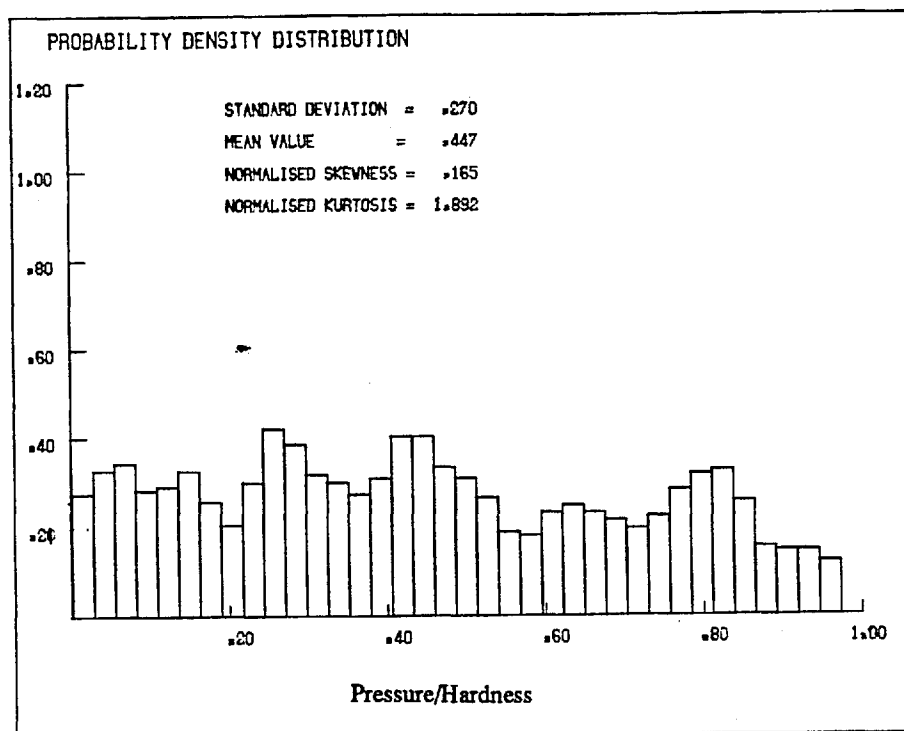


Figure 5.21 Pressure distribution for $\sigma/\beta^*=0.0064$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

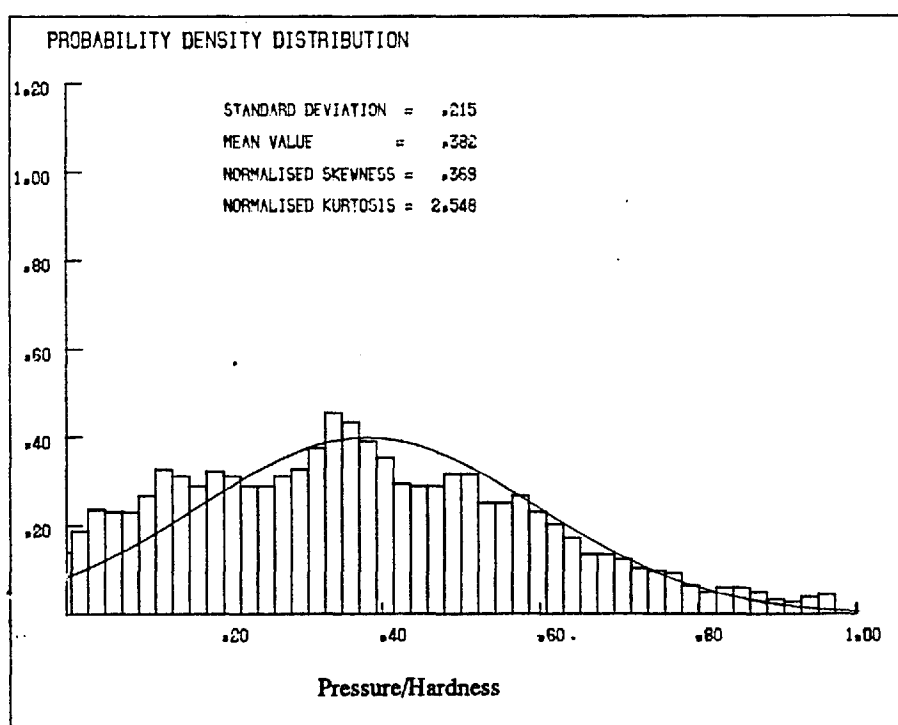


Figure 5.22 Pressure distribution for $\sigma/\beta^*=0.0047$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

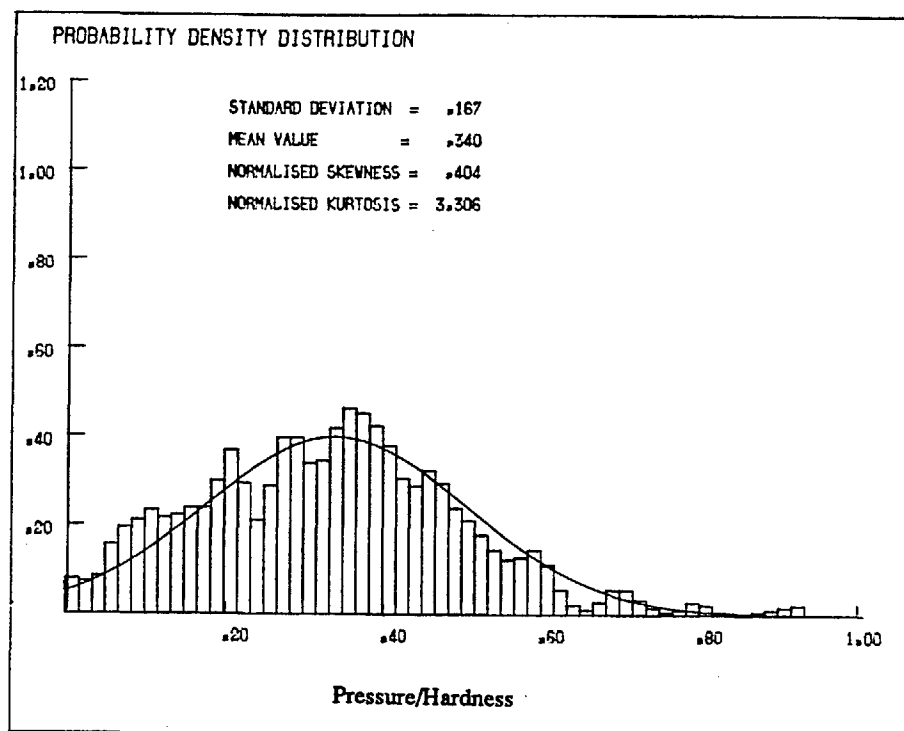


Figure 5.23 Pressure distribution for $\sigma/\beta^*=0.0028$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

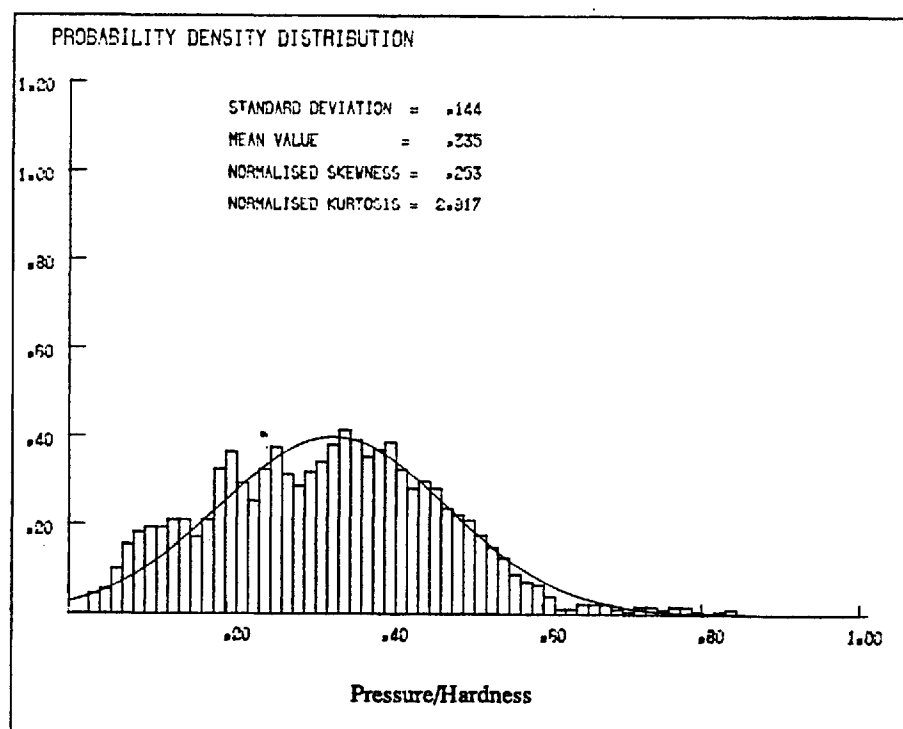


Figure 5.24 Pressure distribution for $\sigma/\beta^*=0.0013$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

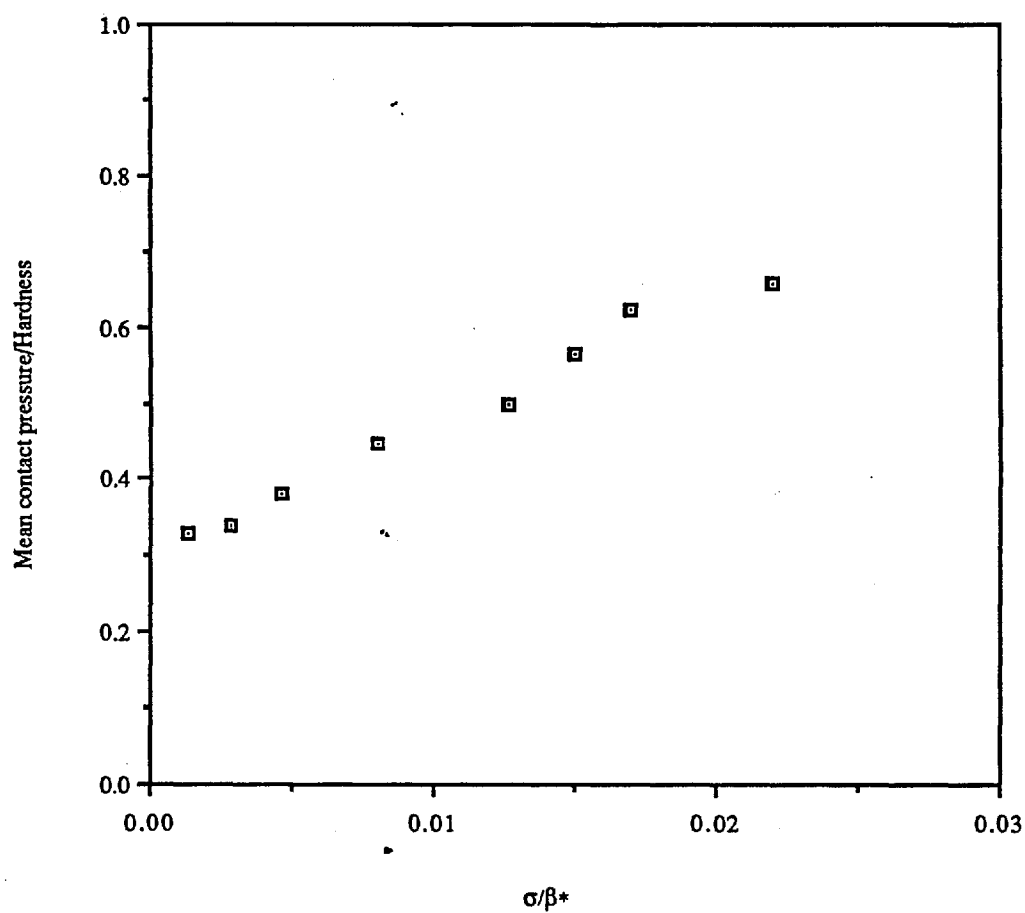


Figure 5.25 Variation of mean contact pressure with σ/β^* .
 $P_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

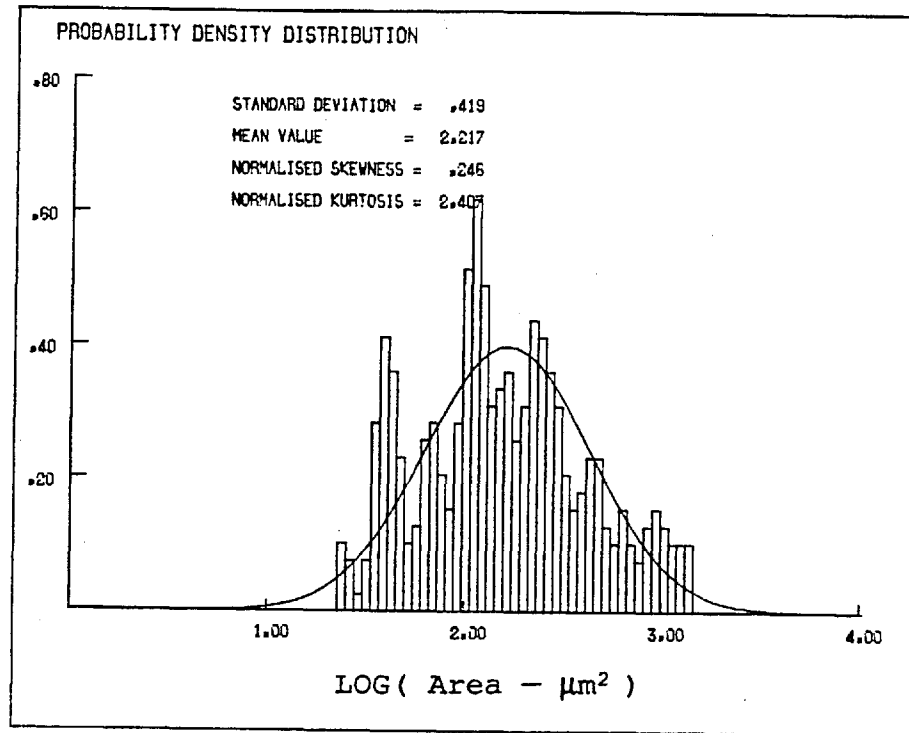


Figure 5.26 Contact area distribution for $\sigma/\beta^*=0.022$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

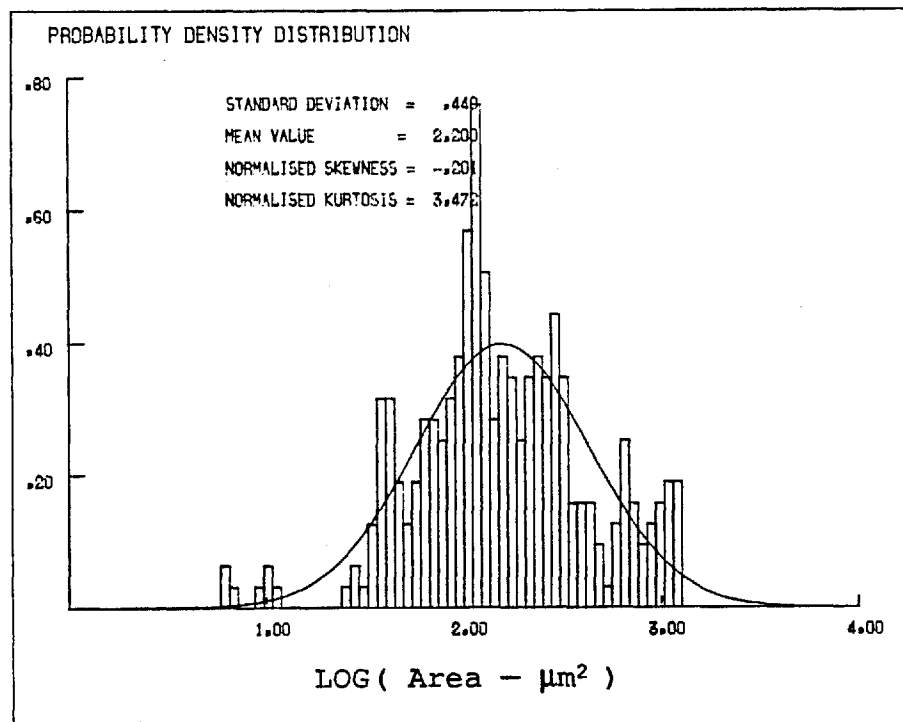


Figure 5.27 Contact area distribution for $\sigma/\beta^*=0.017$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

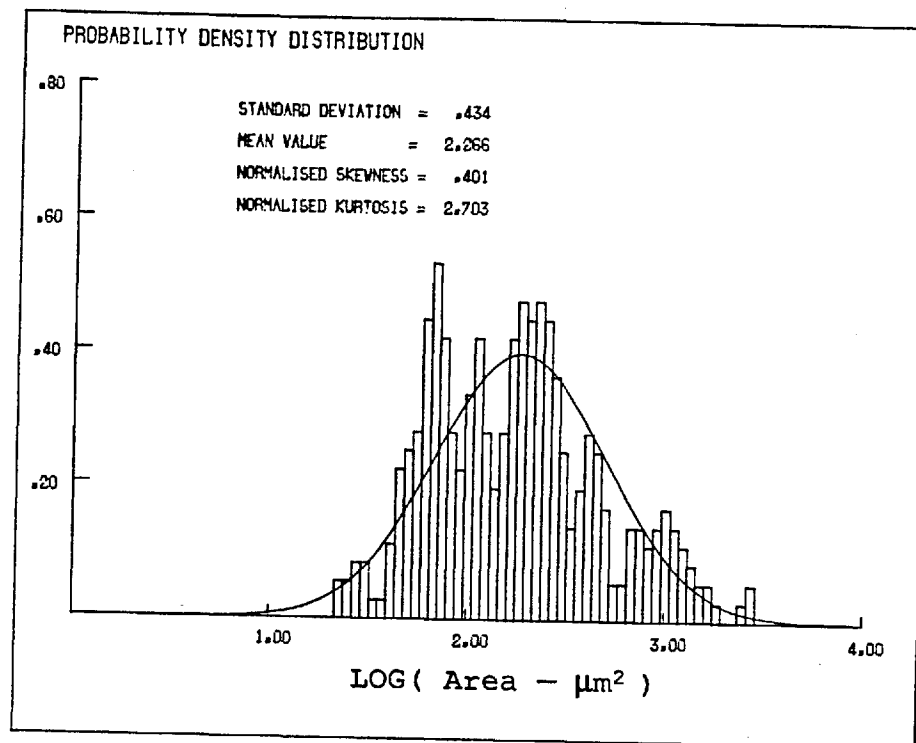


Figure 5.28 Contact area distribution for $\sigma/\beta^*=0.015$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

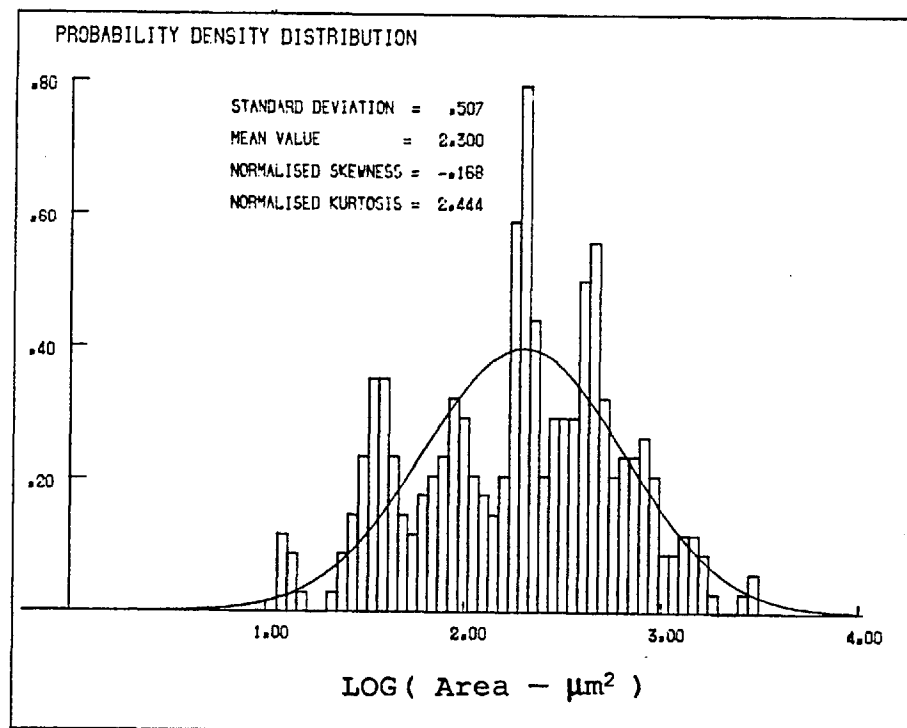


Figure 5.29 Contact area distribution for $\sigma/\beta^*=0.0126$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

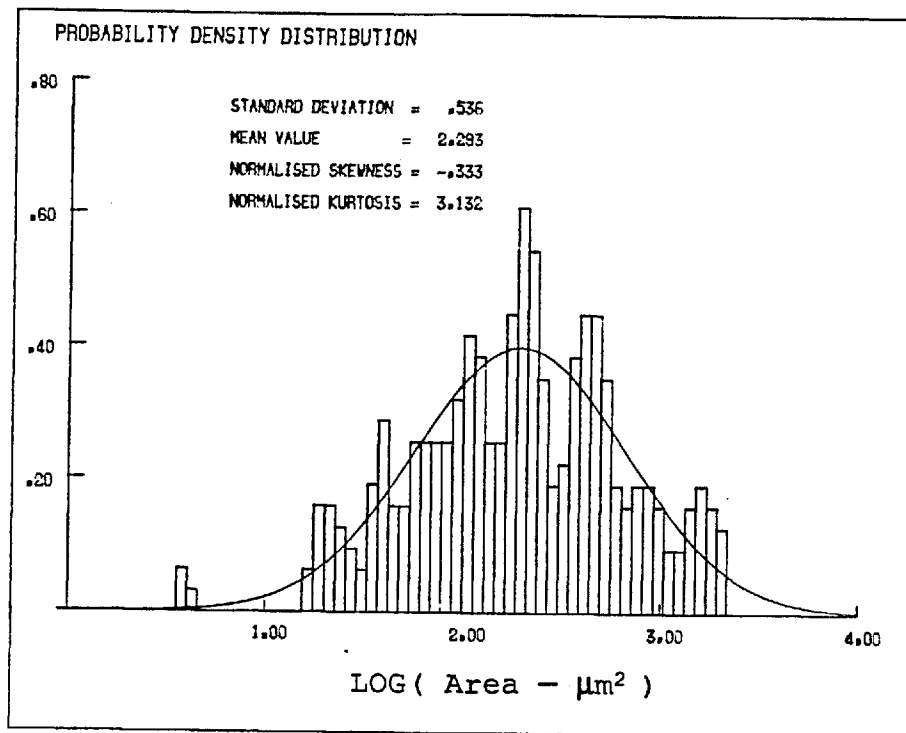


Figure 5.30 Contact area distribution for $\sigma/\beta^*=0.0064$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

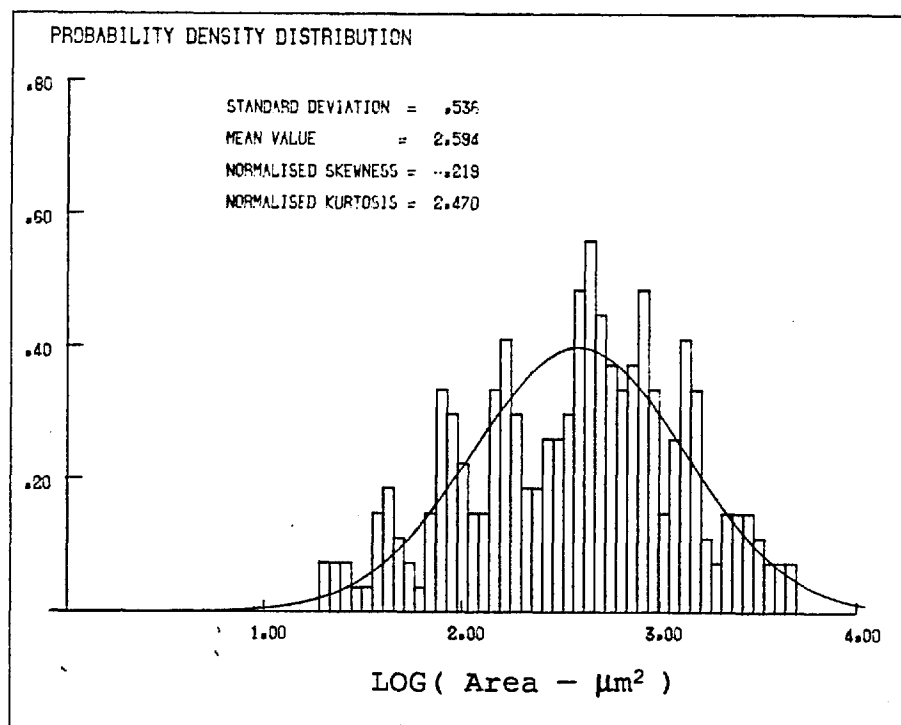


Figure 5.31 Contact area distribution for $\sigma/\beta^*=0.0047$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

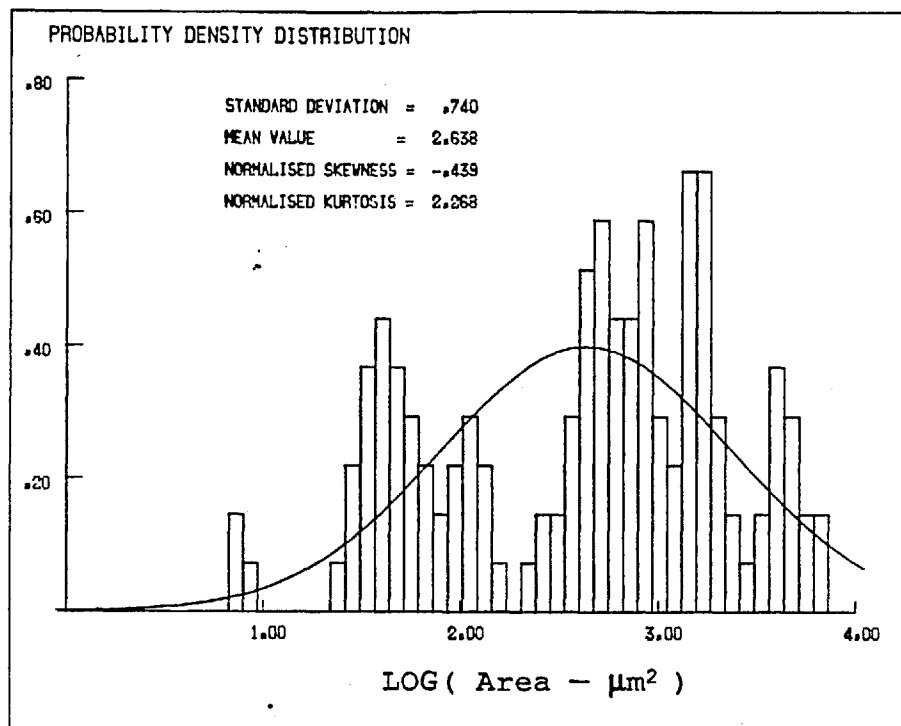


Figure 5.32 Contact area distribution for $\sigma/\beta^*=0.0028$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

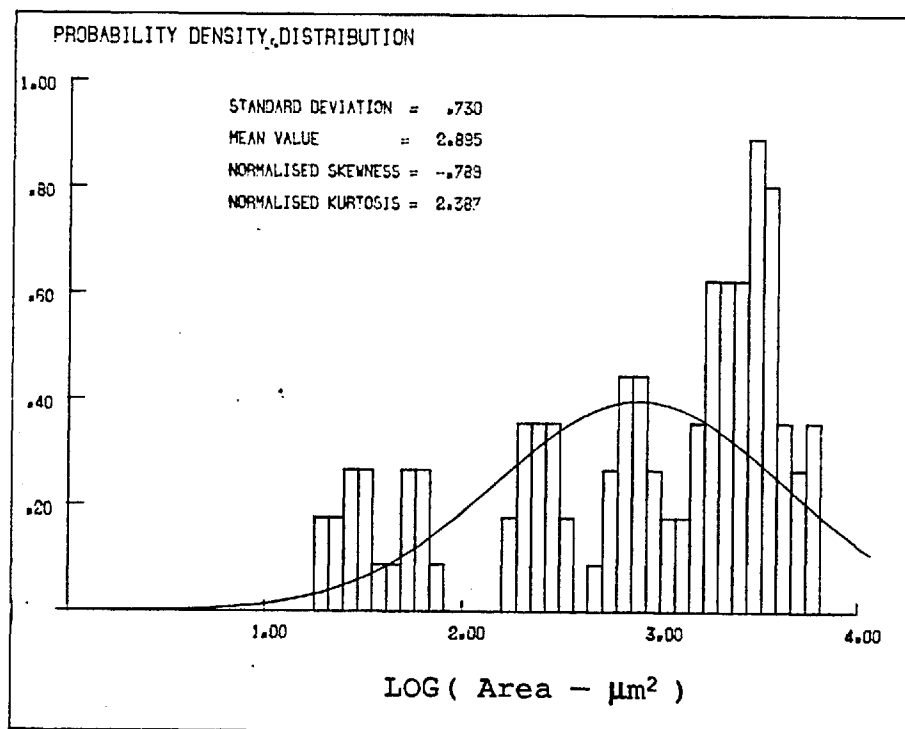


Figure 5.33 Contact area distribution for $\sigma/\beta^*=0.0013$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

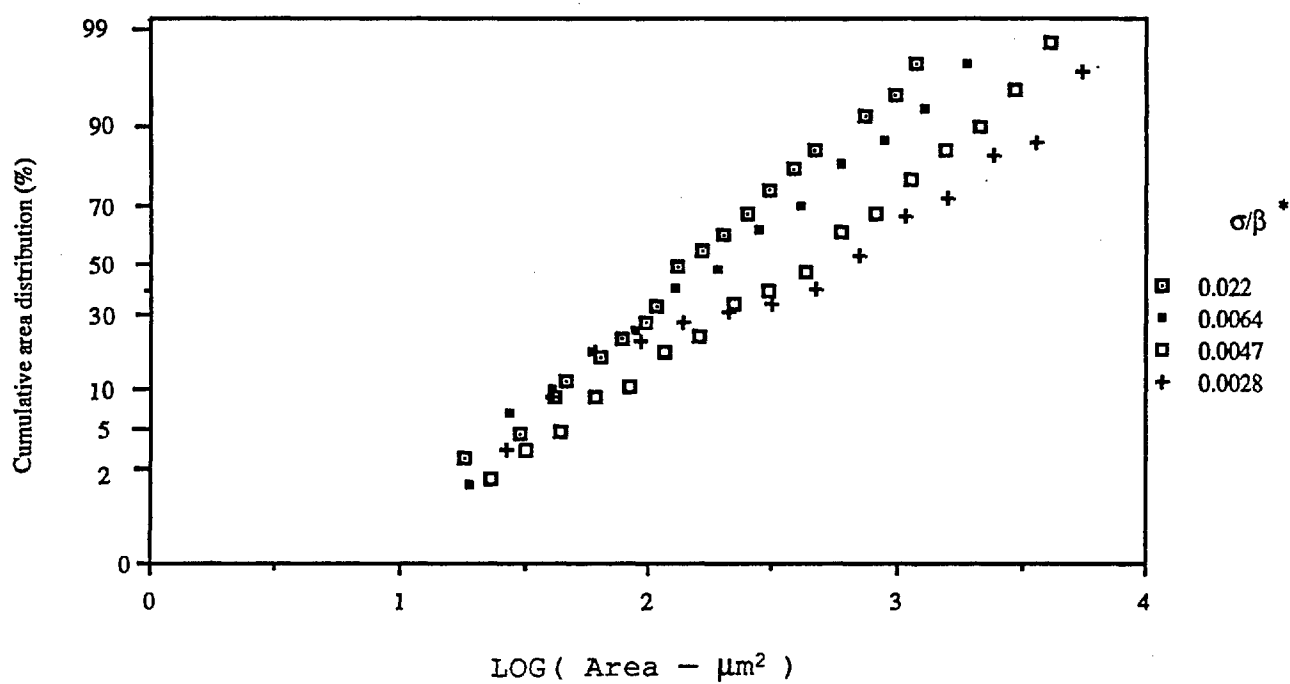


Figure 5.34 Cumulative contact area distribution for various σ/β^* .

inadequacy of the area population as some smaller contacts merge together to form a larger contacts. When the population is high enough, the distributions may tend to a log-normal form.

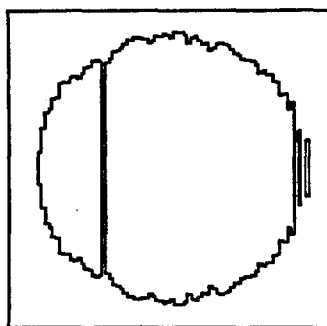
Figure 5.35 shows the contact area of surfaces I01, I03, I04, I06, I08 and I10. Some smaller contacts form a larger contact as the surface roughness decrease. Figure 5.36 shows the variation of the number of contact spots with σ/β^* . As expected, at small σ/β^* , the number of contact spots becomes smaller as a result of the merged contacts. As σ/β^* increases, the number of contact spots gradually increases until at large σ/β^* , appears to approach a constant values.

The mean contact area for each distribution is calculated and shown in Figure 5.37. The mean area changes very little over the range of σ/β^* used, but increases slightly for small σ/β^* . This result is in keeping with Greenwood (1967) theory of how electrical contact resistance and friction theories can be explained in terms of a constant spot size.

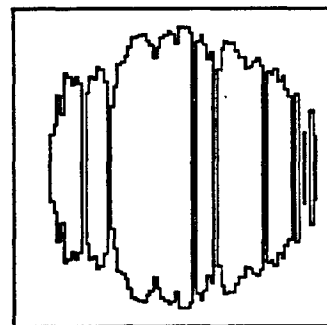
Figure 5.38 to Figure 5.45 show the distribution of spacings for different σ/β^* . Unlike the distribution of contact area, the mode of all the distribution tends to the left, indicating a larger population of smaller spacings between contacts.

Figure 5.46 shows the cumulative spacings distribution for different values of σ/β^* . The graph is again scaled such that a straight line is produced when the distribution is log-normal. A straight line is not evident for all σ/β^* .

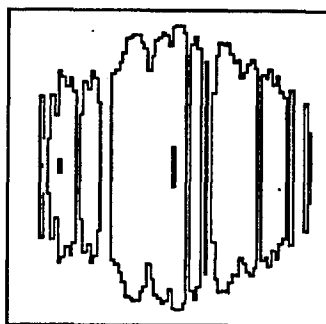
Figure 5.47 shows the variation of the number of spacings with σ/β^* . At small σ/β^* , the number of spacings becomes smaller due to conformity. As σ/β^* increases, the number of spacings gradually increases until at large σ/β^* , it appears to approach a constant value.



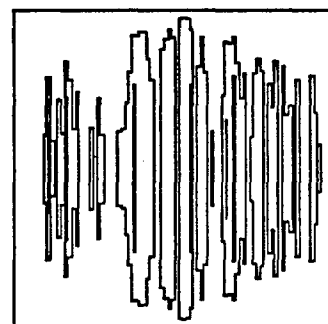
Surface I01



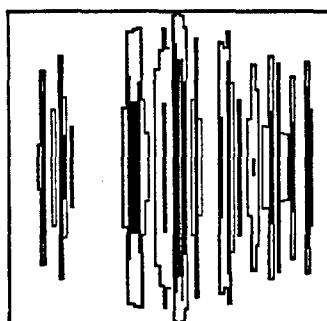
Surface I03



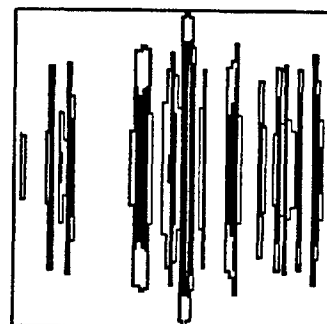
Surface I04



Surface I06



Surface I08



Surface I10

Figure 5.35 Effect of decreasing surface roughness on the size of individual contact.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

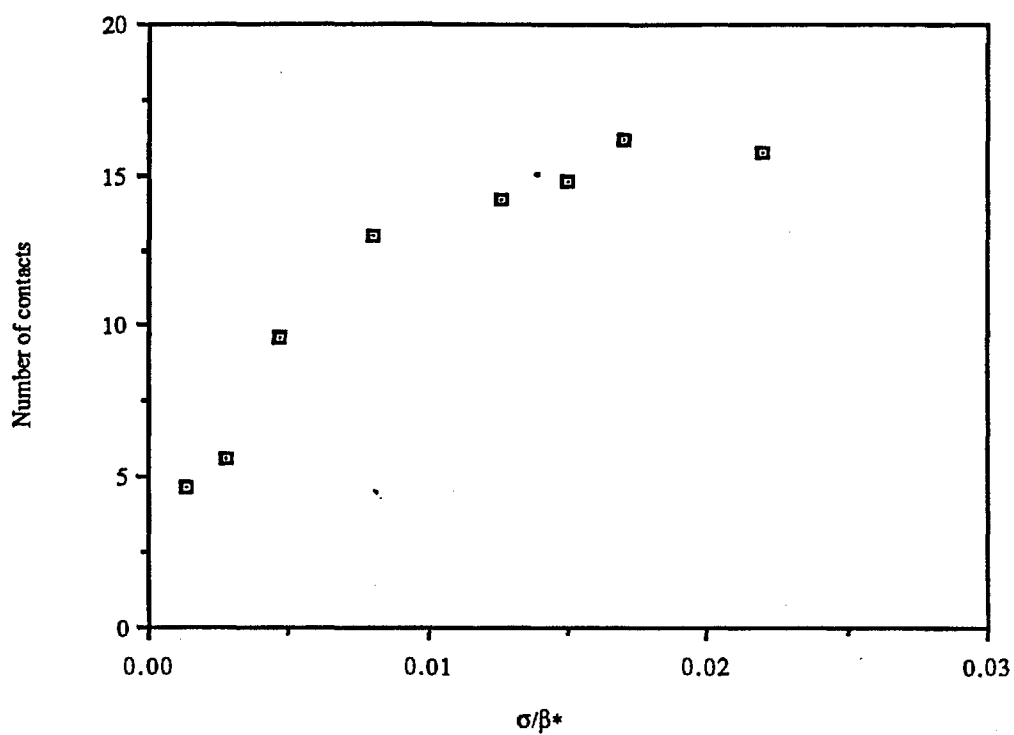


Figure 5.36 Effect of σ/β^* on the number of individual contact.

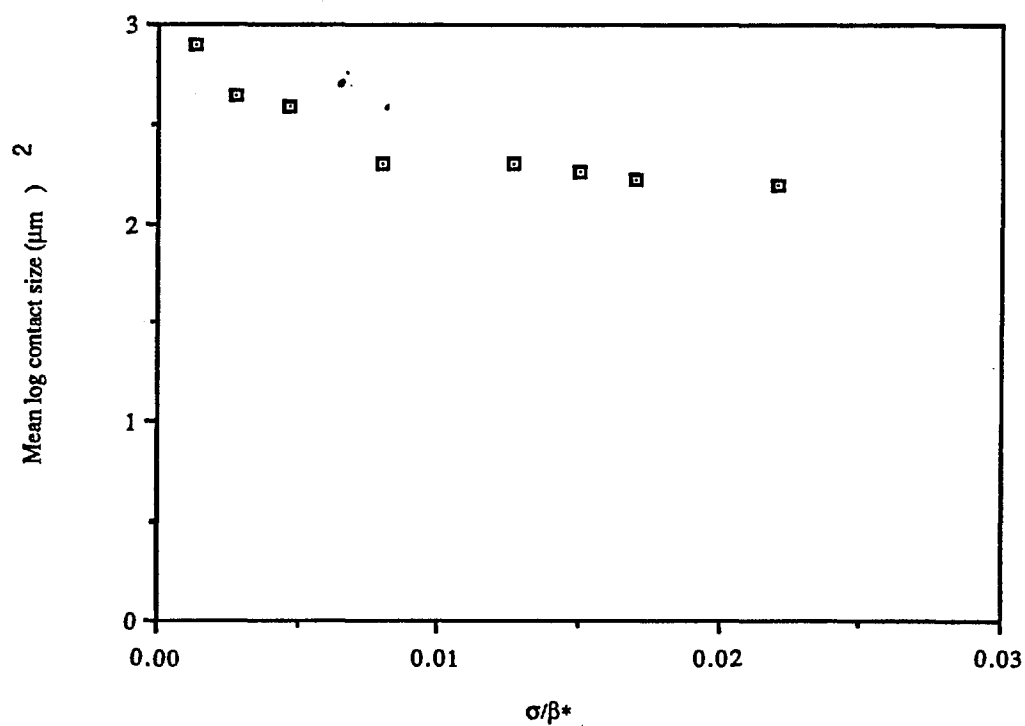


Figure 5.37 Variation of mean contact area with σ/β^* .

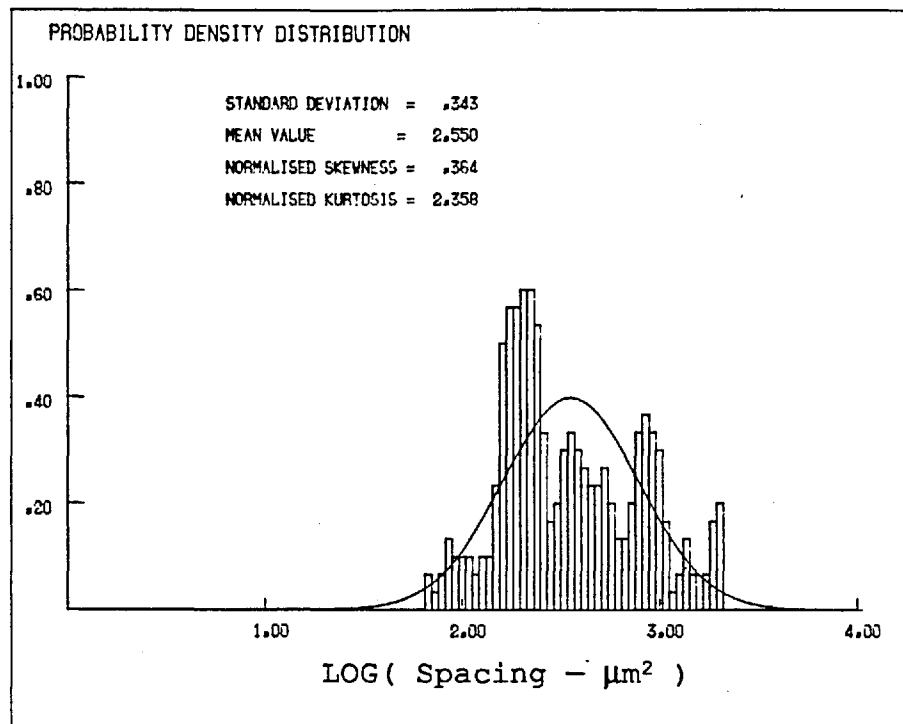


Figure 5.38 Spacing distribution for $\sigma/\beta^*=0.022$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

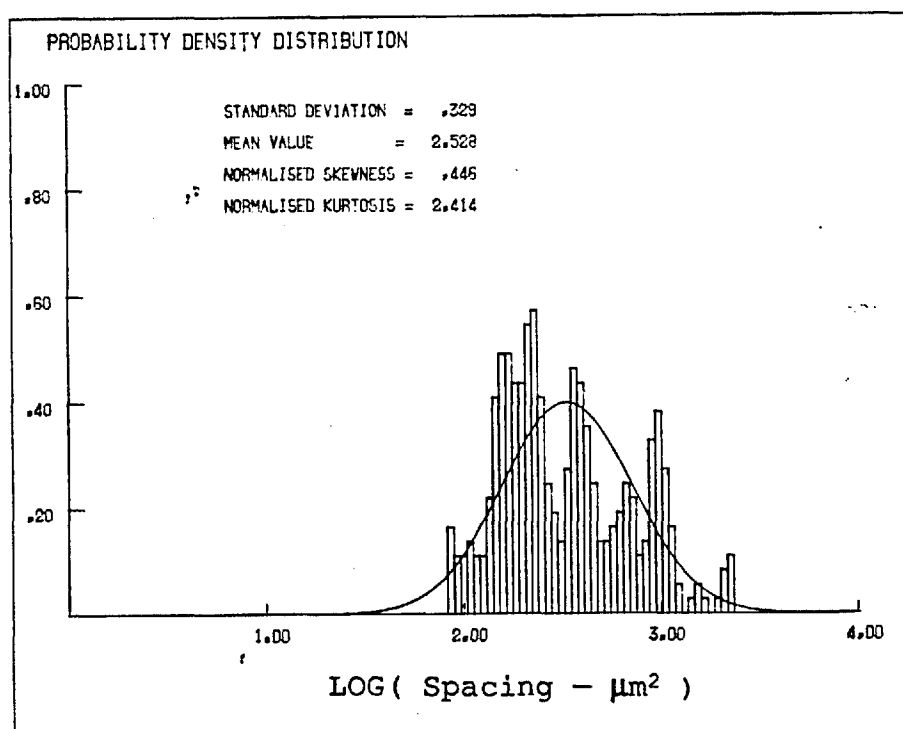


Figure 5.39 Spacing distribution for $\sigma/\beta^*=0.017$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

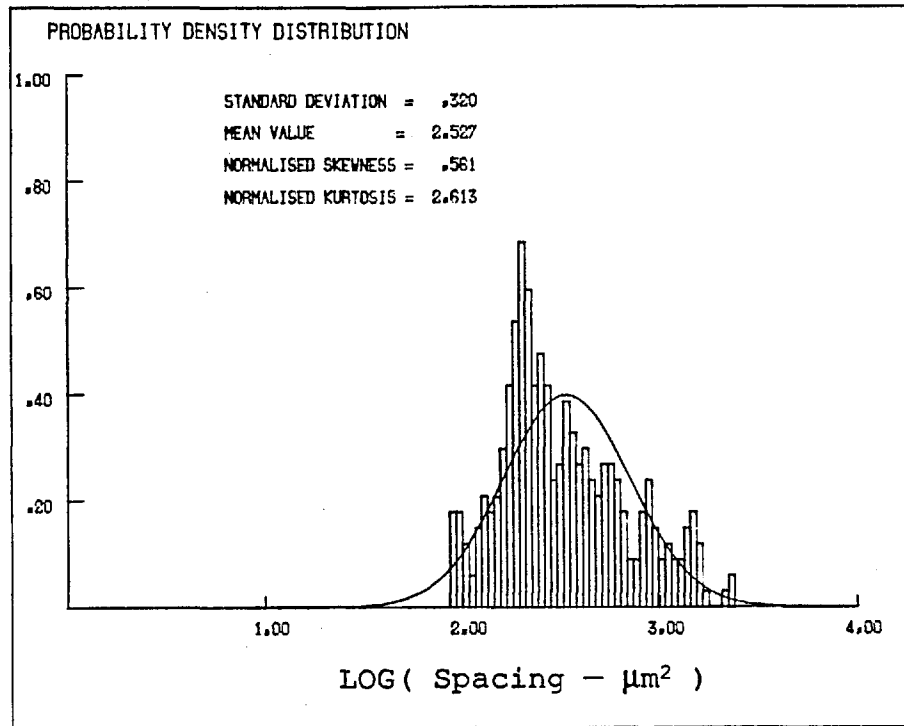


Figure 5.40 Spacing distribution for $\sigma/\beta^*=0.015$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

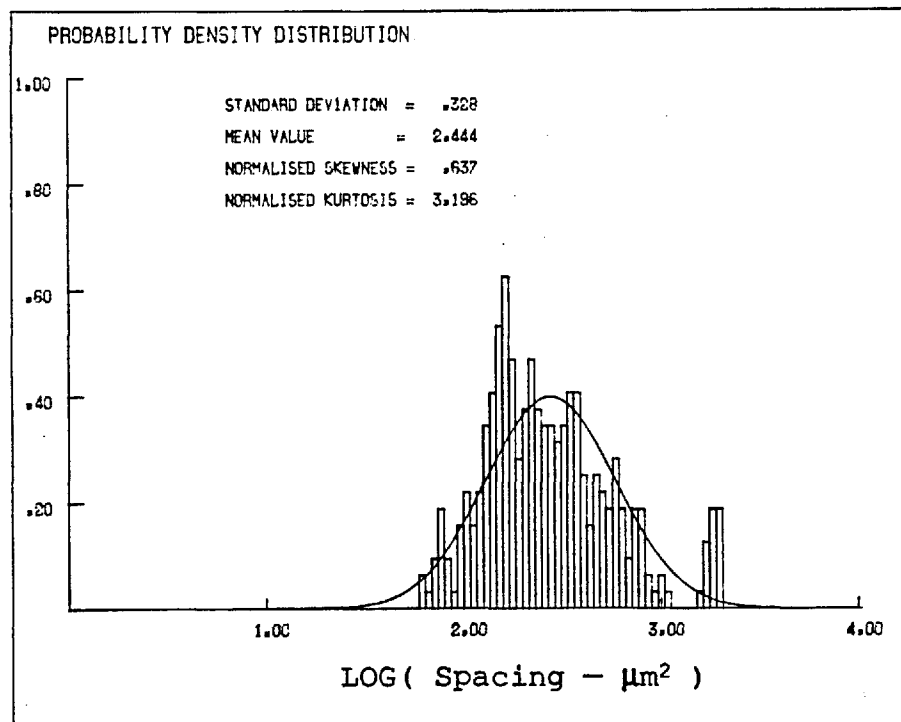


Figure 5.41 Spacing distribution for $\sigma/\beta^*=0.0126$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

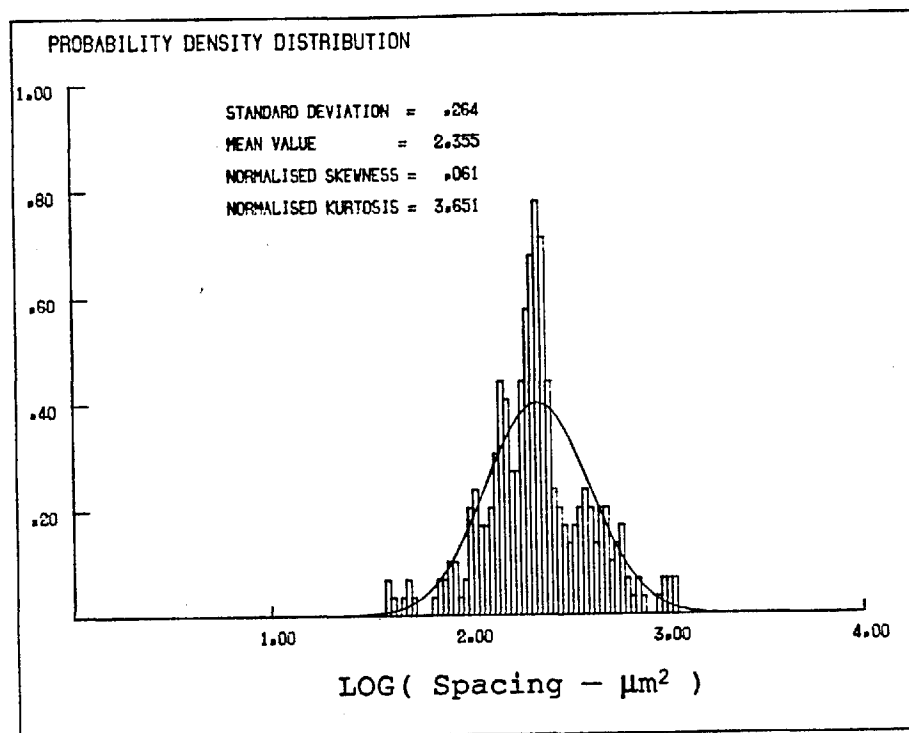


Figure 5.42 Spacing distribution for $\sigma/\beta^*=0.0064$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

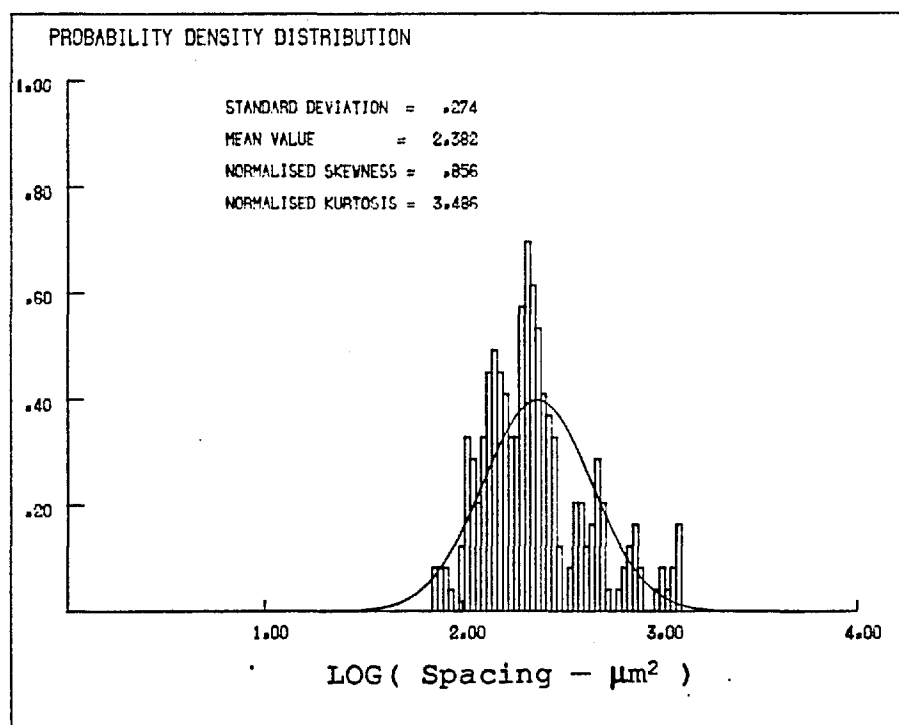


Figure 5.43 Spacing distribution for $\sigma/\beta^*=0.0047$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

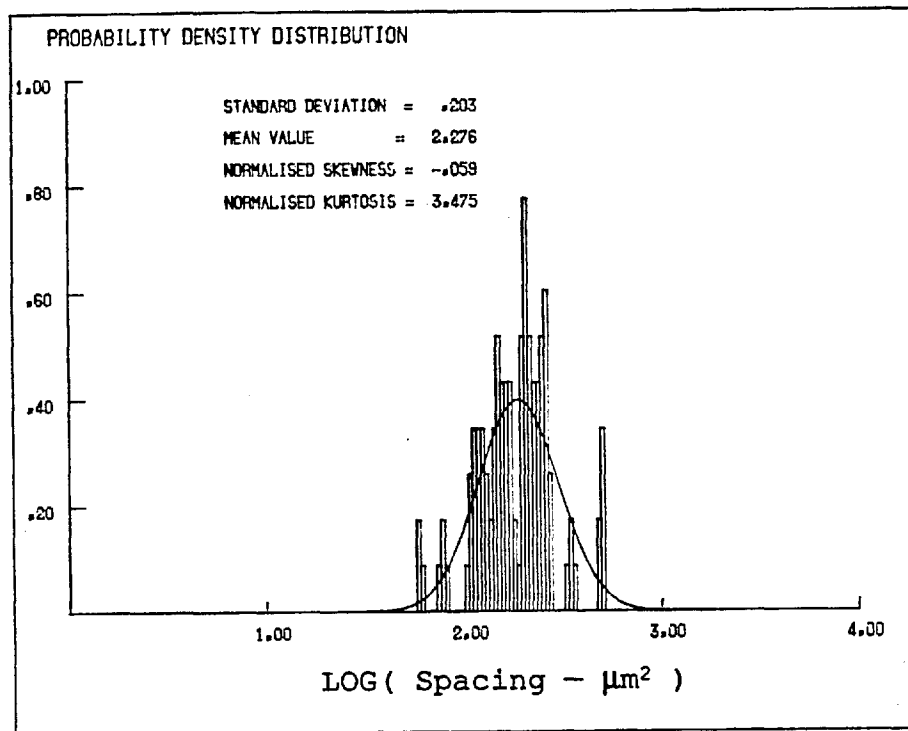


Figure 5.44 Spacing distribution for $\sigma/\beta^*=0.0028$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

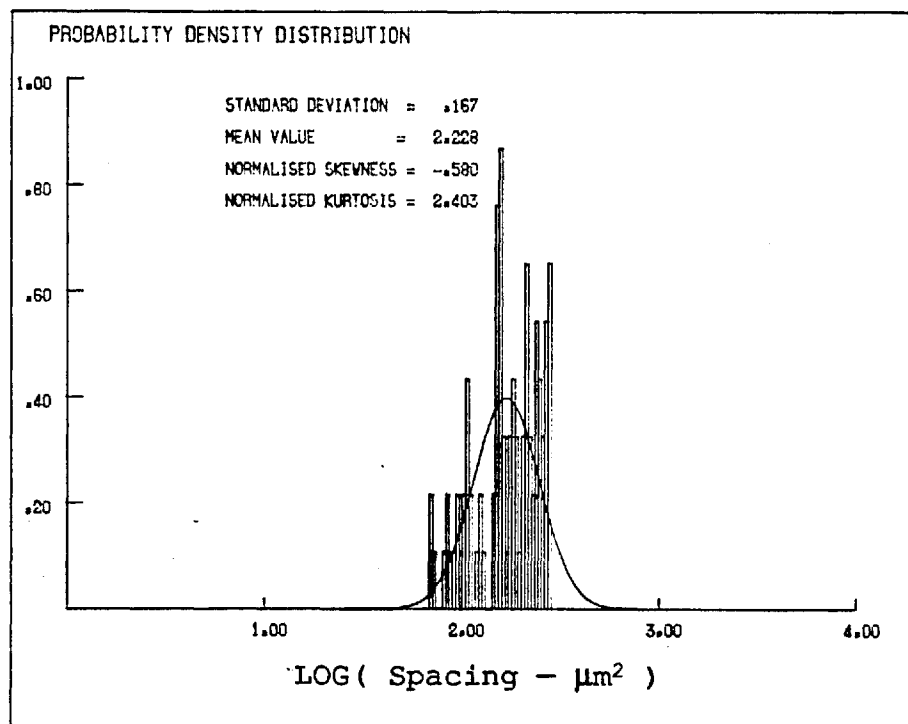


Figure 5.45 Spacing distribution for $\sigma/\beta^*=0.0013$.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

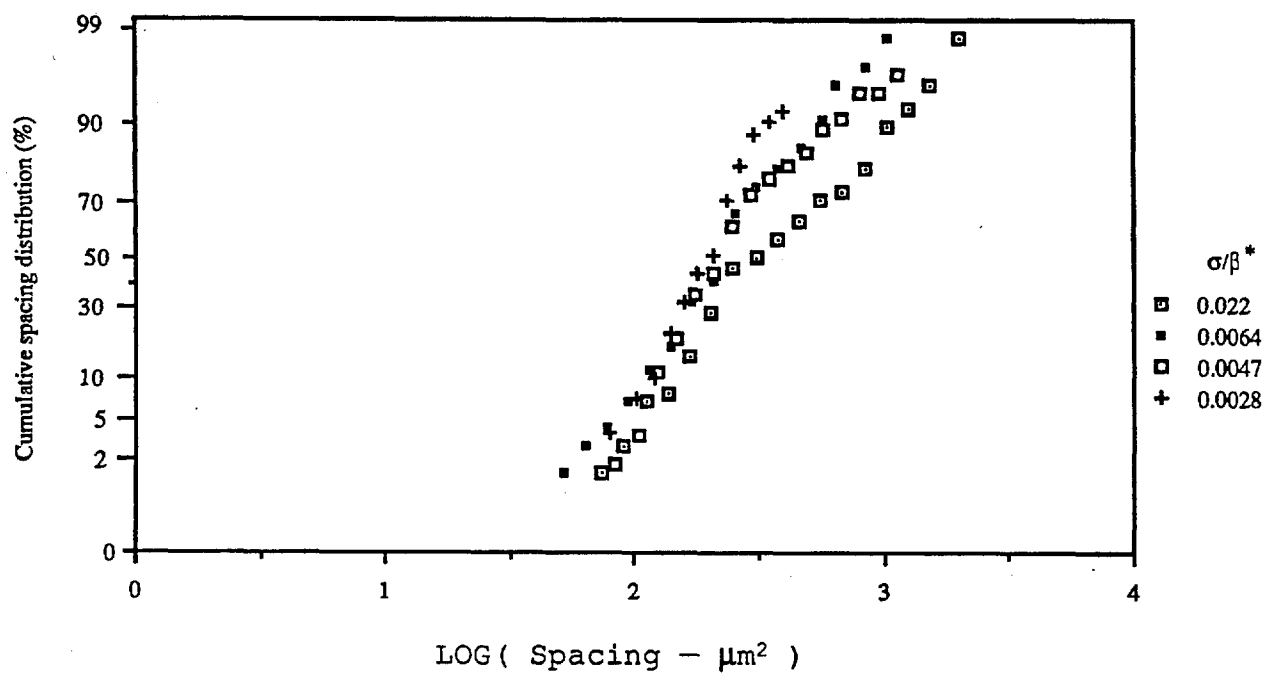


Figure 5.46 Cumulative spacing distribution.

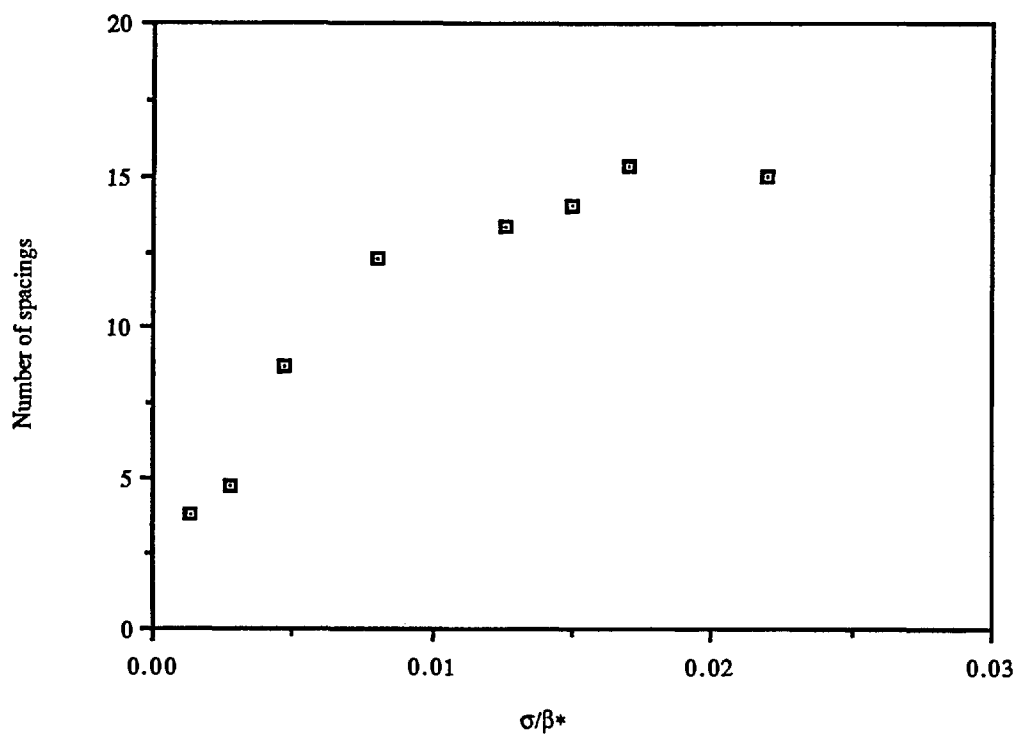


Figure 5.47 Effect of σ/β^* on the number of spacings.
 $p_n=0.56$ Gpa, $H=160$ Hv, $E_1=E_2=207$ Gpa, $\nu_1=\nu_2=0.3$

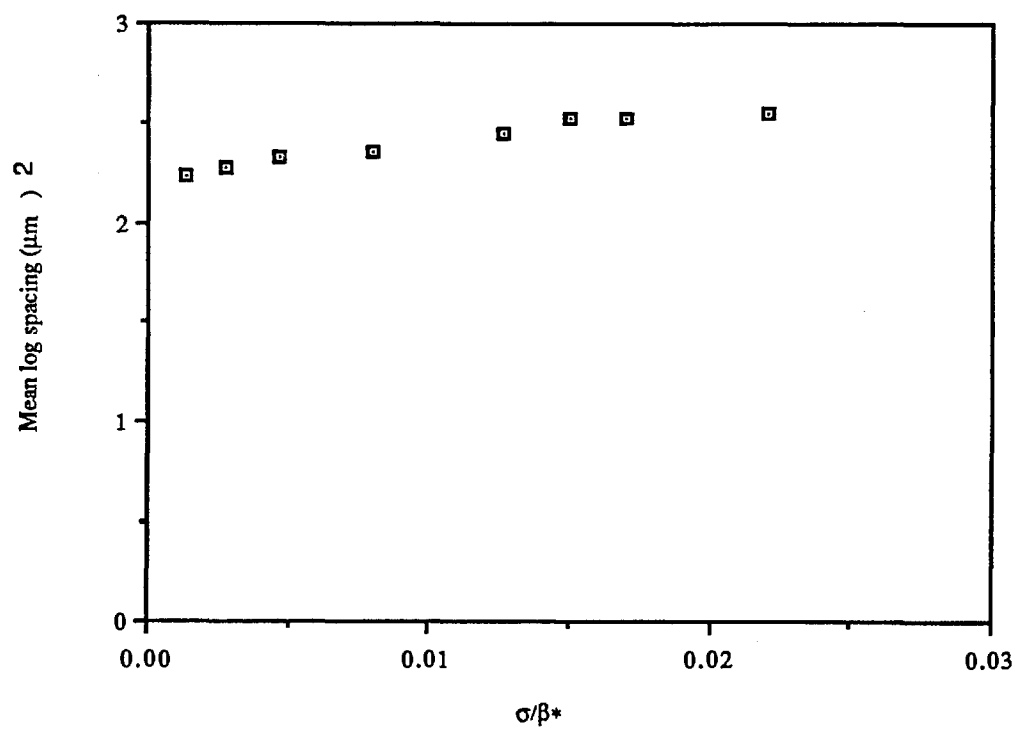


Figure 5.48 Variation of mean spacing area with σ/β^* .

For each distribution curve, the mean spacing is calculated and shown as a function of σ/β^* in Figure 5.48. The results show that the mean spacing increases slightly with σ/β^* .

5.4.6 Density of Contacts

In this section, the density of contacts is examined for different σ/β^* . The contact analysis results for the surfaces described in section 5.3.4. are used. The density of contacts D_c is defined as the number of contact elements divided by the total number of elements. In the case of $p_n=0.56$ Gpa, the nominal width is 150 μm . Thus, the total number of elements of interest, with each contact element of 2 μm width, is 75. Figure 5.49 (curve a) shows the variation of D_c with σ/β^* . The results show that the density of contacts decreases as σ/β^* increases. As σ/β^* becomes greater, the rate of decrease in D_c is smaller.

It is generally believed that contacts take place on the asperity tips and as a consequence wear and plastic deformation is confined to asperity peaks and the valleys remain untouched. For rough surfaces, the statement is reasonably acceptable. However, for very smooth surfaces, contacts can occur in the valleys due to significant amounts of elastic conformity. It is instructive to know among the contacts how many are peak, valley or neither peak nor valley, as surface roughness varies. For convenience, a point which is neither a peak nor a valley is called a shoulder. The proportion of contacts occurring on peaks, valleys and shoulders is also shown in Figure 5.49. The proportion of peaks among contacts increases with σ/β^* . At large σ/β^* , the contacts mainly occur on peaks, and the proportion of valleys in contacts varies only slightly. But as σ/β^* becomes smaller than 0.008, there is a definite increase in the proportion of valleys in contact. The proportion of contacts on shoulder does not change appreciably for $\sigma/\beta^* < 0.012$. But as $\sigma/\beta^* > 0.012$, it

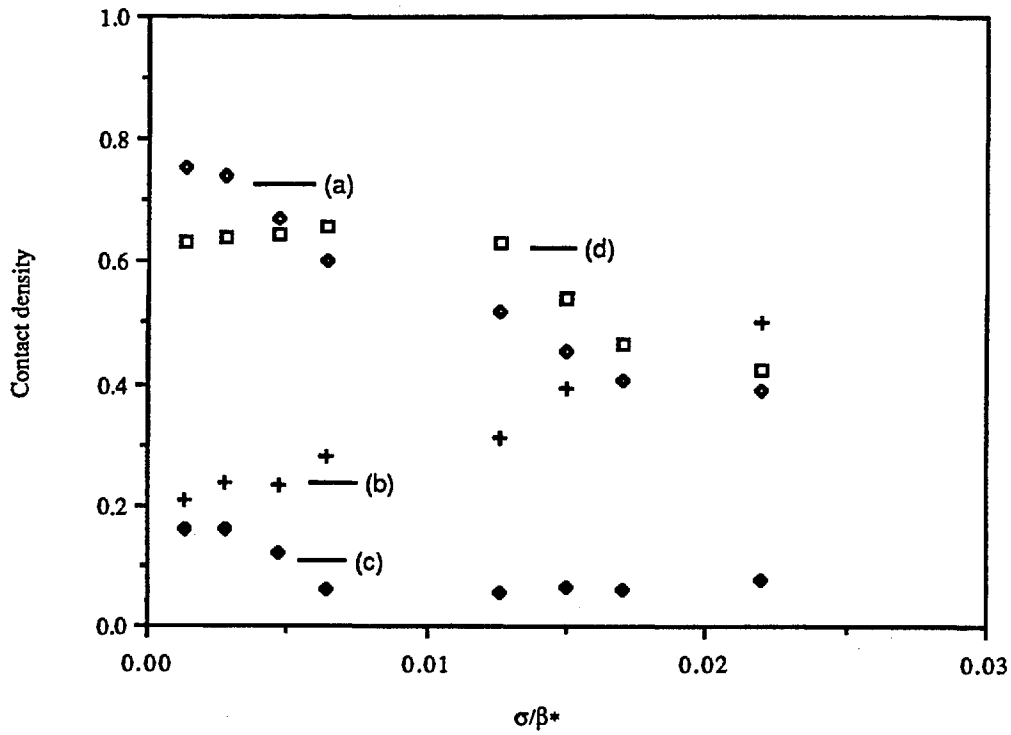


Figure 5.49 Variation of contact density with σ/β^* .
 (a) Number of contacts/Total number of contact element
 (b) Number of peak in contact/Number of contacts
 (c) Number of valley in contact/Number of contacts
 (d) Number of shoulders in contact/Number of contacts.

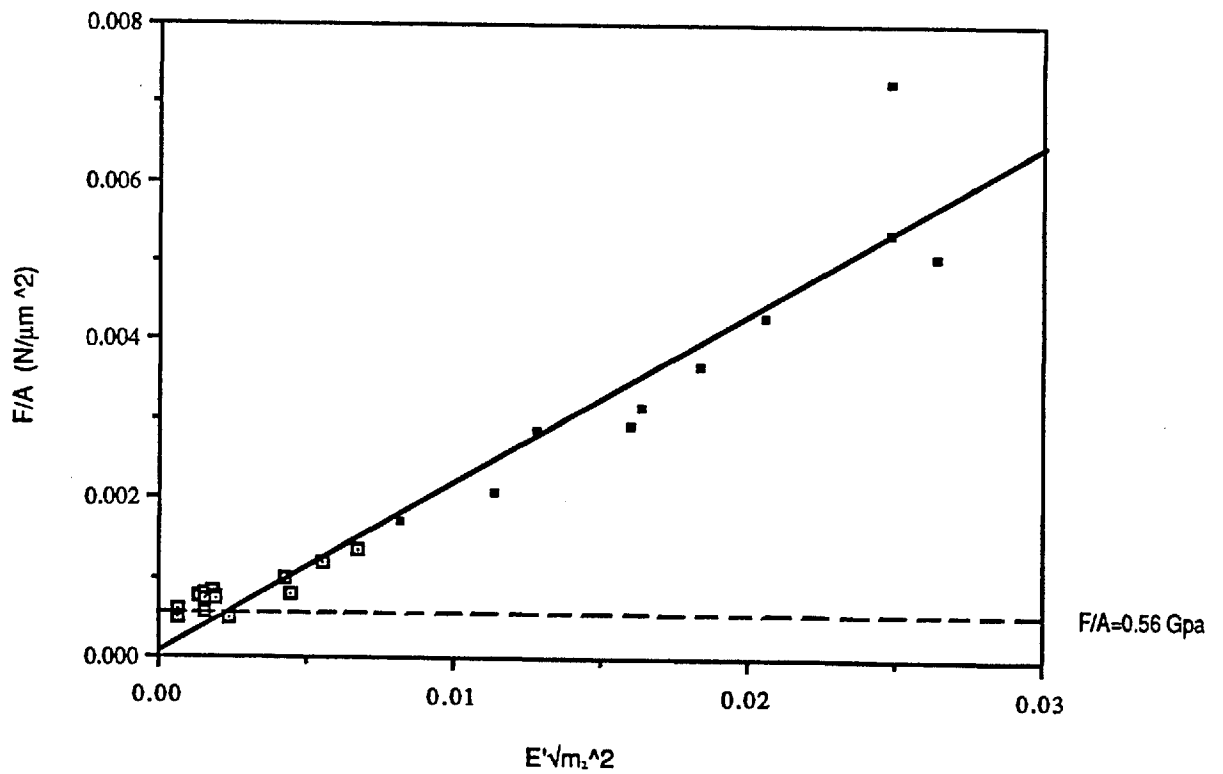


Figure 5.50 Variation of load/area results with $E'\sqrt{m_2}$.

- Rough surface contact results
- West and Webster results
- From Bush's theory.

also decreases with σ/β^* .

5.5 Comparison of Numerical and Stochastic Results

Traditionally, load-area relationships have been derived by a stochastic approach. Bush et al (1975) have studied the contact of strongly anisotropic rough surfaces and therefore provide a suitable model for comparison. Onions and Archard (1973) have provided a model for isotropic surfaces. Although exact agreement would not be expected, it is of interest to know how comparisons behave.

Bush et al modelled the surface as parabolic ellipsoids having a Gaussian distribution of heights. They derived the following expressions for the relationship between the normal load W and the contact area A ,

$$\frac{W}{A} = \frac{\Omega \pi (\bar{\lambda}^2 + \bar{k}'^2)^{\frac{1}{2}}}{2^{\frac{3}{2}} \bar{k}' \bar{k} (\bar{k})} \quad (5.1)$$

where

$$\bar{k}' = \frac{0.4777 \bar{\lambda}}{1 - 1.321 \bar{\lambda}} \quad (5.2)$$

$$\Omega = \sqrt{\frac{m_{02}}{\pi}} E' \quad (5.3)$$

$$\bar{\lambda} = \left(\frac{m_{20}}{m_{02}} \right)^{\frac{1}{2}} \quad (5.4)$$

$$\bar{k} = (1 - \bar{k}'^2)^{\frac{1}{2}} \quad (5.5)$$

$$E' = \left[\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right]^{-1} \quad (5.6)$$

m_{02} and m_{20} are spectral moments in orthogonal directions, λ is the anisotropic bandwidth parameter and $K(\bar{k})$ is the complete elliptical integral of the first kind.

In order to apply the Bush model to compare with the numerical model, the value of λ is chosen 0.01, which corresponds to a very high degree of anisotropy. The elliptical integral is evaluated using tables (Dwight, 1966) for $\lambda=0.01$, thus the theory of Bush can be stated as $W/A=0.214E'\sqrt{m_2}$

Onions and Archard modelled the surface as possessing a Gaussian height distribution with an exponential autocorrelation function, and derived an expression for isotropic surfaces, i.e. $W/A=0.3E'\sigma/\beta^*$.

The numerical results are summarised in Table 5.2. Figure 5.50 shows the load/contact area results from the numerical contact model plotted against $E'\sqrt{m_2}$. West et al (1987a), using numerical analysis of a 2-dimensional contact of a cylinder on a nominally rough surface, have shown that their results agree well with Bush's theory. Their results are also shown for comparison in Figure 5.50. The straight line gives the result from Bush's analysis. The agreement is good for surfaces with $E'\sqrt{m_2}>0.002$. However, for surfaces with small $E'\sqrt{m_2}$, the results seem to deviate. The deviation can be explained by conformity and the effect of concentrated contacts. For very smooth surface, it has been shown that the contact area approaches the nominal area which is given by,

Surfaces	σ (μm)	β^* (μm)	m_2	F/A ($\text{N}/\mu\text{m}^2$)
I01	0.005	4.60	4.55E-6	4.85E-4
J06	0.079	4.73	1.43E-4	7.09E-4
K06	0.065	7.80	3.03E-4	7.36E-4
A06	0.041	18.4	2.96E-5	5.00E-4
H03	0.156	17.8	1.86E-4	8.00E-4
B10	0.114	26.8	1.97E-4	5.66E-4
C02	0.015	3.40	3.64E-5	5.83E-4
C10	0.190	6.25	3.67E-3	13.5E-4
D10	0.180	7.54	2.51E-3	11.9E-4
E05	0.046	5.80	1.51E-4	7.54E-4
E06	0.065	6.20	2.74E-4	8.30E-4
F08	0.068	6.12	2.82E-4	7.71E-4
G08	0.046	4.72	3.01E-4	7.56E-4

Table 5.2 Numerical results for load/area analysis

$$A_n = \pi \left(\frac{3WR}{4E^*} \right)^{\frac{2}{3}} \quad (5.7)$$

where W is the load and R is the radius of the ball. Then the load/area is given by,

$$\frac{W}{A_n} = \frac{4E^*W^{\frac{1}{3}}}{3\pi R} \quad (5.8)$$

For a conforming surface F/A is proportional to $W^{1/3}$ and inversely proportional to the radius R . Thus the results for a ball with a small radius are expected to be higher than results for two normally flat contacts.

The average F/A_n is given by the nominal pressure which is 0.56Gpa. Therefore, the load/area results would be expected to converge to 0.56 Gpa as shown in Figure 5.50 when the surfaces become very smooth. As the surface roughness increases, the contacts are fewer and the load is spread more evenly like that shown in Figure 5.1 and 5.2. In this case the effects of having so called concentrated contacts become less significant and results agree equally well with Bush's theory for nominally flat surfaces, and with the results of West and Webster for 2-D cylindrical contacts.

Figure 5.51 shows the F/A results from the numerical model plotted against σ/β^* . The straight line gives the results from Onions and Archard's theory. Again, as σ/β^* becomes smaller, the load/area value approaches the value given by the nominal pressure.

5.6 Conclusions

A numerical elastic-plastic contact model has been used to analyse a sample set of

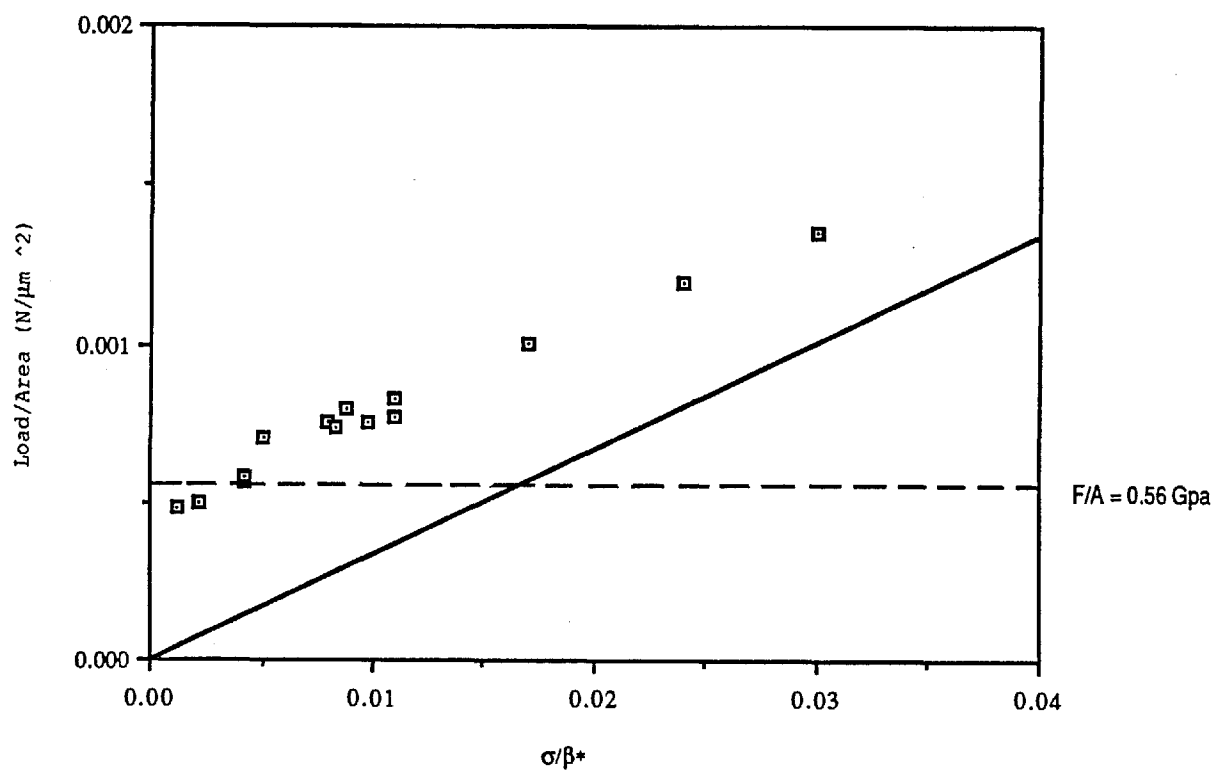


Figure 5.51 Variation of load/area with σ/β^* . The straight line gives the results from Onions and Archard's theory.

rough surface profiles. Results are obtained based on the contact of a steel ball of diameter 12.7 mm on a nominally flat rough surface. The rough surfaces are characterised by two parameters, namely rms roughness σ and correlation length β^* .

The real area and plastic area are found to be dependent on the ratio σ/β^* . For a particular value of σ/β^* , the real and plastic area can be determined by the curve shown in Figure 5.10. In general, there exists different types of contact depending on the value of σ/β^* . For large σ/β^* , the contact is predominantly plastic. The area is less dependent on σ/β^* . For intermediate σ/β^* , the contact is predominantly elastic. As σ/β^* becomes smaller, there exists a finite value of σ/β^* such that the real area is equal to the nominal area.

The load-area relationship for different σ/β^* has been investigated. The hypotheses that the real area of contact is proportional to load has been shown to be approximately true for surfaces with intermediate values of σ/β^* and large values of σ/β^* at relatively low loads. For a surface with a very small value of σ/β^* , the real area approaches the nominal area asymptotically. Results have been compared with Bush's stochastic contact model for highly anisotropic rough surfaces and agreement is good for surfaces with relatively large values of σ/β^* . However, as σ/β^* becomes smaller, the load/area approaches the values given by the nominal pressure for smooth surface contacts.

The distribution of contact pressures, contact areas and spacings have been investigated for various σ/β^* . The distribution of pressures can follow a Gaussian form for intermediate σ/β^* or small σ/β^* while the distributions become skewed towards the flow pressure of the softer material at large σ/β^* .

In general, the distribution of contact areas follow a Gaussian form for intermediate and high values of σ/β^* . The average size and number of the contact spots does not vary appreciably for intermediate and large values of σ/β^* . But at small values of σ/β^* , the number of contact spots is small and their sizes are large.

The mode of distribution of log spacings tends to left for all values of σ/β^* . The average spacing increases slightly with σ/β^* . There is only a very small number of spacings at small σ/β^* . The number of spacings increases with σ/β^* and approaches a constant value at large σ/β^* .

CHAPTER 6

ROUGH SURFACE CONTACT: PLASTICITY AND ELASTIC CONFORMITY

6.1 Introduction

The plasticity and elastic conformity of rough surface contact are important in tribology. Plastic deformation causes rupture of surface asperities or surface films resulting in high friction and surface damage. On the other hand, elastic conformity leads to an increase in contact area and adhesion (Tabor 1977; Johnson et al 1971). The lack of spacings between contacts might reduce the effectiveness of the lubricant in that once a welded junction has formed, it will continue to grow causing severe surface damage.

In the past, much of the attention has been on plasticity in the study of rough surface contact problems. This may be due to the belief that contact takes place on a small fraction of asperity tips which support the load. Then the contact would inevitably be plastic. For a rough surface, this is the case. If the roughness of a surface is reduced, the true area of contact will be increased resulting in the reduction of the contact stress and the contact will be predominantly elastic. For a very smooth surface, the state of contact could reach total elastic conformity such that the true area is equal to the nominal area.

Nowadays, the plasticity index for many engineering surface is low, for example ball bearing raceways where the surface finish can be typically be as low as $R_q=0.05\text{ }\mu\text{m}$. The failure or life performance of such a surface will be much less affected by plasticity and such the surface stresses caused by elastic conformity may dominate the failure mode of the

surface.

Elastic conformity based upon a sinusoidal surface or simple asperity model has been studied by Westergaad (1939), Blok (1952), Johnson (1985) and Sayles et al (1981). They all reach similar conclusions that the conformity is related to two parameters which define the vertical and spatial structure of a surface. For a random surface consisting of a range of asperity shapes and sizes, it is apparent that in contact, the very broad-based asperities might deform elastically under relatively light loading. It is expected that elastic conformity is dependent upon the normal load, elastic modulus and topographic parameters. It would thus be useful to know the limit of wavelength features that will conform related to the surface topography and normal load.

In this chapter, the plasticity and elastic conformity of rough surface contact are studied. Two simple contact models of wavy surfaces, one with a sinusoidal structure and the other with a fixed angle of asperity slope, are presented. They are used to compare the results obtained from numerical contact analysis and investigate the suitability of a wavy surface model to represent a random rough surface. A stochastic approach to the problems of conformity and plasticity is also attempted.

6.2 One Dimensional Sinusoidal Surface

An elastic half-space solution of surface pressures acting on a sinusoidal surface has been analysed by Westergaad(1939). The pressure distribution at $x \leq a$ is given by:

$$p(x) = \frac{2p_m \cos(\frac{\pi x}{\lambda})}{\sin^2(\frac{\pi a}{\lambda})} \left[\sin^2(\frac{\pi a}{\lambda}) - \sin^2(\frac{\pi x}{\lambda}) \right]^{\frac{1}{2}} \quad (6.1)$$

and

$$p_m = \left(\frac{\pi E' \Delta}{\lambda} \right) \sin^2 \left(\frac{\pi a}{\lambda} \right) \quad (6.2)$$

where p_m is the mean pressure, $2a$ is the contact width, Δ is the amplitude and λ is the wavelength of the sinusoidal surface. Equation (6.2) can be inverted to express the ratio of the 'real' to the 'apparent' area of contact and writing $p^* = \pi E' \Delta / \lambda$,

$$\frac{2a}{\lambda} = \frac{2}{\pi} \sin^{-1} \left(\frac{p_m}{p^*} \right)^{\frac{1}{2}} \quad (6.3)$$

For an elastic-plastic solution, the contact solution can be obtained in a similar manner as the numerical contact model described in Chapter 4. For the half space contact solution, the contact pressure in the direction perpendicular to the profile is uniform instead of Hertzian. The repetitive nature of the sinusoidal surface enables the surface elements to be set lying on half the wavelength of the surface. The bearing length, in the direction perpendicular to the profile's plane, can be chosen sufficiently large such that the half-space condition is reached. In the analysis, the bearing length is taken as two thousand times the sample length. The influence coefficients are then set up due to the pressures acting on the surfaces using equation (4.4). It is noted that when surface forces are far away from the contact elements, then the surface deflection at the contacting region will be displaced downwards as a whole. The asperity pressures will not be affected by these forces. Therefore, it would seem sensible to limit the consideration of surface forces to a certain number of wavelengths in order to reduce the computing time.

It was found that the solution of contact pressure reaches 98% accuracy when ten wavelengths of contact pressures on both sides are considered. This is shown in Figure 6.1. The numerical pressures of a purely elastic contact almost coincides with that obtained from the analytical equation (6.1). For the elastic-plastic contact, it can be seen that the

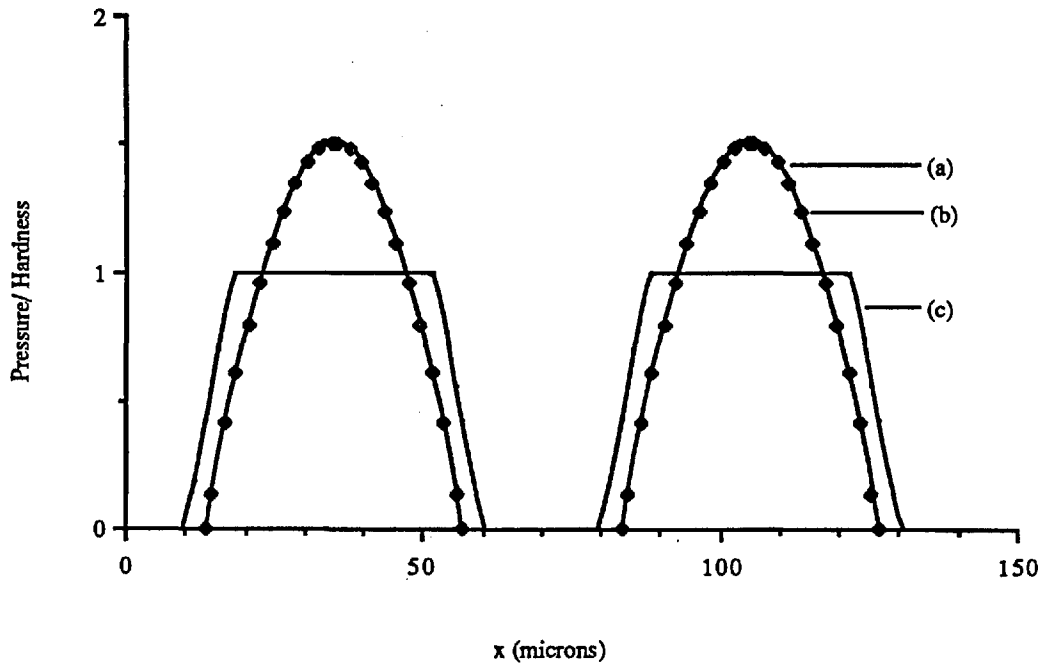


Figure 6.1 Pressure distribution of a one-dimensional sinusoidal surface. $p_n/\pi E' = 0.0028$, $H/\pi E' = 0.00455$, $\Delta = 0.3 \mu\text{m}$, $\lambda = 70 \mu\text{m}$.
 (a) by equation (6.1)
 (b) purely elastic from contact analysis
 (c) elastic-plastic contact from contact analysis

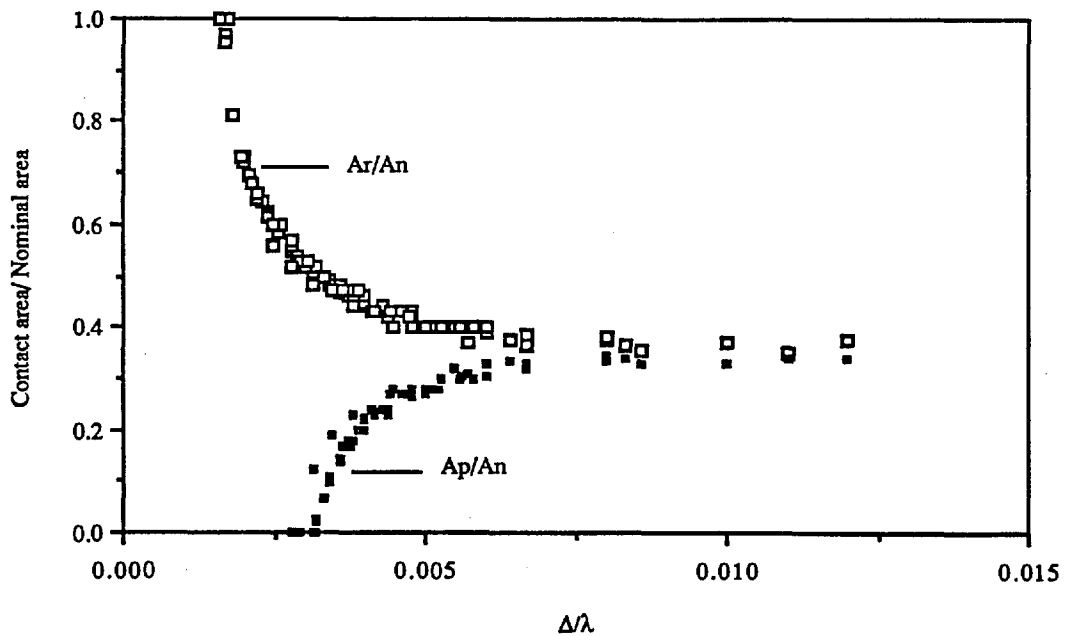


Figure 6.2 Variation of real area and plastic area with Δ/λ . Results represent 60 surfaces with different Δ and λ . $p_n/\pi E' = 0.0016$, $H/\pi E' = 0.00455$.

pressures at the crest have been flattened as expected. The true area of contact is slightly larger than it is in the purely elastic contact.

It has been realised that the real area and the plastic area of contact are dependent on materials modulus, hardness, load and geometry of the contacting bodies. It is instructive to investigate the effect of each parameter on the real area and plastic area of the surfaces in order to obtain a relationship between contact area and material parameters and surface topographies. For convenience, the surface deflections, pressures and hardness are normalised by $\pi E'$, so that the investigation can be applied more generally to various materials.

A set of 60 sinusoidal surfaces with different combinations of Δ and λ were subjected to contact analysis under different loading. The effects of changing Δ , λ and loading and hardness of the material were investigated.

6.2.1 Effect of Δ and λ

Figure 6.2 shows the variation of A_r/A_n and A_p/A_n with Δ/λ for the set of 60 sinusoidal surfaces with different Δ and λ at a load of $p_m/\pi E' = 0.0016$ and a hardness $H/\pi E' = 0.00455$ where A_r , A_p and A_n are the real area, plastic area and nominal area respectively. It can be seen that the real area and plastic area are dependent on the ratio Δ/λ , not the individual value of Δ and λ . This becomes more apparent when the distribution of contact area is plotted on a Δ - λ map as shown in Figure 6.3.

The results in Figure 6.2 show that as Δ/λ increases, the real area or plastic area approaches a value of $A_r/A_n = 0.35$ and this coincides with the value predicted by plastic deformation. i.e.

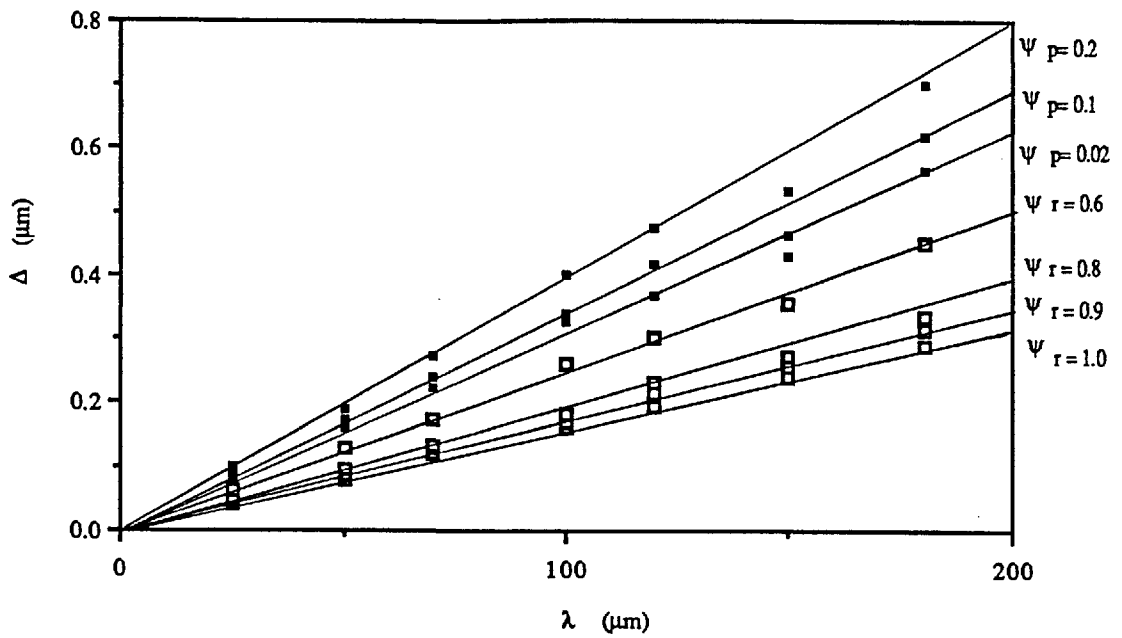


Figure 6.3 Real area and plastic area distribution on a Δ - λ map with lines $\psi_r = A_r/A_n$ —□— and $\psi_p = A_p/A_n$ -□-. $p_n/\pi E' = 0.0016$, $H/\pi E' = 0.00455$.

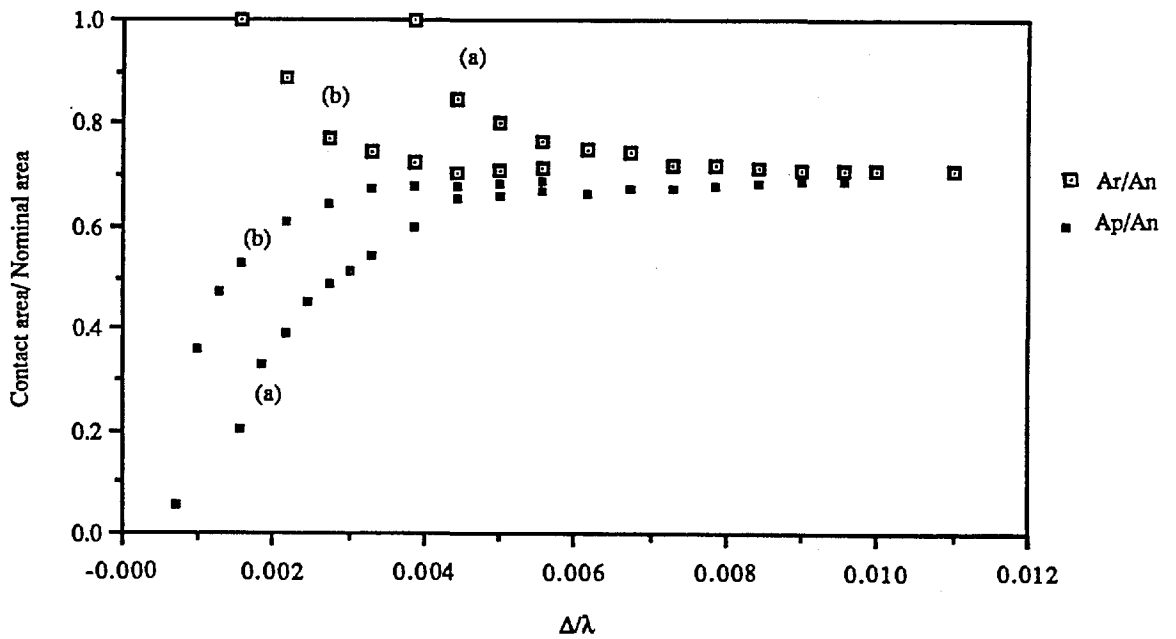


Figure 6.4 Effect of increasing p_n/H by twice on the value of A_r/A_n and A_p/A_n .
 (a) $p_n/\pi E' = 0.0016$, $H/\pi E' = 0.0023$
 (b) $p_n/\pi E' = 0.0032$, $H/\pi E' = 0.00455$

$$\frac{A_p}{A_n} = \frac{p_m}{H} \quad (6.4)$$

It is instructive to know whether the value of A_p/A_n still approaches p_m/H for the other values of p_m and H . Figure 6.4 shows the variation of A_r/A_n and A_p/A_n with Δ/λ by increasing the mean pressure by twice or decreasing the hardness by half such that the ratio of p_m/H is increased by a factor of two. At small values of Δ/λ , where the contact is predominantly elastic, the shape of the curves is dependent upon the value of p_m . At large Δ/λ , both curves appear to approach the ratio of p_m/H .

Therefore, when Δ/λ is large, the real contact area will be that produced by plastic deformation and can be determined by the ratio p_m/H .

The mean pressure is equal to the normal load W divided by nominal area, i.e.

$$p_m = \frac{W}{A_n} \quad (6.5)$$

Putting equation (6.5) in equation (6.4) leads to the well known expression, $A_p = W/H$.

6.2.2 Effect of Load

Figure 6.5 shows the variation of real area and plastic area with the load at a hardness of $H/\pi E' = 0.00455$ for 5 surfaces with different Δ/λ . The curves of A_r/A_n appear to be symmetrical about the diagonal of $A_r/A_n = p_m/H$. With a higher value of hardness, the curves of A_r/A_n and A_p/A_n for a given value of Δ/λ also appear to be symmetrical (not shown). But in this case, the curves of A_r/A_n shifts to the left and those of A_p/A_n shift to

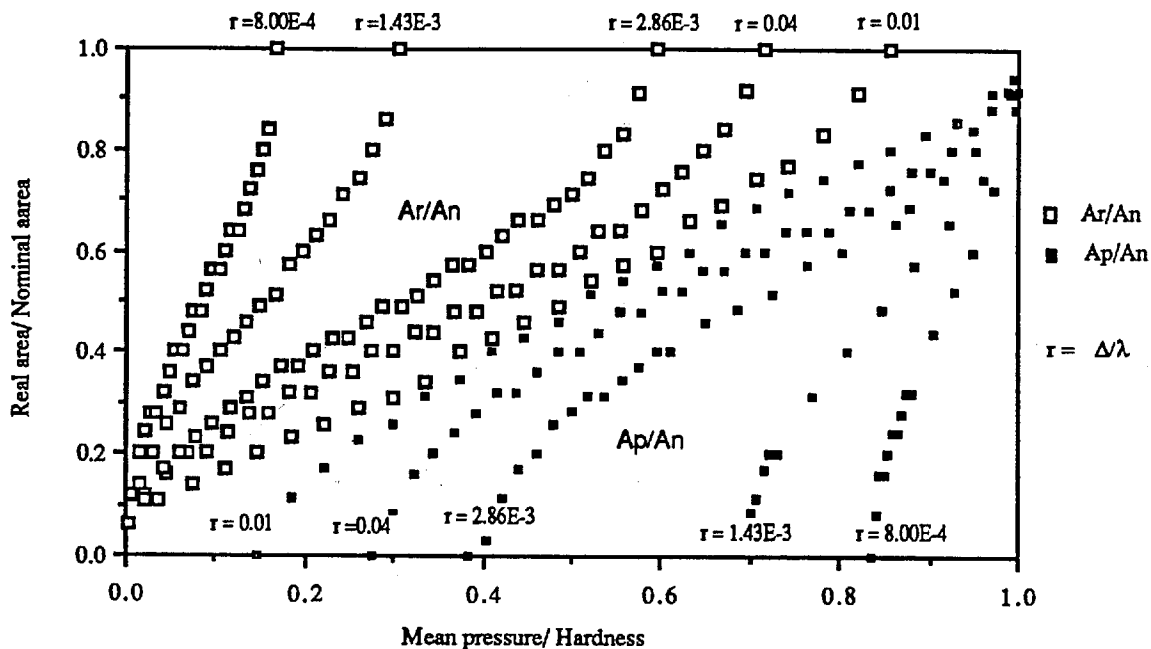


Figure 6.5 Variation of real area and plastic area with load for sinusoidal surfaces. $H/\pi E' = 0.00455$

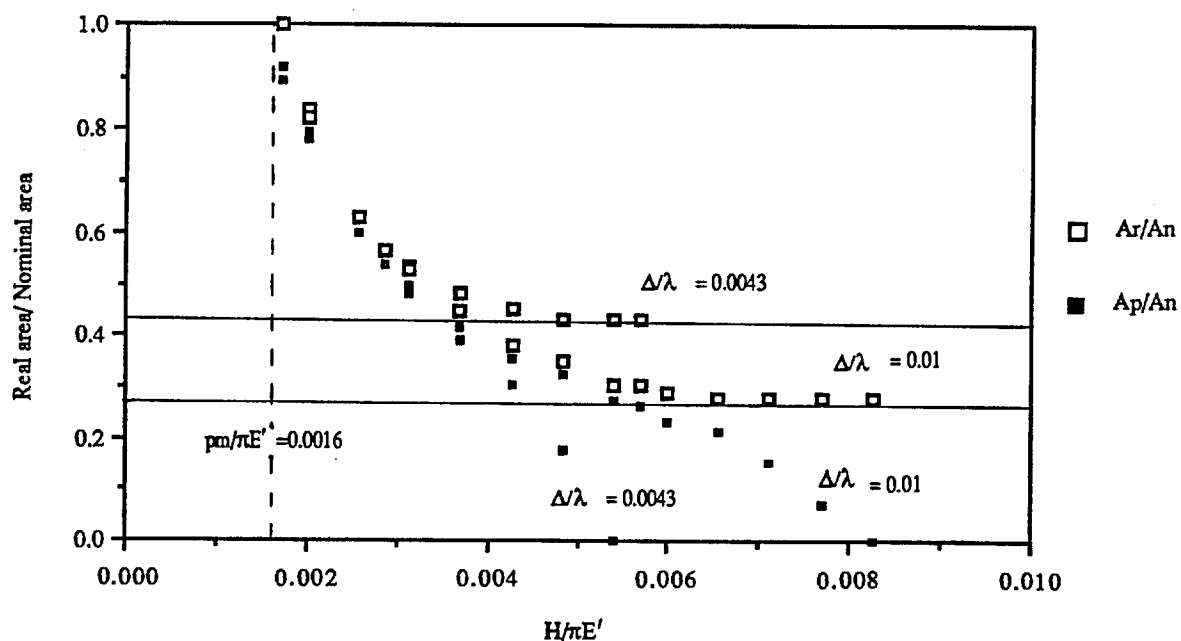


Figure 6.6 Effect of Hardness on real area and plastic area. The lines — represents the real area of purely elastic contact at a constant Δ/λ obtained from equation (6.3)

the right to maintain the symmetry.

For a given load, the rate of change of contact area is greater for a smaller value of Δ/λ . The minimum load required for total conformity increases with Δ/λ . When Δ/λ is large, the contact is predominately plastic and the contact area approaches the diagonal of $A_p/A_n = p_m/H$. In other words, there exists a value of Δ/λ , in this case $\Delta/\lambda \approx 0.01$, after which the contact area is not dependent on Δ/λ and can be determined by the relationship $A_p/A_n = p_m/H$.

6.2.3 Effect of Hardness

Here the problem of asperity persistence is not considered, if the loading is increased to a value equal to H . Then the plastic area is equal to the nominal area. The condition for the plastic area to be smaller than the nominal area is $H > p_m$. This is consistent with the results obtained from contact analysis as shown in Figure 6.6. As H approaches p_m , then A_r/A_n or A_p/A_n is unity regardless of Δ/λ . As H increases, the plastic area decreases and eventually the contact will be predominantly elastic. Then the true area is independent of H and equal to the value given by equation (6.3).

6.3 One Dimensional Triangular Surface

The approach to the problem was similar to that of the sinusoidal surface. A surface profile with a given angle of inclination was created as an input profile instead of a sinusoidal profile. The pressure of a purely elastic contact at the tip is theoretically infinite. This problem was overcome by choosing the surface element sufficiently close to the tip such that the load under the uniform pressure within a sample length was sufficiently close

to that calculated from the purely elastic pressure. In addition, the infinite tip pressure is not physically significant and the tip will undergo plastic deformation when the pressure is higher than the flow pressure of the material. Figure 6.7 shows a typical contact pressure distribution on the surface, for the purely elastic and elastic-plastic cases.

6.3.1 Effect of Δ and λ

Figure 6.8 shows the real area and plastic area against Δ/λ and Figure 6.9 shows the distribution of real area and plastic area on a Δ - λ map at a load of $p_m/\pi E' = 0.0016$ and a hardness $H/\pi E' = 0.00455$. Again they are dependent on the ratio Δ/λ and not the individual value of Δ or λ . At large Δ/λ , the real area is predominantly plastic and it approaches the ratio p_m/H . By increasing the mean pressure by twice or decreasing the hardness by half, the value to which A_p/A_n approaches will correspondingly increase by twice as shown in Figure 6.10. For large Δ/λ , the plastic area is given by $A_p = W/H$, the same expression as that obtained from the contact analysis of sinusoidal surfaces.

6.3.2 Effect of Load

Figure 6.11 shows the variation of real area and plastic area with load for surfaces with different values of Δ/λ . The variation is similar to that of the sinusoidal surfaces except the shape of the curves has a different form. The curve of A_p/A_n also appears approximately symmetric to the curve of A_r/A_n about the diagonal of $A_p/A_n = p_m/H$.

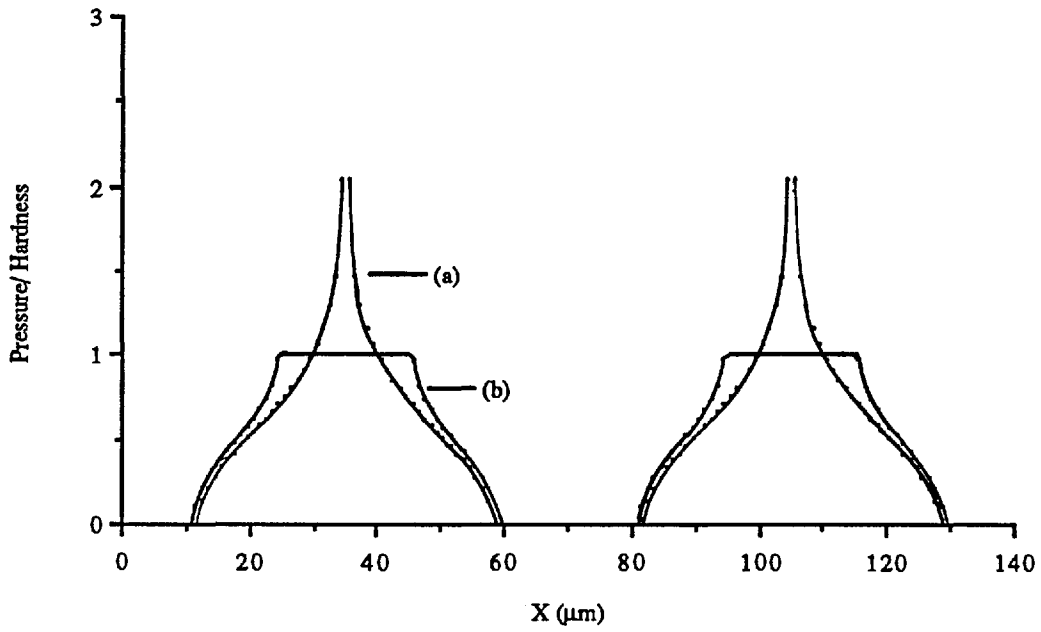


Figure 6.7 Pressure distribution of a one-dimensional triangular surface. $p_n/\pi E' = 0.00224$, $H/\pi E' = 0.00455$, $\Delta = 0.3 \mu\text{m}$, $\lambda = 70 \mu\text{m}$.
 (a) Purely elastic
 (b) Elastic-plastic contact

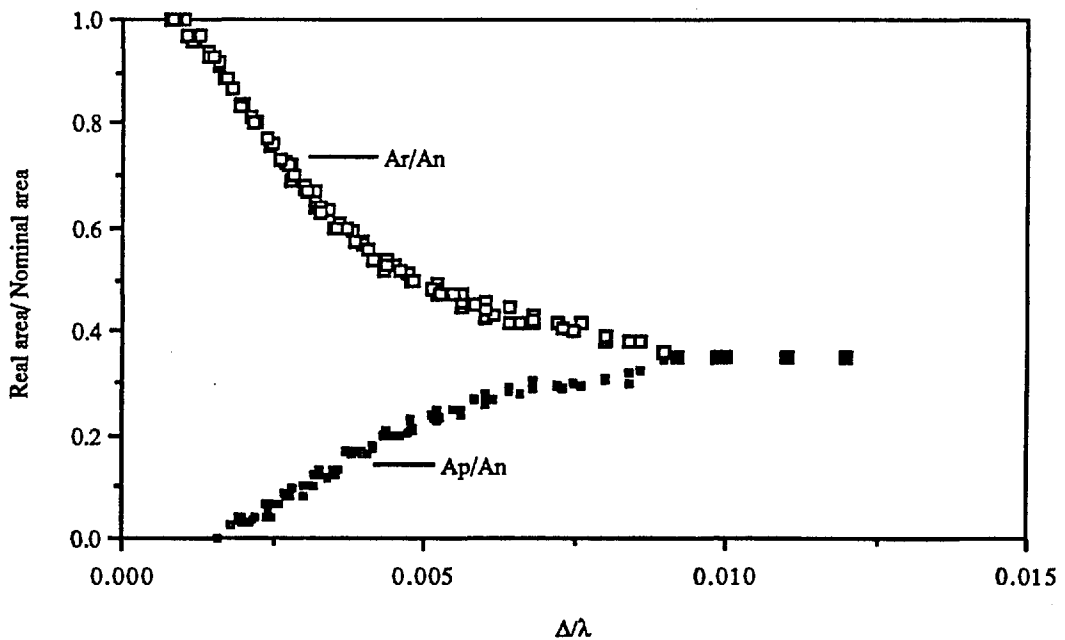


Figure 6.8 Variation of real area and plastic area with Δ/λ . Results represent 60 surfaces with different Δ and λ . $p_n/\pi E' = 0.0016$, $H/\pi E' = 0.00455$.

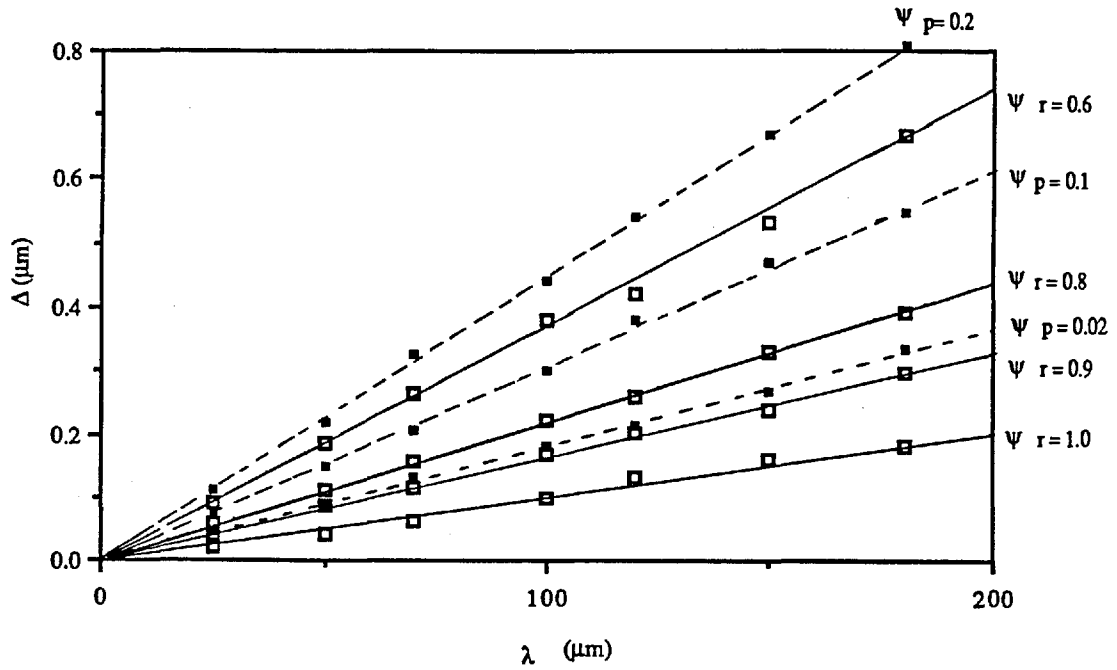


Figure 6.9 Real area and plastic area distribution on a Δ - λ map with lines $\Psi_r = A_r/A_n$ — and $\Psi_p = A_p/A_n$ -----, $p_n/\pi E' = 0.0016$, $H/\pi E' = 0.00455$.

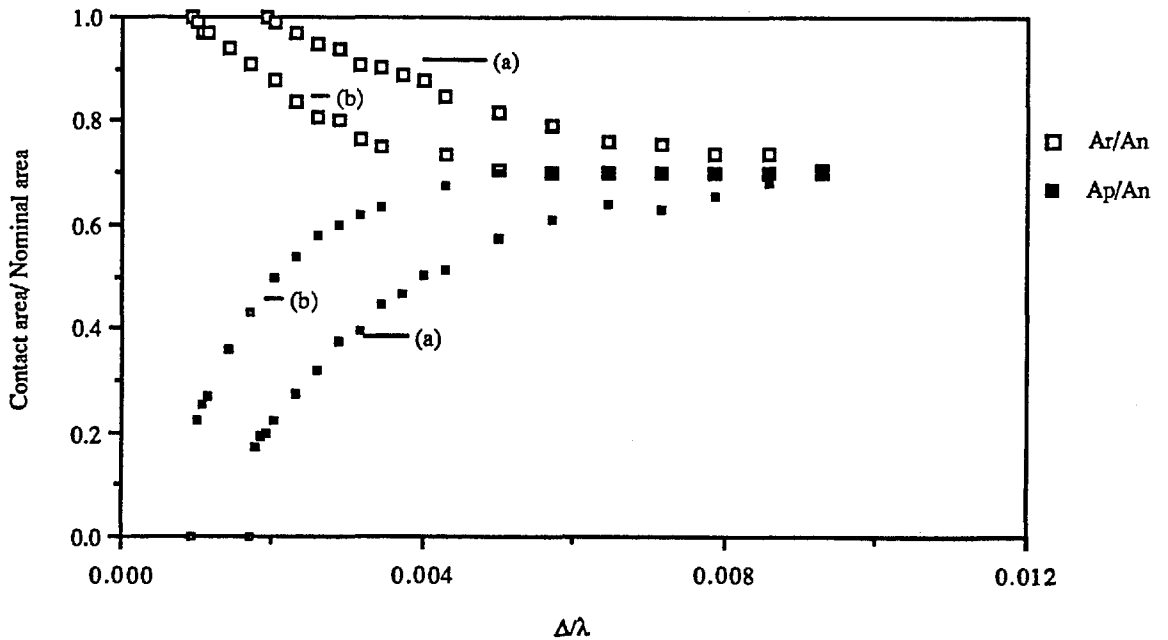


Figure 6.10 Effect of increasing p_n/H by twice on the value of A_r/A_n and A_p/A_n .
(a) $p_n/\pi E' = 0.0016$, $H/\pi E' = 0.0023$
(b) $p_n/\pi E' = 0.0032$, $H/\pi E' = 0.00455$

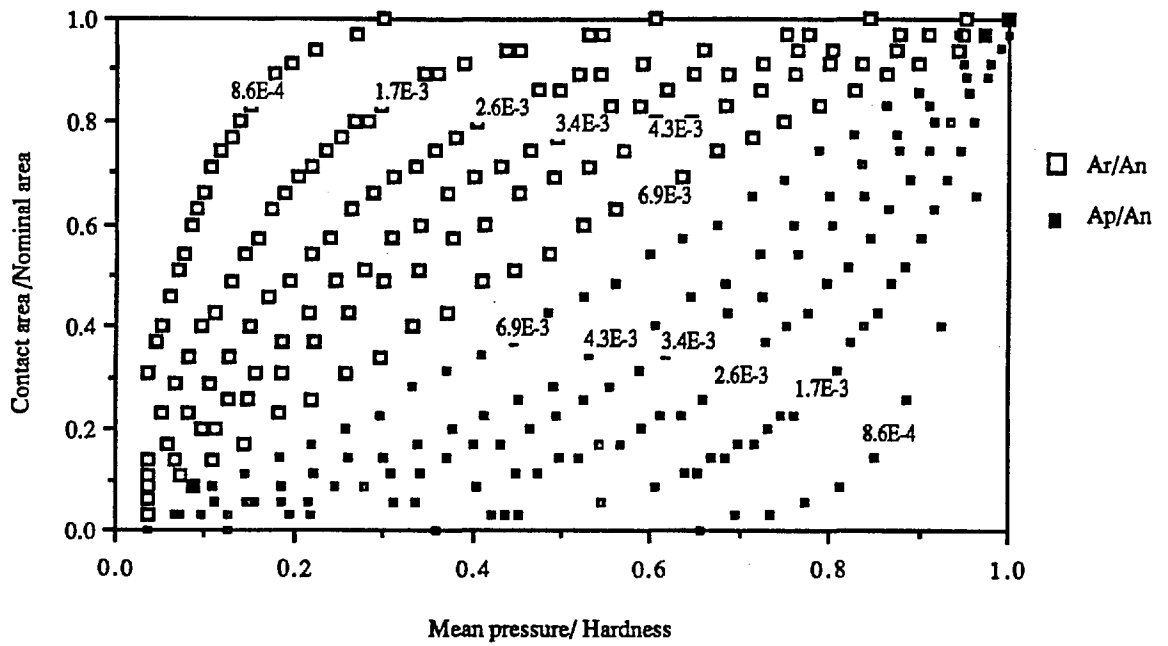


Figure 6.11 Variation of real area and plastic area with load. The value on the curve is the ratio Δ/λ of a triangular surface. $H/\pi E' = 0.00455$.

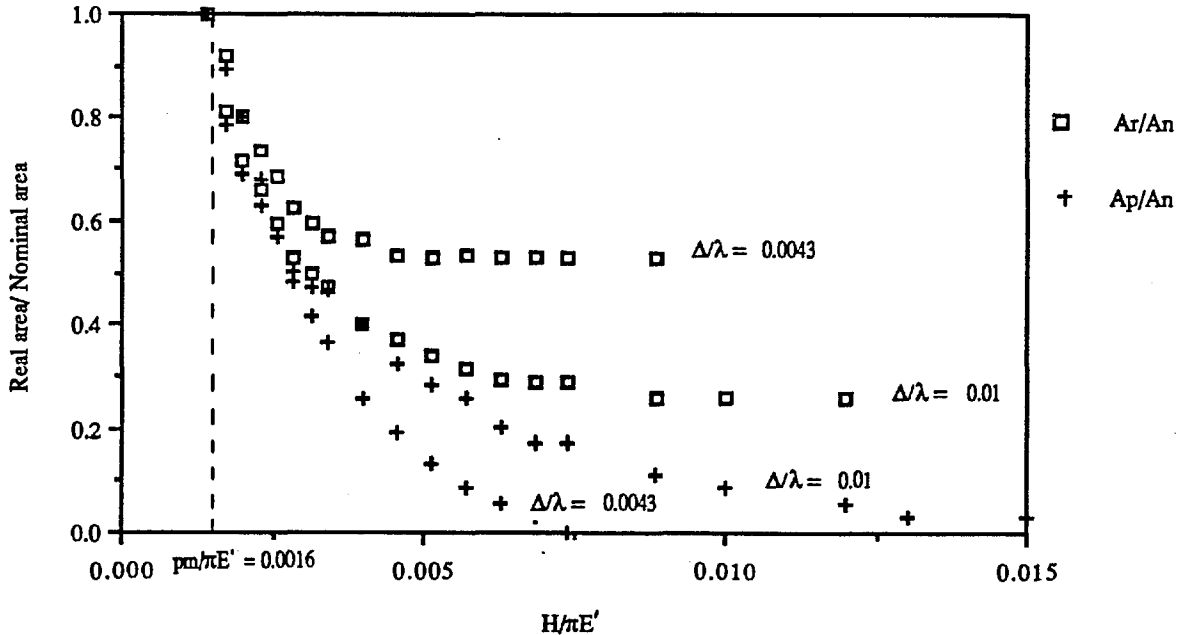


Figure 6.12 Effect of Hardness on real area and plastic area at a constant Δ/λ .

6.3.3 Effect of Hardness

Figure 6.12 shows the variation of the contact area with hardness. As H approaches p_m , then A_r/A_n or A_p/A_n approaches one regardless of the value Δ/λ . As H increases, the plastic area decreases and eventually the contact state will be purely elastic. Then the true area is independent of H .

6.4 Rough Surface Contact and Wavy Surface Contact Comparison

6.4.1 Effect of σ and β^*

The results of rough surface contact analysis in Chapter 5 were compared with the results obtained from sinusoidal and triangular surfaces. The σ and β^* of the wavy surfaces were obtained in the same way as the random surfaces, i.e. the main structure β^* of a wavy surface was defined as the autocorrelation function $\rho(\tau) = E\{y(x)y(x+\tau)\}$ such that $\rho(\beta^*) = 1/e$. Therefore, for the sinusoidal surface $\sigma = \Delta/\sqrt{2}$, $\beta^* = 0.19\lambda$ and for the triangular surface $\sigma = 0.583\Delta$, $\beta^* = 0.19\lambda$. From Figure 6.13 it can be seen that the results compare surprisingly well with those obtained from the triangular surfaces. At large σ/β^* , the results deviate from the triangular surface. This may be due to the effect of finite sampling which smooths out asperity sizes smaller than the sampling interval resulting in the overestimation of the contact area and reduction of plastic area, particularly at small β^* .

The distribution of real and plastic area of rough surface in Figure 5.10 and 5.12 are used to compare to wavy surfaces results which are shown in Figure 6.14 and 6.15. The results from the triangular surfaces show a good agreement in both cases.

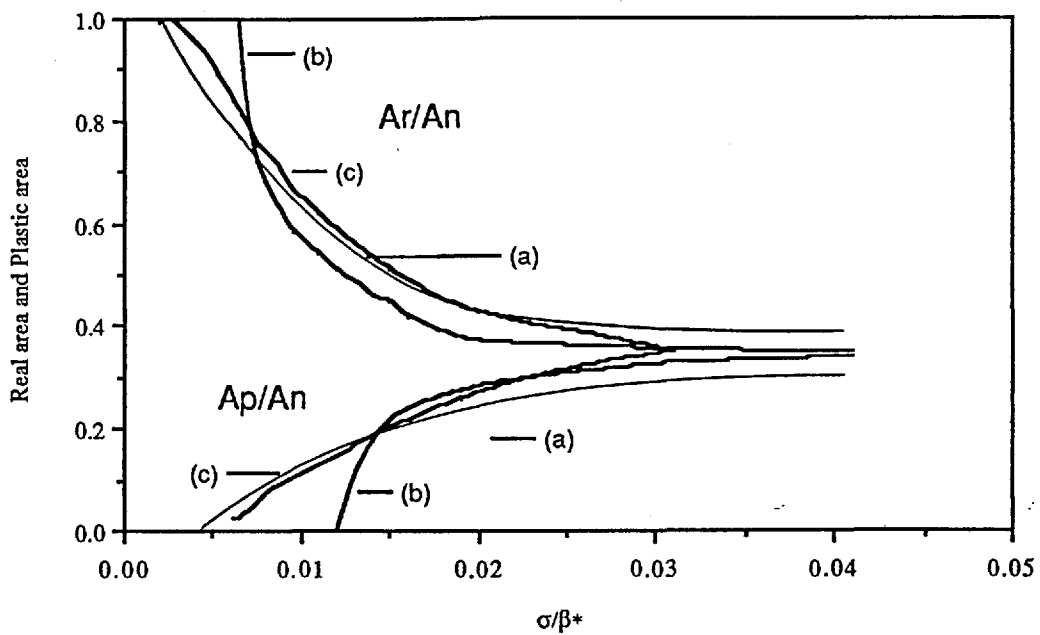


Figure 6.13 Comparison of real area and plastic area with σ/β^* .
 (a) Rough surface
 (b) Sinusoidal surface
 (c) Triangular surface

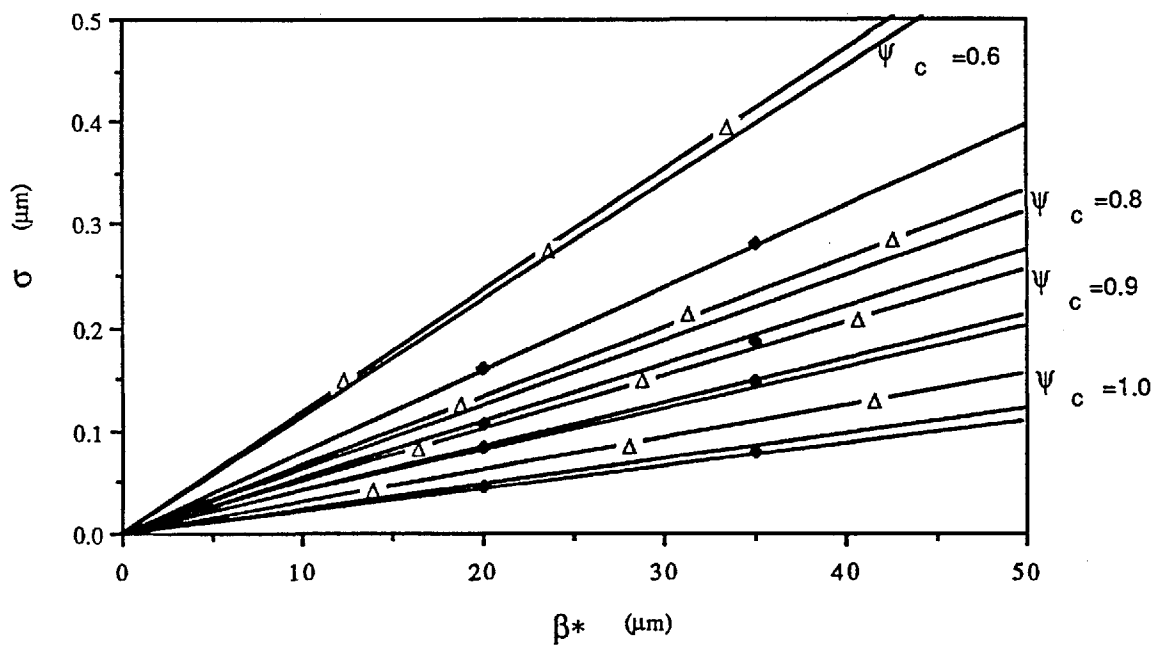


Figure 6.14 Comparison of the distribution of real area on a σ - β^* map.
 Rough surface —
 Triangular surface — Δ —
 Stochastic approach — $\blacklozenge$ —

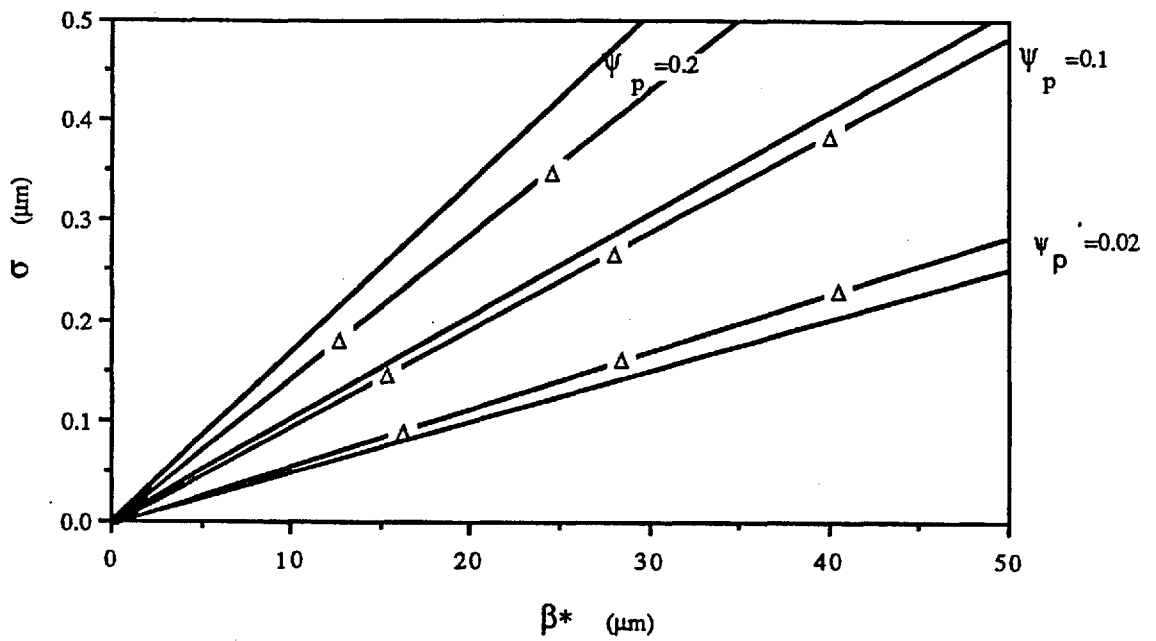


Figure 6.15 Comparison of the distribution of plastic area on a σ - β^* map.
 Rough surface —
 Triangular surface — Δ —

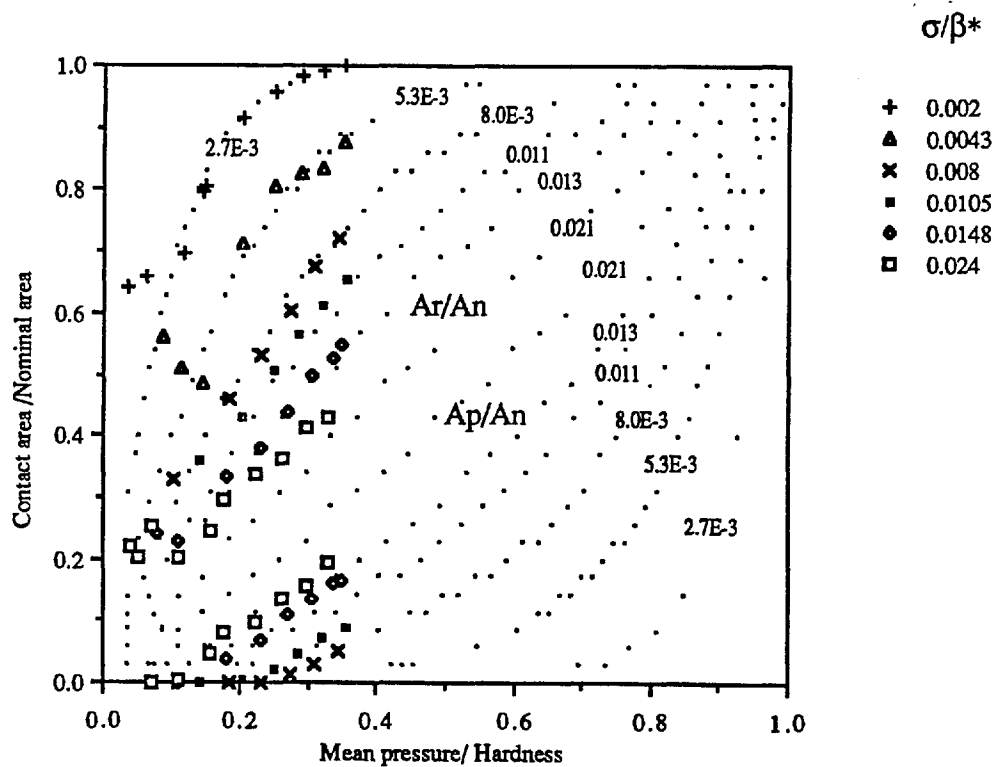


Figure 6.16 Comparison of real area and plastic area with load for different σ/β^* . The symbols represent results from rough surfaces. The values on the dotted curve represent results from triangular surface.

6.4.2 Effect of Load

Figure 6.16 shows the variation of real area and plastic area with load for six surfaces with different σ/β^* . The load was calculated for each separation up to 10N so that the corresponding value of p_m/H was up to 0.35. The results from the triangular surfaces are plotted on the same graph for comparison.

At low values of σ/β^* , the results compare well with that given by the triangular wave surface. However, at high value of σ/β^* , the contact area is higher than that of the triangular surface. This can be explained as being due to the effect of sampling technique as before. As σ/β^* decreases, the load required to bring the contact to total conformity also decreases.

At a very light load, contact area also appears higher than expected. In this circumstances, the load is supported on a few asperities and the contact area becomes smaller. As a result the effect of finite sampling is more significant.

6.5 Conformity of Rough Surfaces

It has been shown the conformity of a surface is dependent on the vertical and spatial parameters of the surface. It was decided to investigate the possibility of obtaining a general expression for a conformity parameter in terms of load, elastic-modulus and topographic parameters. Two methods were attempted. The first was to use the numerical results obtained from the triangular wave contact. The second was to employ a stochastic approach.

6.5.1 Surface Contact Approach

It has been shown that the results obtained from the triangular surfaces are reasonably close to the rough surfaces. Therefore, the solution given by the triangular wave contact was used to study the conformity conditions for random rough surfaces and this allowed the study be carried out in a short time.

For each value of Δ/λ of a triangular surface, the geometric approach of two bodies making contact was incremented until the contact area equal the nominal area at which the load p_c , required to attain total conformity, is obtained from contact analysis. Figure 6.17 shows the variation of $p_c/\pi E'$ with Δ/λ . For a purely elastic contact, the mean pressure is directly proportional to Δ/λ . The proportionality is found to be 1.60. When plasticity is included, as Δ/λ increases, the rate of increase of load for conformity is reduced due to the plastic deformation. However, in this analysis we have ignored the asperity persistence problem. For high values of H or small Δ/λ , we have

$$\frac{p_c}{\pi E'} = 1.6 \frac{\Delta}{\lambda} \quad (6.6)$$

Writing Δ/λ in terms of σ and β^* , i.e. $\sigma=0.583 \Delta$ and $\beta^*=0.19 \lambda$, we obtain the following relationship for conformity,

$$\frac{p_c}{\pi E'} = 0.52 \frac{\sigma}{\beta^*} \quad (6.7)$$

6.5.2 Stochastic Approach

An expression for the conformity parameter was also found by a stochastic

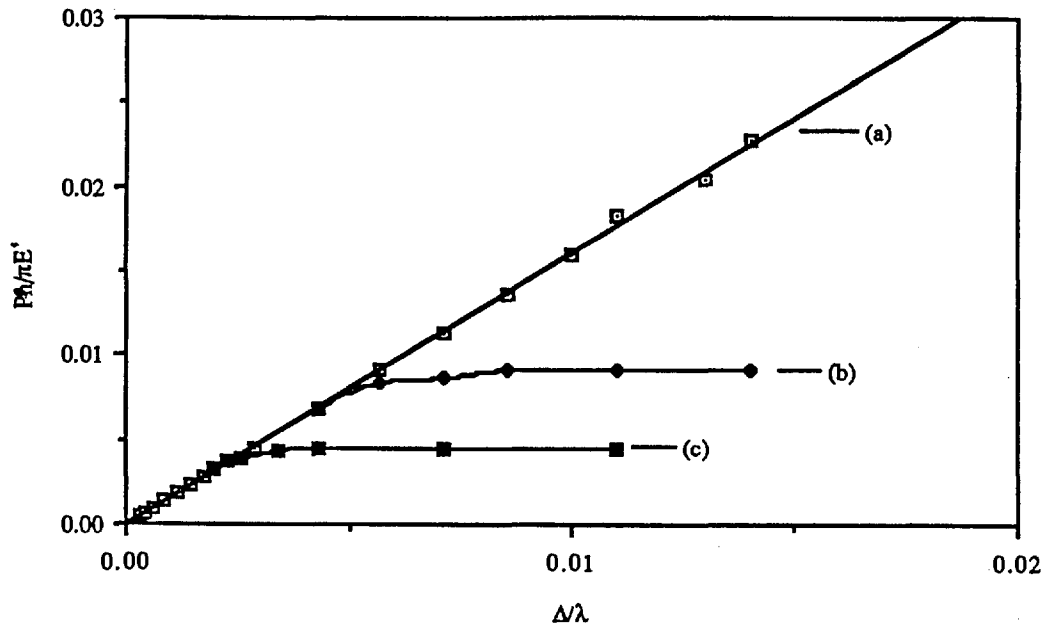


Figure 6.17 Variation of the nominal pressure at total conformity with Δ/λ of a triangular wave contact.

- (a) purely elastic
- (b) elastic-plastic, $H/\pi E' = 0.0091$
- (c) elastic-plastic, $H/\pi E' = 0.00455$

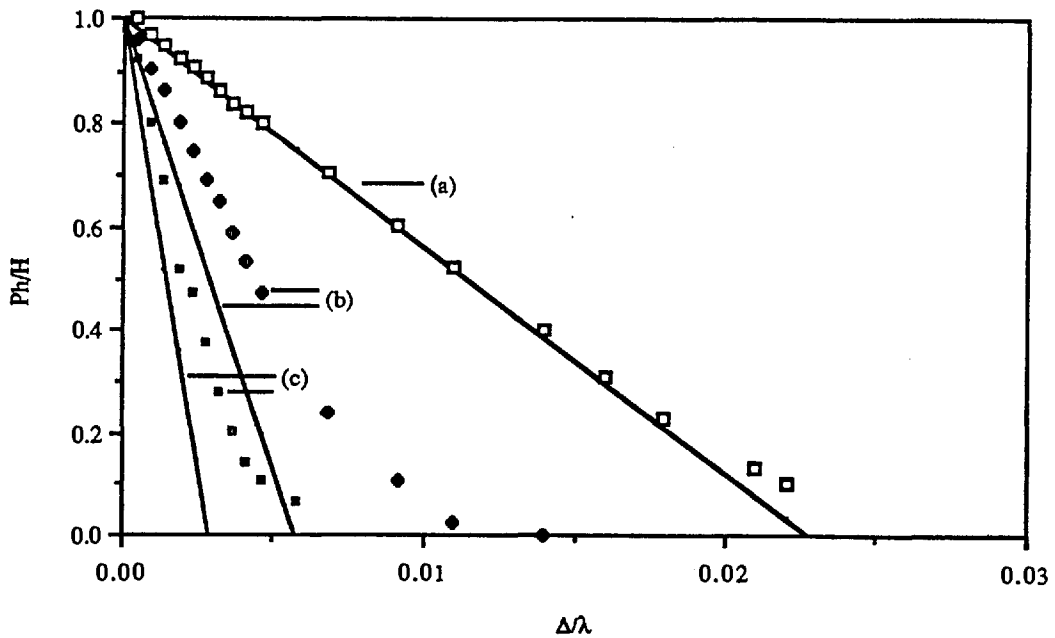


Figure 6.18 Variation of the nominal pressure at the onset of plasticity with Δ/λ of a triangular wave contact. The lines — are given by equation (6.20).

- (a) $H/\pi E' = 0.0227$
- (b) $H/\pi E' = 0.0091$
- (c) $H/\pi E' = 0.004551$

approach. It was assumed the profile slope, θ , of a rough surface has a Gaussian distribution. We then have,

$$f(\theta) = \frac{1}{\sigma_\theta \sqrt{2\pi}} \exp\left(-\frac{\theta^2}{2\sigma_\theta^2}\right) \quad (6.8)$$

and

$$\theta = \frac{z(x + \tau) - z(x)}{\tau} \quad (6.9)$$

where τ is the sample interval and $z(x)$ is the surface height at x .

The variance of the slope is given by,

$$\begin{aligned} \sigma_\theta^2 &= E(\theta^2) = E\left[\left(\frac{z(x + \tau) - z(x)}{\tau}\right)^2\right] \\ &= \frac{1}{\tau^2} E\{z^2(x + \tau) + z^2(x) - 2z(x + \tau)z(x)\} \\ &= \frac{1}{\tau^2} [E\{z^2(x + \tau)\} + E\{z^2(x)\} - 2E\{z(x + \tau)z(x)\}] \\ &= \frac{1}{\tau^2} [2\sigma^2 - 2R(\tau)] \quad \text{because } E\{z^2(x + \tau)\} = E\{z^2(x)\} = \sigma^2 \\ &= \frac{2\sigma^2}{\tau^2} (1 - \rho(\tau)) \end{aligned} \quad (6.10)$$

where $R(\tau)$ and $\rho(\tau)$ are the correlation function and autocorrelation function respectively. According to Whitehouse and Archard (1970), the autocorrelation function of many engineering surfaces has an exponential form. i.e. $\rho(\tau) = \exp(-\tau/\beta^*)$. Then equation (6.10) becomes,

$$\frac{\sigma_\theta}{\sigma} = \frac{\sqrt{2(1 - \exp(-\frac{\tau}{\beta^*}))}}{\tau} \quad (6.11)$$

The expression shows that the variance of profile slope is not an intrinsic property of a random rough surface but depends on the choice of sampling interval. In the problem

of elastic conformity, it is likely that it is the broad scale structure which is of interest. Therefore, taking $\tau=2.3\beta^*$ as a spacing of events such that the profiles can be regarded as statistically independent, equation (6.11) becomes,

$$\sigma_\theta = 0.583 \frac{\sigma}{\beta^*} \quad (6.12)$$

As shown previously, the load required for total conformity of a triangular wave profile is given by equation (6.6). Writing the Δ/λ in terms of the profile slope, θ_c , i.e. $\theta_c=4\Delta/\lambda$ for small angles of triangular wave, we obtain,

$$\theta_c = \frac{p_c}{0.4\pi E'} \quad (6.13)$$

For a random surface, it can be assumed that any profile slope $\theta < \theta_m$ under the influence of mean pressure p_m will conform, where $\theta_m = p_m/0.4\pi E'$. Therefore, the probability that an asperity will conform at a mean pressure p_m is given by,

$$\begin{aligned} \Pr(|\theta| < \theta_m) &= \frac{1}{\sigma_\theta \sqrt{2\pi}} \int_{-\theta_m}^{\theta_m} \exp\left(-\frac{\theta^2}{2\sigma_\theta^2}\right) d\theta \\ &= 2\text{erf}\left(\frac{\theta_m}{\sigma_\theta}\right) \end{aligned} \quad (6.14)$$

The real contact area is the total area of asperities that conform. Therefore A_r/A_n is merely equal to the probability of an asperity to conform which is given by equation (6.14). For $A_r/A_n=1$, then from equations (6.12), (6.13) and probability tables, we have

$$\frac{p_m}{\pi E'} = 0.70 \frac{\sigma}{\beta^*} \quad (6.15)$$

This is close to that predicted from the triangular wave contact solution given by

equation (6.7). The difference can be explained by the shape of the asperity model. With reference to Figure 6.13, the variation of elastic area with Δ/λ or σ/β^* is sensitive to the asperity shape. For a random rough surface, the surface can be considered to be formed by a range of triangular asperities with different angles. The load required to conform the rough surface is expected to be higher than that for a triangular surface with a single asperity angle due to asperity persistence or asperity stress field interaction. Therefore equation (6.14) is preferable to equation (6.7) for predicting the contact conformity of random rough surfaces.

As a matter of interest, it is to be expected that the load required to conform a sinusoidal surface is much lower due to the smooth form of the waviness. The expression for total conformity is given by equation (6.2), putting $2a=\lambda$ and writing Δ, λ in terms of σ and β^* , we obtain $p_m/\pi E' = 0.27\sigma/\beta^*$ for sinusoidal surfaces. Thus the load required to conform a sinusoidal surface, triangular surface and rough surface increases on that order for a given σ/β^* .

Similarly, using the stochastic approach for $A_r/A_n=0.9$

$$\frac{p_m}{\pi E'} = 0.38 \frac{\sigma}{\beta^*} \quad (6.16)$$

and for $A_r/A_n=0.8$

$$\frac{p_m}{\pi E'} = 0.30 \frac{\sigma}{\beta^*} \quad (6.17)$$

and for $A_r/A_n=0.6$

$$\frac{p_m}{\pi E'} = 0.20 \frac{\sigma}{\beta^*} \quad (6.18)$$

and so on.

Equations (6.15) to (6.17) were used to compare the results obtained from rough surfaces contact analysis. In the case of a steel ball on stainless steel at a load of 10N, $p_m/\pi E' = 0.0016$. The lines of $\psi_c = A_r/A_n$ from the stochastic approach were plotted on the σ - β^* map shown in Figure 6.14. The lines of $\psi_c = 0.9$ and 1.0 agree well with the rough surface contact results. However when the values of ψ_c less than 0.8, the lines of ψ_c tend to deviate from the rough surface contact results.

The contact area given by stochastic approach is always less than that given by contact analysis. There are two possible reasons for this. Firstly, the expression obtained from the stochastic approach ignores the effect of plastic deformation. If the plasticity of a surface is significant, the elastic area will be underestimated. Secondly, the load concentration of a ball on a flat surface is higher than that in the contact of two normal flat surfaces. It is expected that the contact area of a ball on a flat surface for a given nominal pressure will be higher than that given by the contact of two nominally flat surfaces. The results given by the triangular wave contact conform to the results given by a ball on rough surfaces. This may be due to the load concentration occurring in both types of contact.

6.6 Plasticity of Rough Surfaces

In this section, contact results given by the triangular surface model are used to investigate the conditions of plasticity, defined to be when a surface is at the state of the onset of plastic deformation, say 2% plastic area. This can be determined by two methods.

The first method is to obtain the load required to bring the contact to 2% plastic area by numerical contact analysis on triangular surfaces with different Δ , λ and H and then express the onset of plasticity in terms of load, Δ , λ and H . The second method is to make use of the symmetry of the curves of A_p/A_n to the curves of A_r/A_n , see Figure 6.11. If the conditions for conformity are known, then the onset of plasticity can be determined. Here, the second method was used, firstly to determine the conditions for the onset of plasticity and then the results were verified by the first method. By symmetry, we have

$$\frac{p_h}{H} = 1 - \frac{p_c}{H} \quad (6.19)$$

where p_h is the mean pressure at the onset of plastic deformation. From equation (6.6), and noting that this is only holds for large H or small Δ/λ , we have

$$\frac{p_h}{H} = 1 - \frac{1.6\pi E' \Delta}{H \lambda} \quad (6.20)$$

where p_h is the mean pressure at the onset of plastic deformation.

Results obtained from numerical analysis on triangular surfaces are shown in Figure 6.15 to check the validity of this equation. For small H and large Δ/λ , the numerical results appear to deviate from that predicted by equation (6.20). This is due to the fact that the equation (6.20) is based on the assumption that H is high and the effect of plastic deformation is small. For large H or small Δ/λ , the results show a good agreement with that predicted by equation (6.20).

For a random surface, the expression for conformity predicted by the stochastic approach is given by equation (6.15). Substituting this equation into equation (6.19), we have,

$$\frac{p_h}{H} = 1 - \frac{2.2E' \sigma}{H \beta} \quad (6.21)$$

If $H \gg p_h$, the left hand terms will become zero. The onset of plasticity is given by $\sigma/\beta^* = 0.45 H/E'$, a form of the well known plasticity index (Greenwood 1966, Whitehouse 1970) which is only dependent on the topographic and material parameters. However, if p_h is comparable to H , then the plasticity is also dependent on the load.

It is apparent that if two materials with different hardness are subjected to an increased load, the softer material will undergo plastic deformation first. Hirst and Hollander showed that rough surfaces damage can be related to plasticity. Their results were based on a hard stainless steel ball ($H=800H_v$) on a soft stainless steel ($H=160H_v$). They used three different loads which give $p_h/H=0.22, 0.35$ and 0.6 . From the consideration of equation (6.21), it is apparent the plasticity decreases significantly with increasing loading to hardness ratio. Hirst and Hollander's results also showed that the boundary of safe sliding on a $\sigma-\beta^*$ map decreases with increasing this loading ratio. Therefore, they confirmed that the plasticity is also dependent on the load if the load is comparable to H as predicted by equation (6.21).

In the case of $H \gg p_h$, we can write $\psi = (E'/H)(\sigma/\beta^*)$, so that with $\psi < 0.45$, the contact will be purely elastic and with $\psi > 0.45$, the contact will be elastic-plastic. Greenwood and Williamson used a stochastic approach and proposed $A_p/A_c < 0.02$ as the condition for an essentially elastic deformation mode. They concluded that for $\psi < 0.6$ the contact will be predominantly elastic and for $\psi > 1.0$ the contact will be predominantly plastic. It should be noted that the Greenwood and Williamson plasticity criterion is a convenient measure of the extent of plastic deformation. Their A_p is not the actual plastic area because it was obtained using elastic contact results. Their model which was constant radius asperity summits is likely to underestimate the degree of plastic area.

From a numerical contact analysis or the stochastic approach described earlier, we

can predict the elastic area or plastic area for a given value of ψ . It is of interest to know the actual plastic area that would correspond to Greenwood and Williamson's plasticity criterion. The calculation is as follows. The expression for a particular value of A_r/A_n is given by equations (6.15) to (6.17), i.e. $p_c/\pi E' = c\sigma/\beta^*$, where c is dependent on the value of A_r/A_n . Using equation (6.19),

$$\frac{p_h}{H} = 1 - \frac{\pi E' c \sigma}{H \beta^*} = 1 - \pi c \psi \quad (6.22)$$

when $H \gg p_h$,

$$c = \frac{1}{\pi \psi} \quad (6.23)$$

For $\psi=0.6$ and 1.0 , c is equal to 0.53 and 0.318 respectively. From equation (6.15) to (6.18), the corresponding values of A_r/A_n are found by interpolation to 0.95 and 0.82 respectively. From the symmetrical relationship of A_r/A_n to A_p/A_n , the plastic areas at $\psi=0.6$ and 1.0 would be 5% and 18% respectively. Therefore with $A_p/A_n < 5\%$, the contact can be considered as predominantly elastic and with $A_p/A_n > 18\%$, the contact can be regarded as predominantly plastic. It will be shown in Chapter 8, that the upper boundary which divide the safe and unsafe sliding regions due to plasticity corresponds to a value of $A_p/A_n \approx 15\%$.

6.7 Conclusion

The extent of elasticity and plasticity of a surface in contact with another can be completely defined by two topographic parameters. One describes the vertical roughness and the other describes the spatial orientation of surface height. For periodic wavy

surfaces, the parameters are the amplitude Δ and wavelength λ of the profile. For random surfaces, they are the surface roughness σ and the correlation length β^* .

The variation of elastic area with Δ/λ depends on the shape of the modelled asperities as shown by the differences between the contact of sinusoidal and triangular profiles. However, for fully plastic flow, the area does not depend on the shape of the asperities and can be defined as the ratio of the load to the flow pressure of the material.

An interesting relationship emerges when A_r/A_n and A_p/A_n are plotted against load. The contact areas defined by A_r/A_n and A_p/A_n appear to be symmetrical about the diagonal, i.e. given a load p_1/H and real area of contact A_1/A_n , the corresponding load p_2/H to produce plastic area $A_p/A_n = 1 - A_1/A_n$ is equal to $1 - p_1/H$.

The variation of elastic area and plastic area with σ/β^* given by the triangular wave contact analysis compares well with that given by the random rough surface contact analysis. It can reasonably be used to predict the contact of a ball on a rough surface for given load, hardness of the material and topographic parameters.

The triangular wave model has been extended to study the conditions for total elastic conformity and plasticity. For total elastic conformity, the condition is given by $p_c/\pi E' = 0.52\sigma/\beta^*$. A stochastic approach has also been attempted to derive an expression for total elastic conformity which is $p_c/\pi E' = 0.7\sigma/\beta^*$. The difference may be due to the shape of the modelled asperities. But as the contact area decreases, the contact area predicted by the triangular wave model compares better than that given by the stochastic approach. This may be explained by the effects of plastic deformation and load concentration.

For the elastic-plastic contact analysis as in the case of a steel ball on a stainless steel which has relatively low hardness, the load concentration and plastic deformation are

significant. The expression due to the stochastic approach ignores this plasticity since it was derived based on the contact of two nominal flat surfaces.

As a whole, the contact analysis results given by the triangular wave model can be reasonably accepted to predict the contact of a ball on a nominal rough surface for a wide range of σ/β^* . If the surface contact condition becomes close to total conformity, the stochastic contact model can be used.

The condition for the onset of plasticity has been derived by using the symmetrical relationship of the curves of A_r/A_n to A_p/A_n , i.e. the load p_r/H required to bring the contact to the onset of plasticity is equal to one minus the load p_o/H necessary to bring the contact to reach total conformity, thus giving $p_r/H = 1 - (1.6\pi E'/H)(\Delta/\lambda)$. This expression has been shown to agree well with the contact analysis of triangular wave surfaces provided that $H \gg p_n$ or Δ/λ is small.

Similarly, for a random rough surface, the conformity expression given by the stochastic model has been used to obtain the condition for the onset of plasticity which is given by $p_r/H = 1 - (2.2E'/H)(\sigma/\beta^*) = 1 - 2.2\psi$. If the load is comparable to H , the plasticity is also dependent on the load as well. The well known plasticity index, which is only dependent on the topographic and material parameters, is only valid if the contact stresses are much smaller than the flow pressure of the softer material.

For a given plasticity index, the corresponding plastic area can be calculated. For $\psi = 0.6$ and 1.0 , the corresponding plastic area is 5% and 18% respectively. To coincide with the Greenwood plasticity criterion, when $A_p/A_n < 5\%$, the contact will be predominantly elastic and when $A_p/A_n > 18\%$, the contact will be predominantly plastic.

CHAPTER 7

MECHANISTIC STUDY OF SLIDING FRICTION AND SURFACE DAMAGE

7.1 Introduction

The mechanistic processes in wear and friction can be generalised as adhesion at the interface, rupture of adhesion, displacement of material by ploughing and shearing, detachment of displaced material, fatigue and cracking, oxidation etc. The overall observation in cases of real wear and friction may involve their interdependent operation, either sequentially or simultaneously. Much of the overt phenomenology of wear consists of orderly and predictable combinations and sequences of the basic mechanisms. The following chapter is concerned with the basic mechanistic phenomena that take place in the interface of contacting bodies based upon the observations obtained by sliding friction tests of a steel or ceramic ball on different materials such as stainless steel, gear steel and copper. Three different types of surfaces, namely rough, smooth and very smooth, were prepared to study the effects of surface finishing on sliding friction and damage. In this chapter, 'rough surface' refers to surfaces having a high value of σ/β^* . The contact will be predominantly plastic except on gear steel. 'Smooth surfaces' have an intermediate value of σ/β^* . The contact will be predominantly elastic. 'Very smooth surfaces' have a low value of σ/β^* . The contact will reach almost, or total, conformity.

7.2 Friction Apparatus

A rig was built for the friction tests. It is similar to a Bowden-Leben machine and consists of three basic parts;

- 1) The horizontal carriage and its drive system.
- 2) The control system and heater.
- 3) The loading arm with its lifting mechanism.

Figure 7.1 shows the interrelationship of the parts.

7.2.1 The Carriage

The carriage rests on two stainless steel rails, circular in cross-section. One rail located in a Vee-groove on the underside of the carriage, whilst the other rests on a flat.

Motion is transmitted to the carriage by means of a lead-screw bolted to the carriage. The lead-screw itself is supported by a bearing arrangement capable of taking thrust in either direction; the lead-screw is directly coupled to the output shaft of a gearbox. The output of the gearbox is connected to the armature of a D.C. motor, with a tachogenerator winding as part of the motor armature. This arrangement gives a very accurate feedback-dependent measure of the motor speed. The tachogenerator and motor are connected to a constructed electronic speed controller, capable of consistent speed control over a 400:1 range, with marginal loss in available torque. The motor could be driven at approximately 120-5000 rpm. A gearbox ratio of 40:1 and a lead screw pitch of 5.08 mm gives carriage traverse speeds of approximately 0.025 mm/s to 10 mm/s.

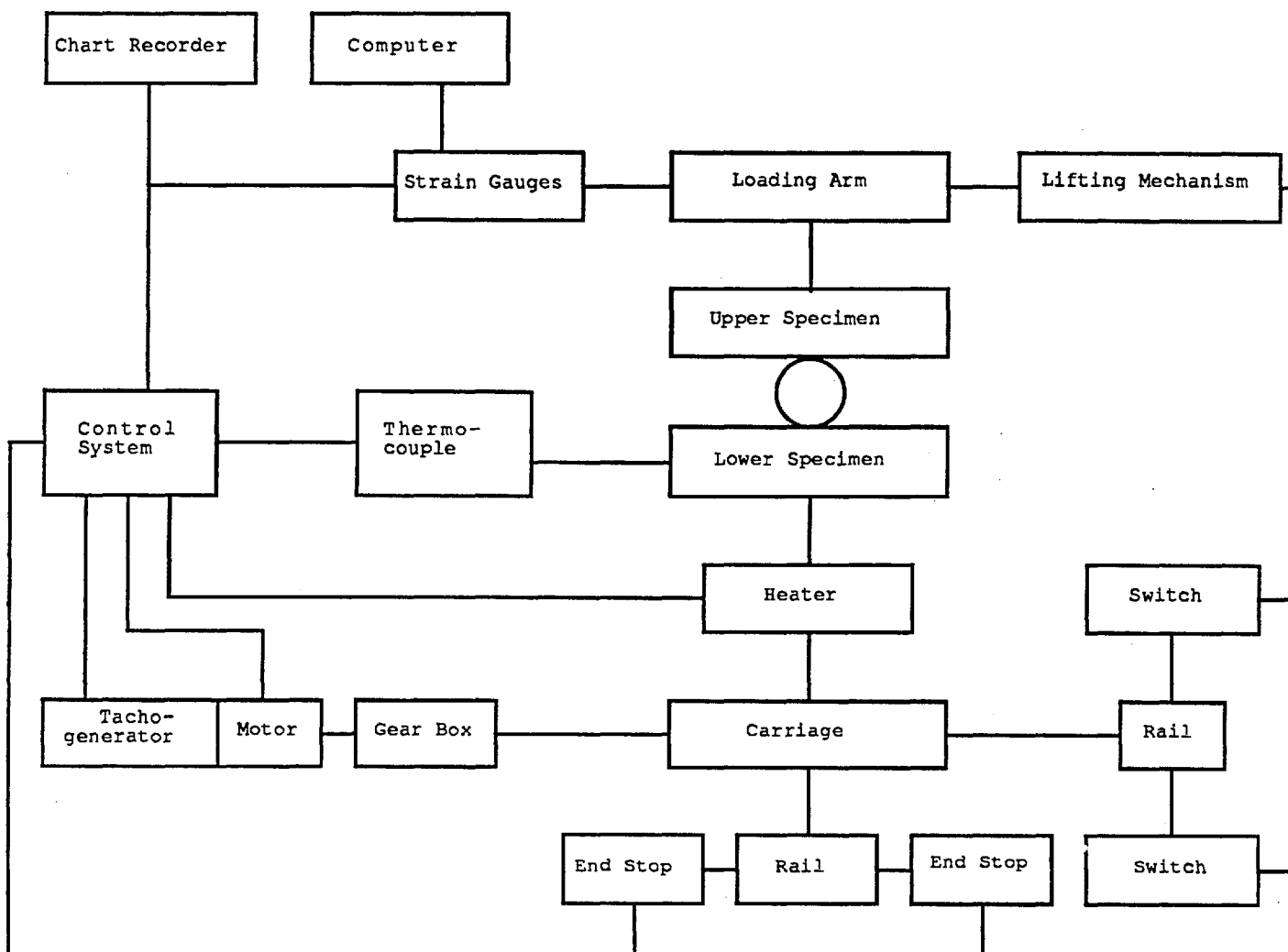


Figure 7.1 Friction rig block diagram

7.2.2 Control System

The control module has four separate functions:-

- (1) It is required to move the slide at constant speed in one direction then return it to the starting position, as quickly as possible, for the next cycle. This is achieved by means of a servo-loop which incorporates a tacho-generator on the DC drive motor. The required speed is set by means of a potentiometer, which increases the DC motor speed from zero to the speed which causes the tacho-generator to output a voltage equal and opposite to the potentiometer voltage. A null output signifying that the motor is running at the required speed. If the loading of the slide varies so as to alter the motor speed, an offset voltage occurs which is fed-back through the servo-loop to quickly re-establish the correct speed.

A magnetic-reed switch at each end of the slide track closes when the magnet attached to the slide approaches within 10 mm. The DC drive motor then reverses direction automatically and cycles back and forth, the number of cycles being counted by a simple electromechanical counter. A motor stop switch is provided for manual use during the setting-up procedure.

- (2) Two strain gauges were mounted on the loading arm to measure the friction at the contacting surfaces. The amplifier gain may be switched in steps of 1,10,100 with a time constant of 1 ms or 10 ms for each gain setting. A strain cut-out is incorporated so that the motorised-slide stops if the friction increases above any particular preset level.
- (3) The oil bath is heated to any given temperature, up to 250 °C, by means of a

thermocouple and heater element servo-control loop. A temperature stability of about 1 °C was found to be typical.

- (4) Oil bath temperature is monitored continuously by means of a platinum resistance sensor. The slide speed, strain gauge output and oil bath temperature can each be read gauge consecutively on a digital meter, by manual switching.

7.2.3 Loading Arm

The main feature of the loading arm is a pivoted assembly which ensures a constant load between the specimens. Friction is measured by a pair of strain gauges mounted on each side on the cantilever. Different thicknesses of cantilever are used so that the signal to noise level at the recorder is negligible. The thickness of cantilever used over each load range was as follows;

Thickness of cantilever(mm)	Loading(kgf)
3	0.1-0.5
4	0.6-1.0
6	1.0-5.0
8	5.0-8.0
10	8.0-10.0

A loading arrangement was designed and built which could automatically lift and gently lower the arm. Lifting was achieved by a pneumatic cylinder. A taper rod, which connects to the cylinder piston rod, pushes the cantilever upwards to lift the specimen.

A flow regulator is used to control the flow of air and hence the speed of the piston

rod. In this way the arm can be lowered slowly so as not to damage the specimen surface, and also be lifted up quickly. Pneumatic switches were clamped on the rail to change the direction of piston movement. These can be adjusted on each side so as to co-ordinate with the movement of the carriage.

The lateral position of the track on the lower specimen can be set by moving the whole arm and its base in a slideway fixed on the base of the machine. After each movement of the carriage, a new track can be set prior to the next measurement. Normally the slideway is clamped to prevent any free-play in the friction measuring system.

A ball holder is attached to the loading arm to form the upper specimen support. The nominally flat lower specimen is mounted in the carriage (which is traversed at constant speed by the lead screw). The friction trace were plotted directly on a chart recorder, and the equivalent analogue signals were also recorded and stored on a BBC microcomputer.

7.3 Friction Experiments

7.3.1 Experimental details

In all the experiments the lubricant used was a 1% solution of stearic acid in white oil. A sliding speed of 0.04 mm/sec and a load of 10 N were used. The upper specimen consisted of a steel ball, (En31 bearing steel), or a ceramic ball. The lower specimen was stainless steel, gear steel or copper. The test was carried out on a new track for each combination of contacting bodies,. The surfaces were prepared by using silicon carbide abrasive papers of varying grade and condition, to produce different roughness type. Very smooth surfaces were produced by using P1200 grit size of abrasive paper and then

lapping with 1 μm diamond paste. The specimens and ball were cleaned ultrasonically in toluene and acetone prior to testing. They were then dried under infra-red light prior to each friction test. Measurements were made using a Talysurf 4 with a fine tip stylus, which recorded profiles on an LS1-11/23 based micro-computer. The profiles were subjected to a high-pass E-system filter. The radius of the filter's envelope was taken as equal to that of the ball.

7.3.2 Steel Ball on Stainless Steel

7.3.2.1 Rough surfaces

The friction traces of a surface with $\sigma=0.36 \mu\text{m}$ and $\beta^* = 4 \mu\text{m}$ are shown in Figure 7.2a, b and c. The friction curves show transitional behaviour from relatively stable and constant friction to an increasing and highly variable trace. A typical wear track at which the transitional friction occurred is shown in Figure 7.3a. It can be seen that a series of fine parallel scratches appeared at a distance from the starting point of sliding. This is consistent with the sliding distance at which the frictional transition occurred. Figure 7.4 showing the surface profiles after sliding, also revealed the transitional behaviour from steady plastic deformation of the higher ridges to the complete obliteration of the original profile. The extent of scratching increased significantly with sliding distance, as shown in Figures 7.3b and 7.3c. Figures 7.5 and 7.6 show the wear scar on the ball and a set of parallel surface profiles across it. The profiles indicated that a certain type of material adhered to the ball. This may have come from the countersurface, the ball or some other source. The EDAX facility on the scanning electron microscope showed that the spectrum of the wear scar on the ball was consistent with that of the countersurface, indicating that the adhered material originated from the countersurface.

Figure 7.7 shows the friction traces of a less rough surface with $\sigma=0.24 \mu\text{m}$ and

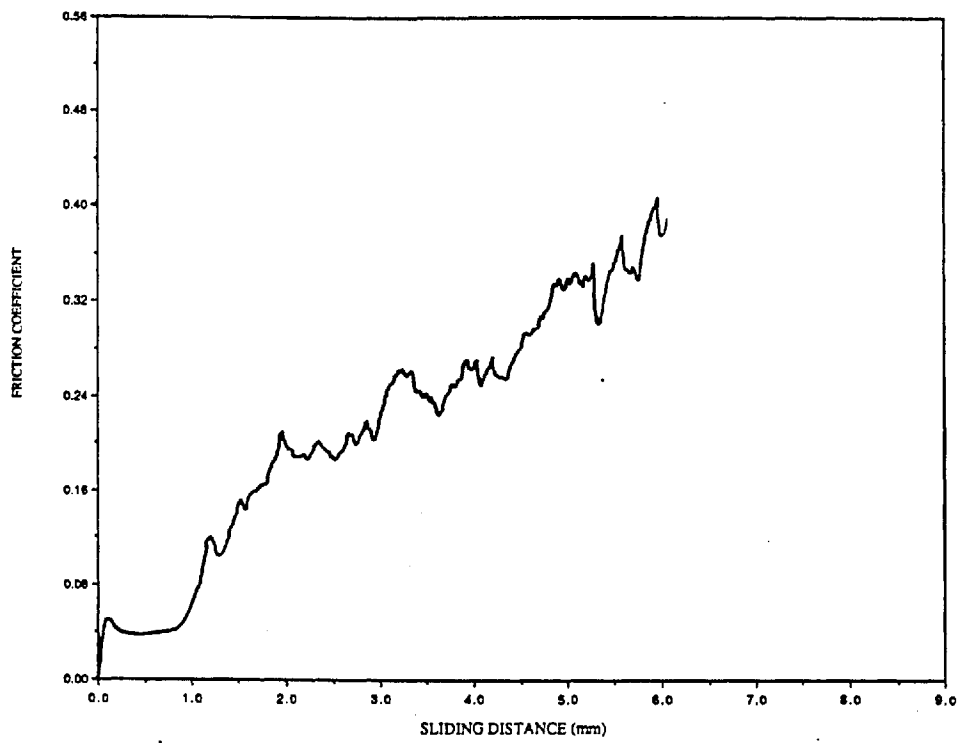


Figure 7.2a Friction traces of a steel ball on a rough stainless steel surface, $T = 50^{\circ}\text{C}$.

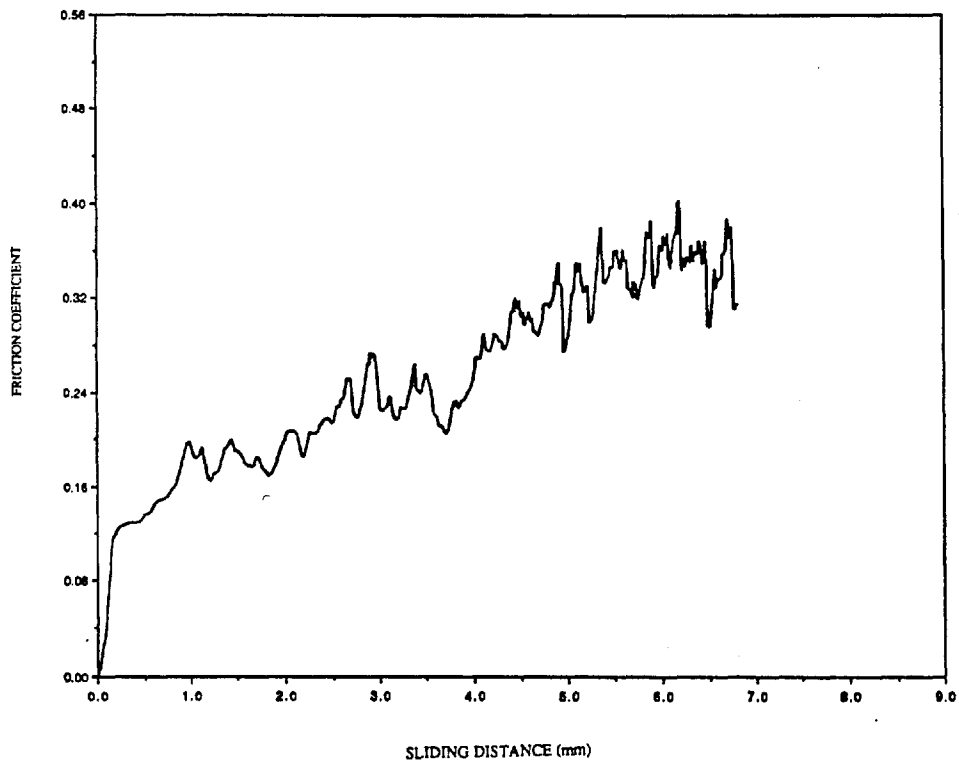


Figure 7.2b As Figure 7.2a but with $T = 100^{\circ}\text{C}$.

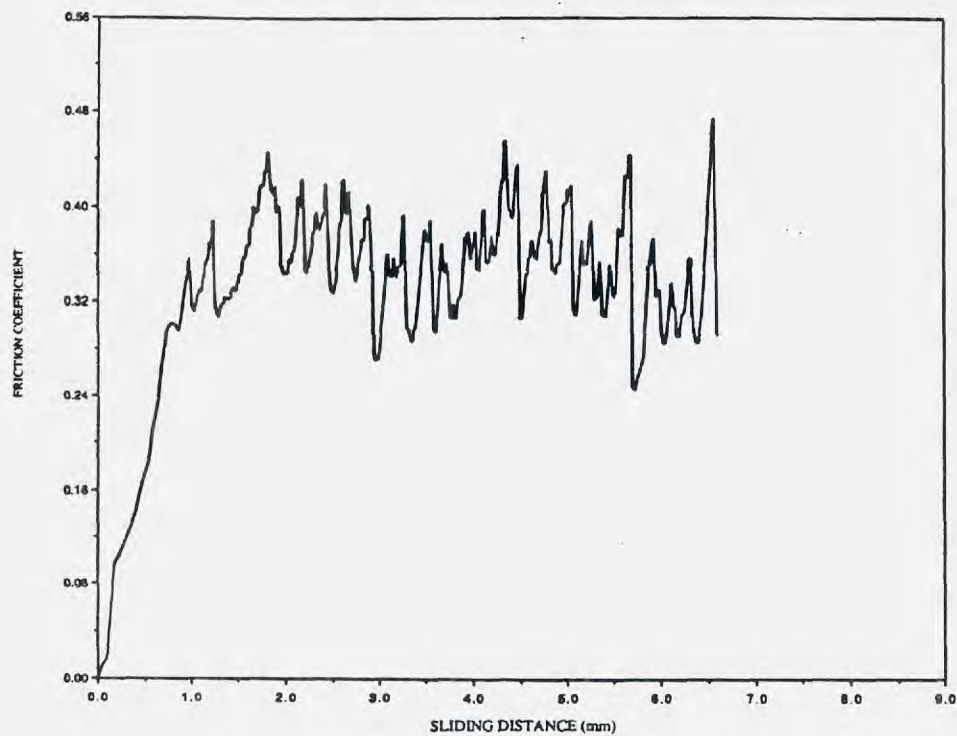


Figure 7.2c As Figure 7.2a but with $T = 150^{\circ}\text{C}$.

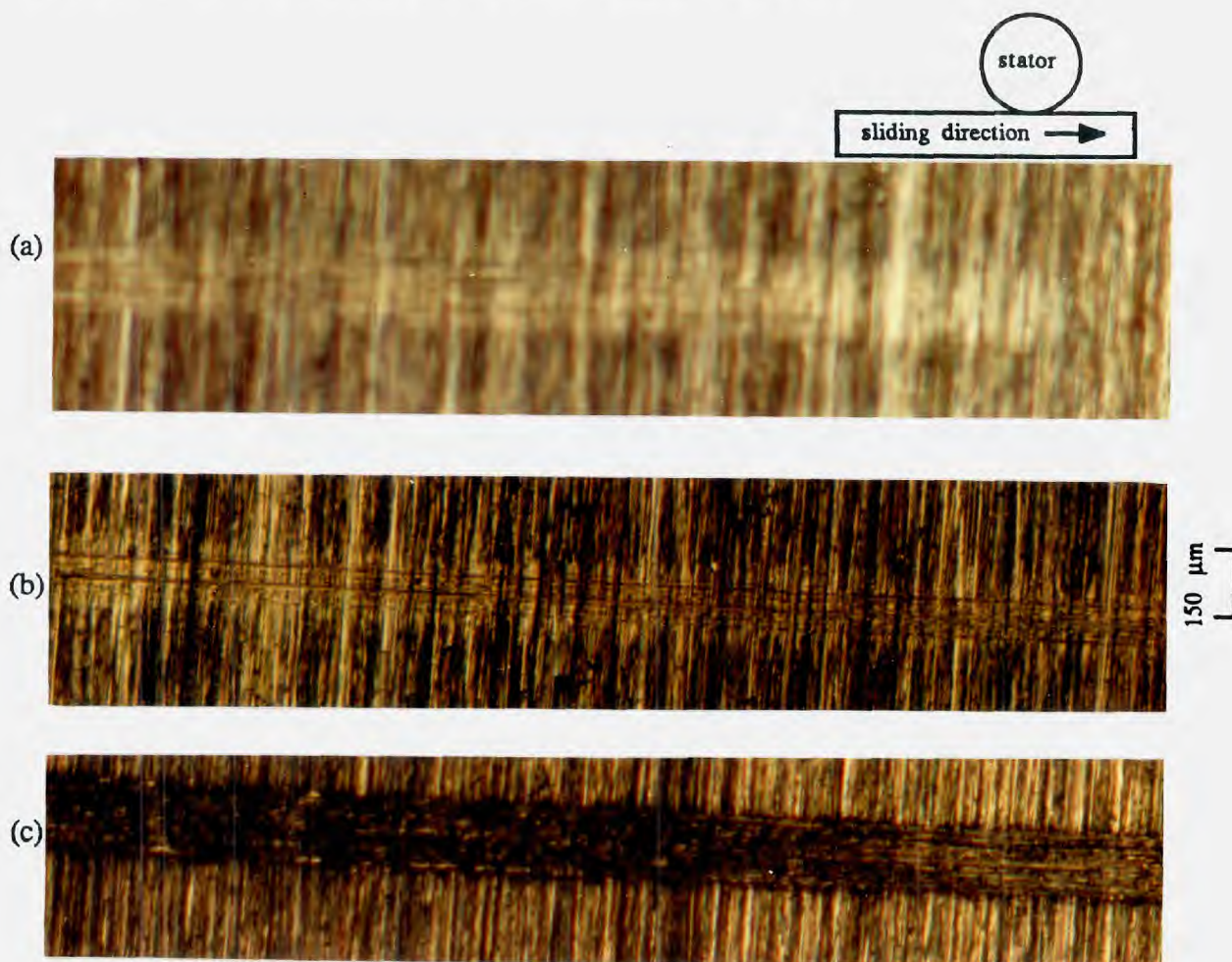


Figure 7.3 Wear track on the countersurface at $T = 100^{\circ}\text{C}$. Sliding position (a) at the beginning, (b) at the middle, (c) at the end.

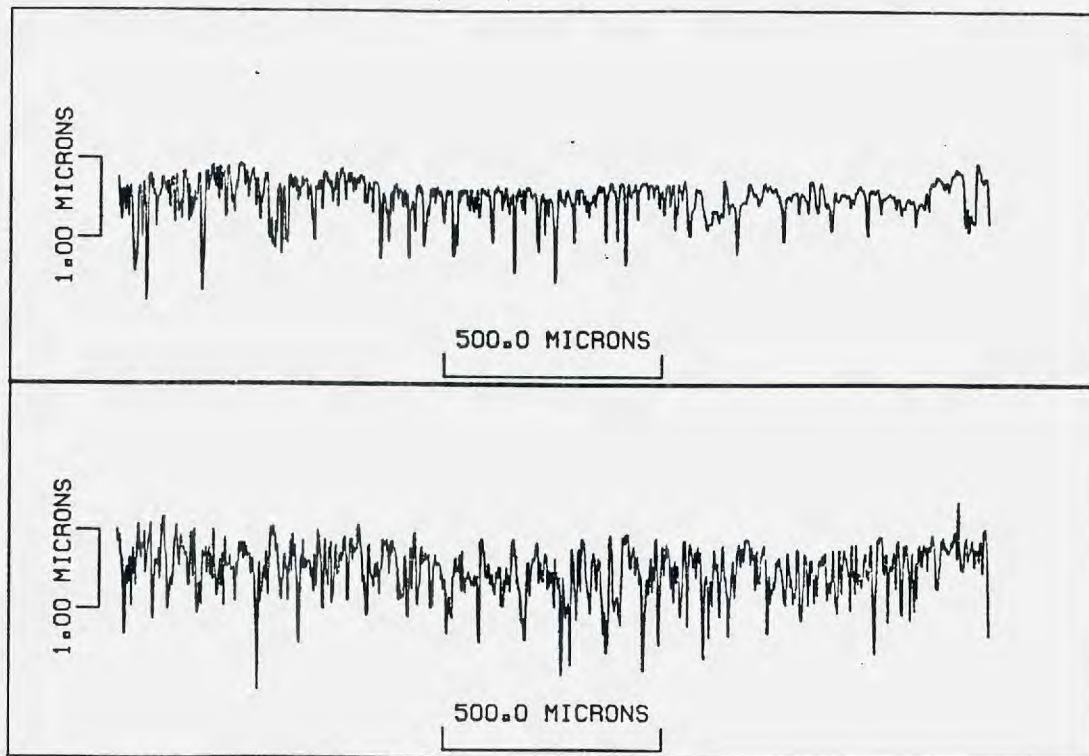


Figure 7.4 Surface profiles along the wear track at $T = 100^{\circ}\text{C}$, Lower: before sliding, Upper: after sliding.

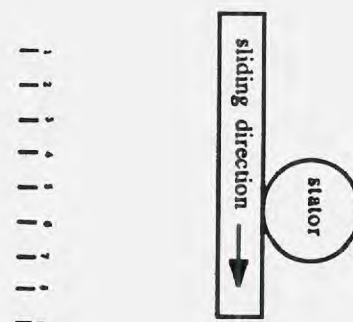
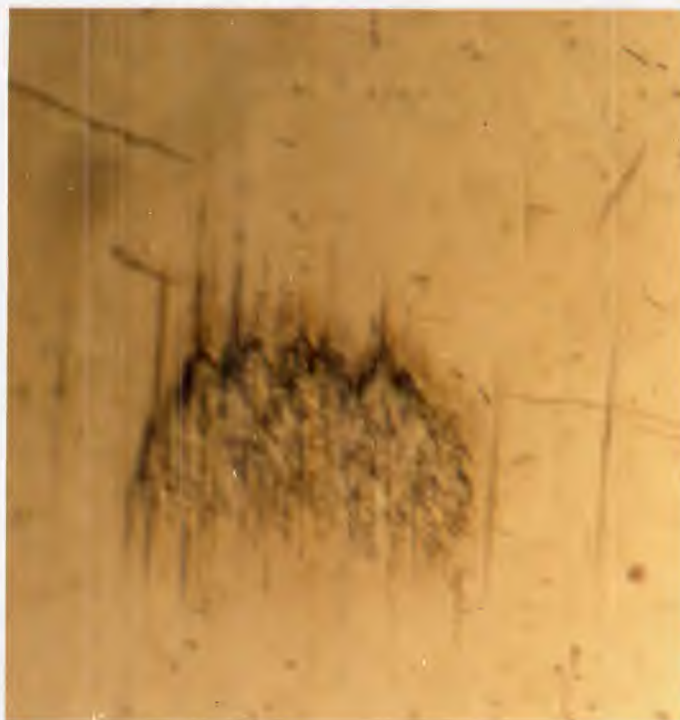


Figure 7.5 Wear scar on ball at $T = 100^{\circ}\text{C}$.

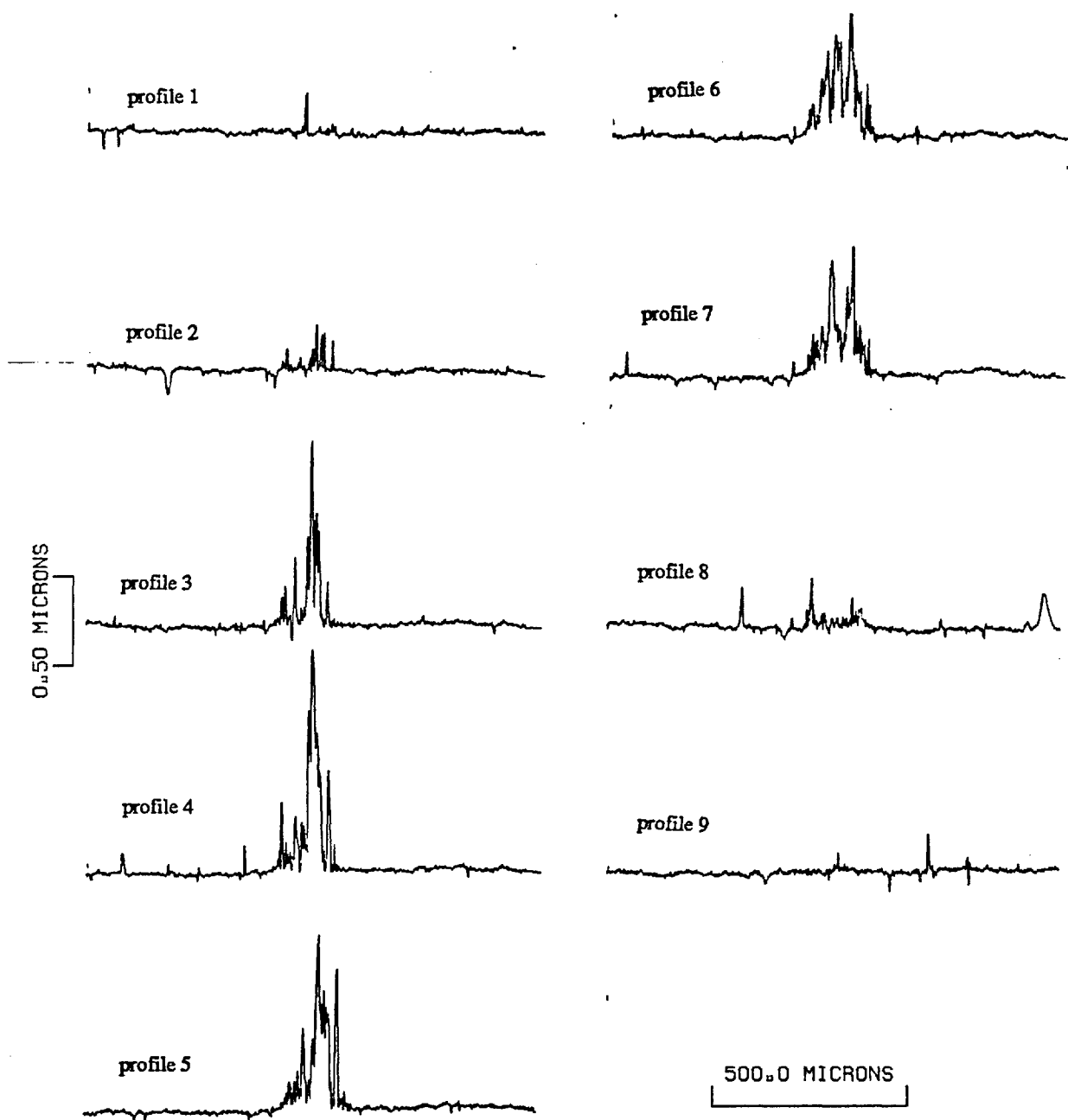


Figure 7.6 Surface profiles over the wear scar shown in Figure 7.5. Each profile is 20 μm apart, measured perpendicular to the direction of sliding. The relative position of each profile is indicated.

$\beta^*=9\mu\text{m}$. The results show that a transitional behaviour also existed but at a longer sliding distance. The overall surface damage and friction were less than that observed with the previous surface.

7.3.2.2 Smooth surfaces

Figure 7.8 shows the friction traces of a surface with $\sigma=0.048\mu\text{m}$ and $\beta^*=2.4\mu\text{m}$. The transitional behaviour occurred at $T=150^\circ\text{C}$. At $T=50^\circ\text{C}$ or 100°C the frictional force remained low and steady. No distinguishable scratching was found on the wear track, which is shown in Figure 7.9, but the track showed ^amark which was apparently produced by plastic deformation. Very little material was found to have adhered to the ball at these temperatures. However, at $T=150^\circ\text{C}$ significant scratching and adhesive transfer were observed and the surface damage was severe (as shown in Figure 7.10).

7.3.2.3 Very smooth surfaces

Figures 7.11a, b and c show the friction traces of a surface with $\sigma=0.048\mu\text{m}$ and $\beta^*=24\mu\text{m}$. Friction went up almost instantaneously and increased with temperature. The friction curves appeared to fluctuate at a regular wavelength. This was consistent with the repetitive stick-slip pattern found on the wear track. A typical wear track is shown in Figure 7.12(b). The wavelength of the repetitive patterns were approximately 80mm, 100mm and 120mm at $T=50^\circ\text{C}$, 100°C and 150°C respectively. Figure 7.13 shows the surface profile of the wear track at $T=100^\circ\text{C}$. The profile revealed a repetitive pattern with a wavelength of approximately 100 μm . The original profile was completely obliterated.

Adhered material was found on the ball at all temperatures. A typical micrograph of

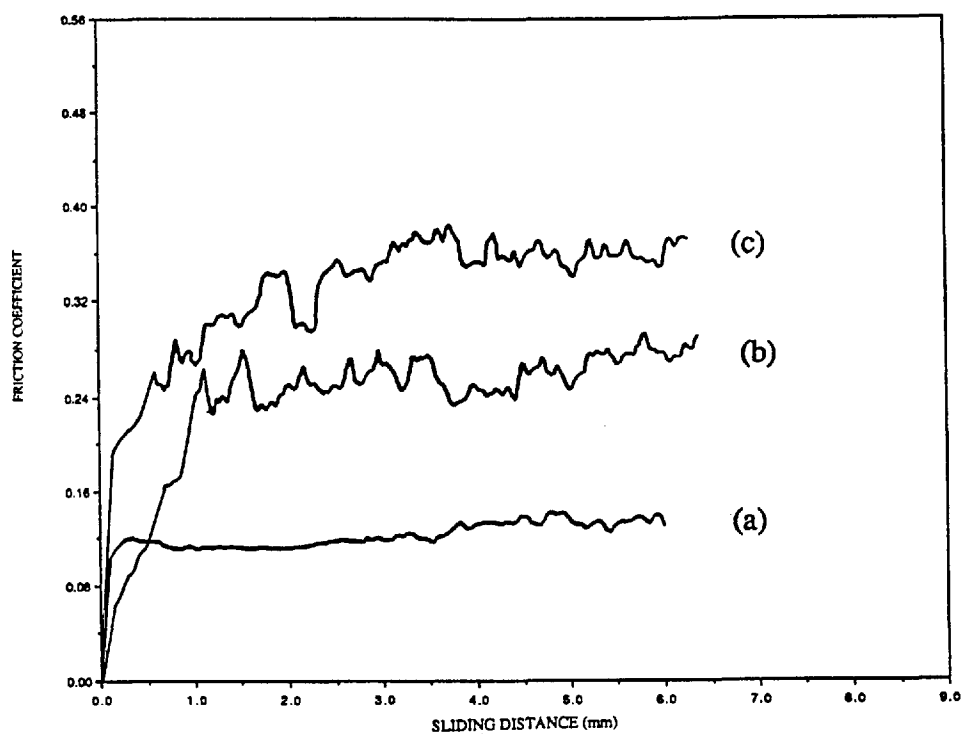


Figure 7.7 Friction traces of a steel ball on a less rough stainless steel surface (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

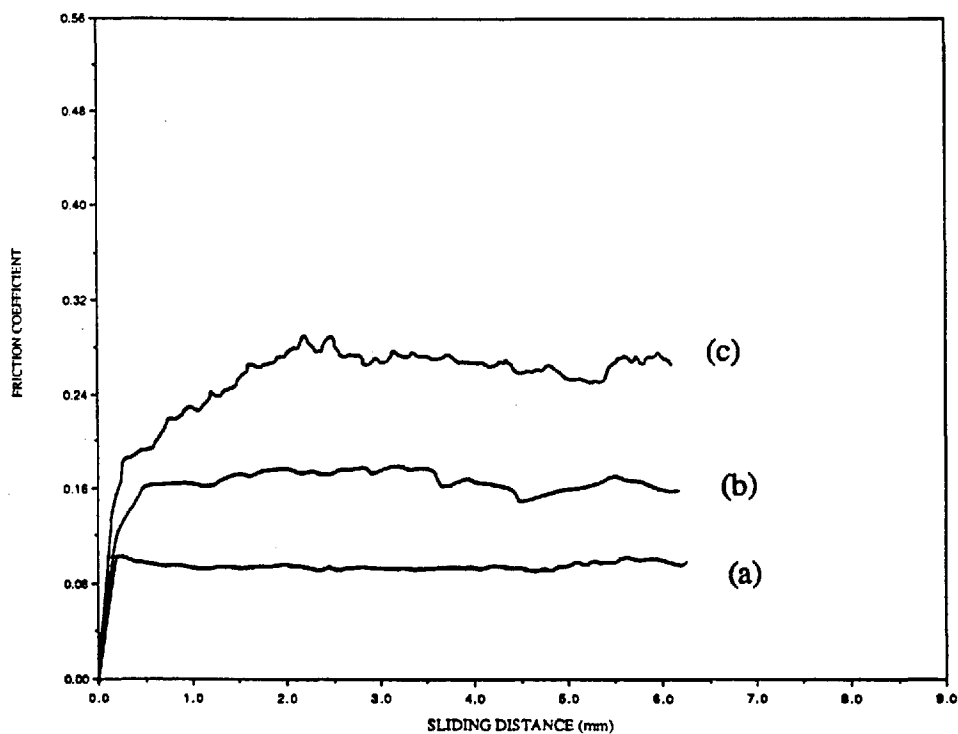


Figure 7.8 Friction traces of a steel ball on a smooth stainless steel surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$

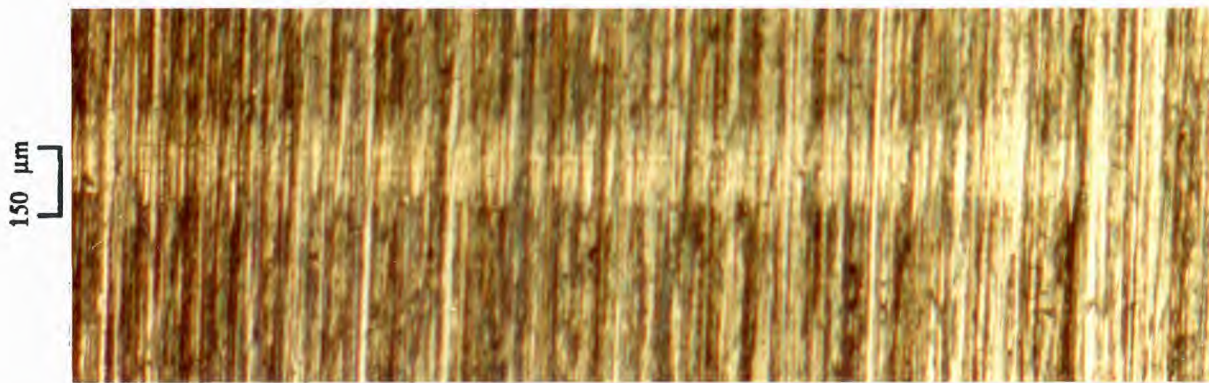


Figure 7.9 Wear track on the countersurface at $T = 100\text{ }^{\circ}\text{C}$, showing an apparent mark of plastic deformation.

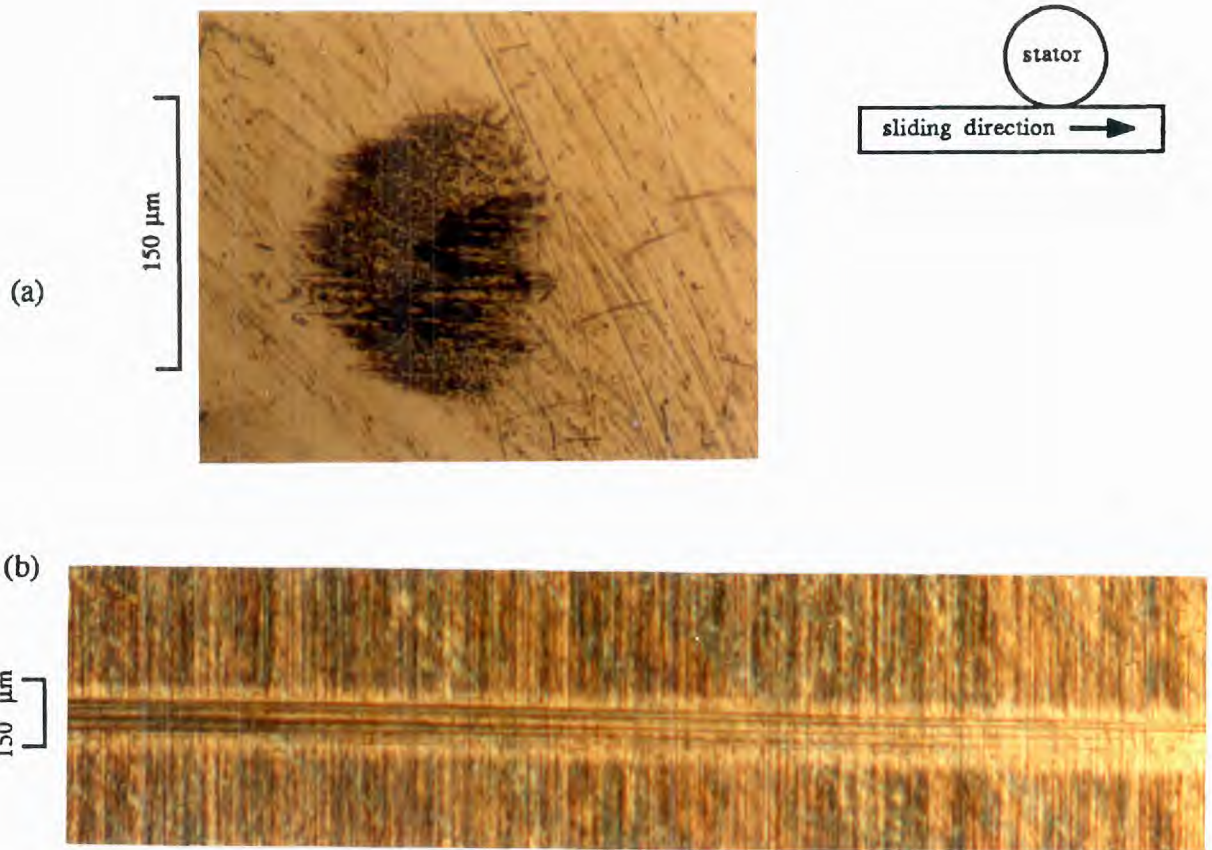


Figure 7.10 (a) Wear scar on ball at $T=150\text{ }^{\circ}\text{C}$
(b) Corresponding wear track on the countersurface.

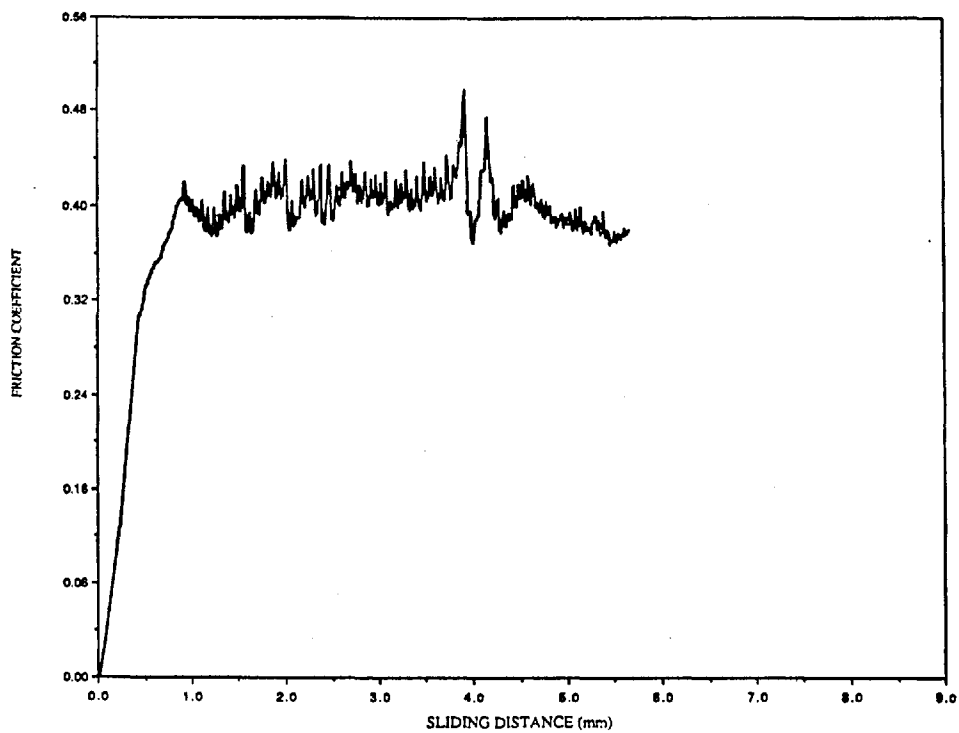


Figure 7.11a Friction trace of a steel ball on a very smooth stainless steel surface , $T = 50^{\circ}\text{C}$.

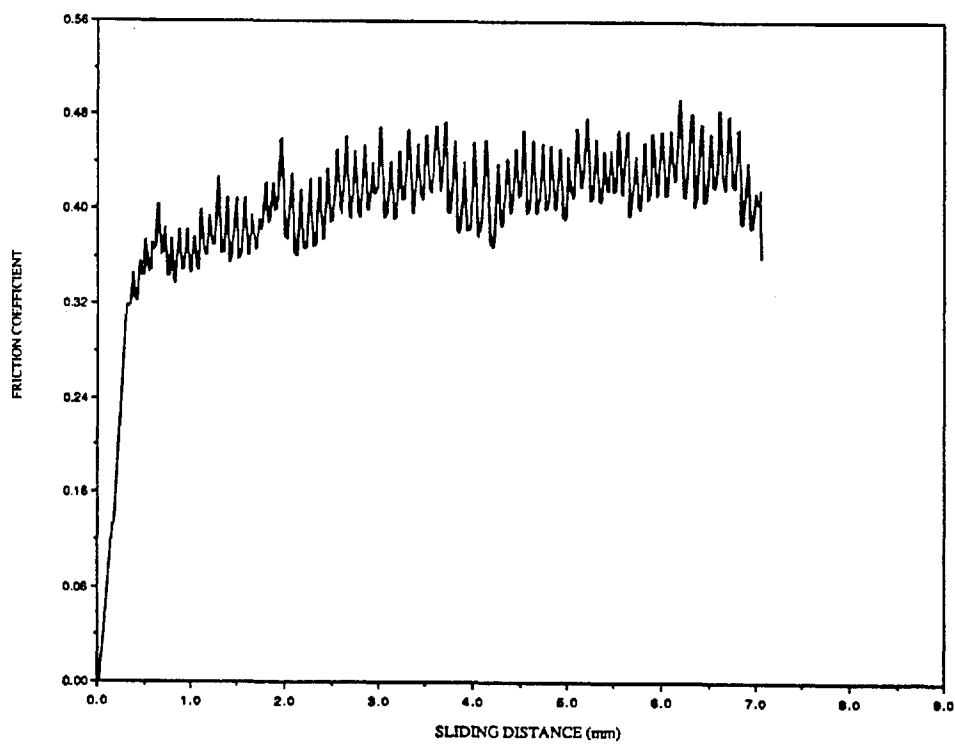


Figure 7.11b As Figure 7.11a but with $T = 100^{\circ}\text{C}$.

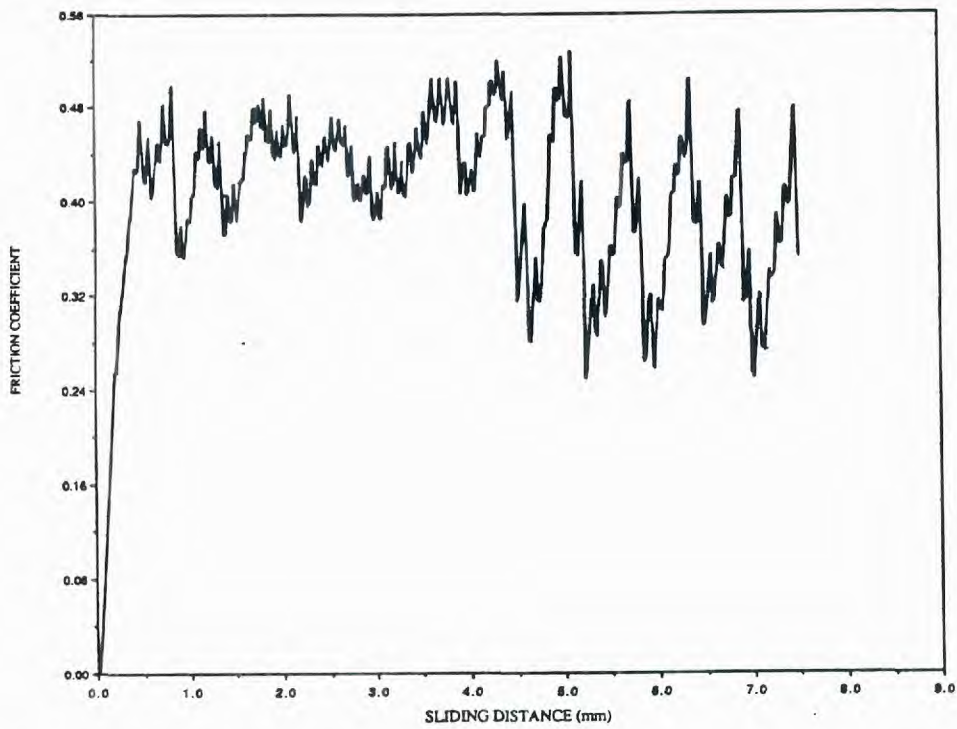


Figure 7.11c As Figure 7.11a but with $T=150^{\circ}\text{C}$.

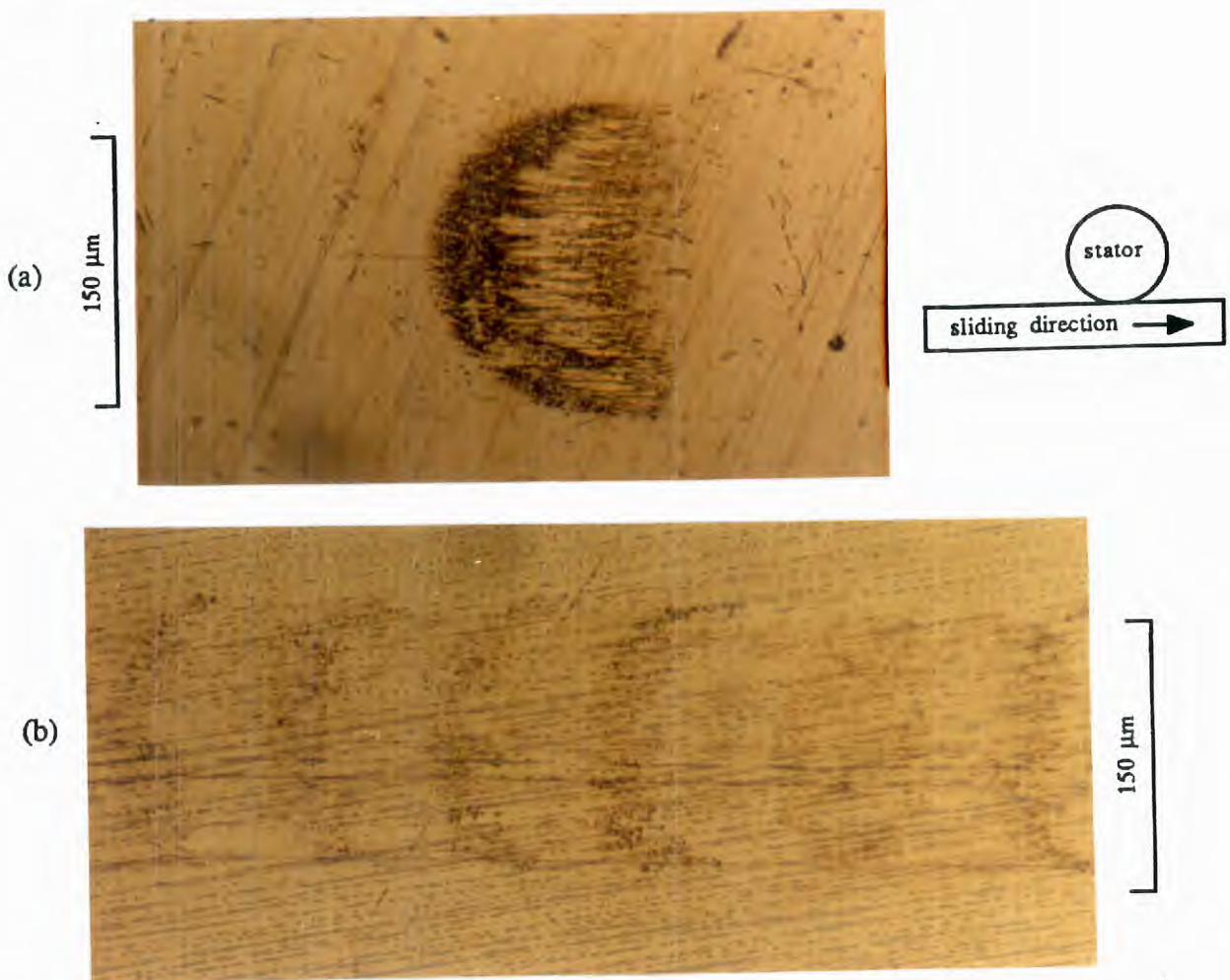


Figure 7.12 (a) Wear scar on ball at $T = 100^{\circ}\text{C}$.
(b) Corresponding wear track on the counter-surface, showing stick-slip pattern.

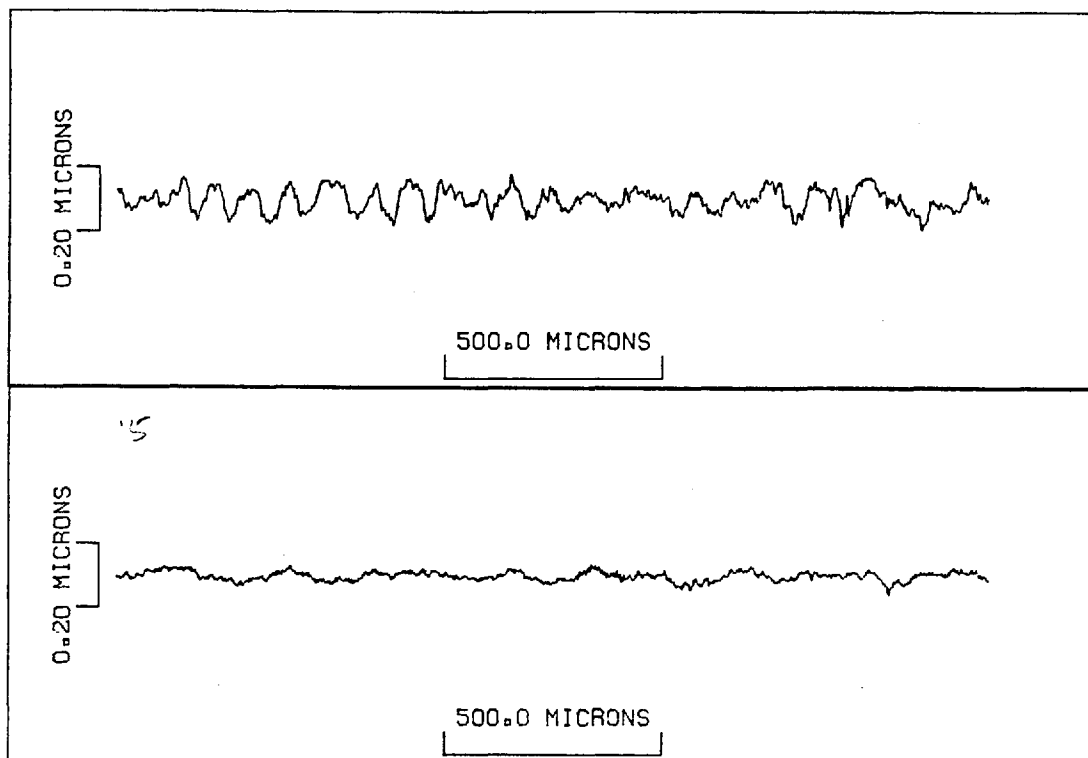


Figure 7.13 Surface profile of wear track shown in Figure 7.12(b). Upper: after sliding, Lower: before sliding.

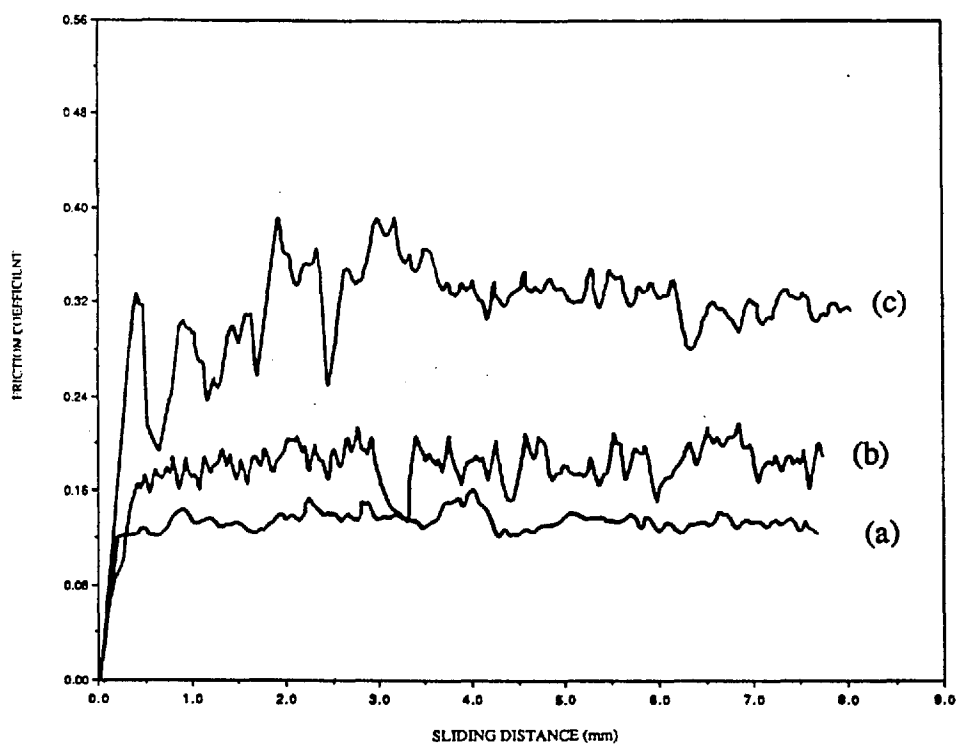


Figure 7.14 Friction traces of a steel ball on a rough gear steel surface. (a) $T = 50\text{ C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

the wear scar is shown in Figure 7.12(a). Talysurf measurements were made on the wear scar and the surface topographies (not shown) revealed that the amount of adhered material ahead of the slider was greater than that at the central region. This suggested the stick-slip patterns were initiated ahead of the ball.

7.3.2.4 Discussion

The study of the surface contact of two bodies has shown that there are three types of contact related to the surface roughness. For a rough surface, the contact will be predominantly plastic. For a smooth surface, the contact will be predominantly elastic. For a very smooth surface, the contact will also be elastic and the true area of contact will be equal or close to the nominal area, i.e. little or no part of the nominal contact area remains out of contact. In the friction tests described above, it was shown that there are three types of sliding behaviour and that the consequence surface damage can be related to the surface roughness.

It is well known that a metal surface is covered with a layer of contaminant film (Bowden & Tabor, 1950) even when the surface is carefully cleaned prior to operation under normal atmospheric conditions. Extremely small amounts of contaminants can greatly reduce the adhesion between metals. Under mild sliding conditions, two contacting bodies are separated by the contaminant films which results in low friction and negligible surface damage. A surface with an intermediate roughness provides such a condition for 'safe sliding'. The contact is predominantly elastic, which means that the possibility of contaminant films being penetrated to allow metal-metal contact is minimised. Even if a local adhesive junction is formed, the spacing between contacts will break the junction and prevent it from growing.

For a rough surface, the plastic deformation causes the rupture of the contaminant films which permits contact of the underlying metal. The increase in shear force, due to the formation of metal-metal junctions would in turn cause the asperity to collapse and the junction to grow. Thus the breakage of a much larger contact than that predicted by simple normal stresses is necessary. This is consistent with Tabor's junction growth theory (Tabor 1959). The claim that friction is originated from breaking the adhesion junctions between the interface is consistent with the experimental results of the friction tests, since a lump of material that was identified as coming from the countersurface was seen to be adhered securely to the ball.

For a very smooth surface, the contact area is high and there is little spacing to trap the lubricant, which may result in 'lubricant starvation'. The sliding process will also facilitate the breakdown of the oxide film. In any subsequent sliding, fresh oxide film has little time to form and protect the underlying metal. This increases the likelihood of clean metal-metal contact occurring. The small welded junctions formed will also have a tendency to continue to grow due to lack of surface irregularities, until the energy stored up in the elastic deflection arm exceeds the strength of the interface. The excess energy in the slider then acts against the direction of translation resulting in a 'slip'. This action is consistent with the stick-slip pattern observed in the wear tracks. The friction increased with temperature, thus the slider would jump further due to the additional energy released and the wavelength of the stick-slip patterns would correspondingly increase with temperature.

7.3.3 Steel Ball on Gear Steel

7.3.3.1 Rough surface

Figure 7.14 shows the friction traces of a surface with $\sigma=0.3\text{ }\mu\text{m}$ and $\beta^*=4.2\text{ }\mu\text{m}$.

At lower temperatures the friction remained low. Figure 7.16 shows the surface topography before and after sliding, which indicates that very little surface damage took place. However the wear track shown in Figure 7.15, reveals that scuffing has occurred. This was shown to increase with temperature. The scuffing produced loose wear particles resulting in a 'third body' abrasion process. This is illustrated by the wear track and the corresponding wear scar on the ball in Figure 7.17. It can be seen that two distinct patterns of wear scar appeared on the ball. One was produced by direct contact with the countersurface. The other, observed as a long and dark patch outside of the nominal contact, was apparently produced when a loose wear particle was squeezed and adhered to the ball. Figure 7.18 shows a series of parallel surface profiles of the wear scar. It can be seen that some scratches appeared on both regions of the wear scar. The scratches were probably produced by 'three-body' abrasion. The wear particles would be expected to penetrate both surfaces to the same extent due to the same order of hardness of the materials. This would result in scratches on both surfaces, until they became free. The friction force can be concluded as coming from shearing the contacting interface and ploughing of the wear particles interlocked between the surfaces.

7.3.3.2 Smooth surfaces

Figure 7.19 shows the friction traces of a surface with $\sigma=0.095\text{ }\mu\text{m}$ and $\beta^*=3.6\text{ }\mu\text{m}$. The overall frictional behaviour and surface damage were similar to the previous surface.

7.3.3.3 Very smooth surfaces

Figures 7.20a, b and c show the friction traces of a surface with $\sigma=0.013\text{ }\mu\text{m}$ and

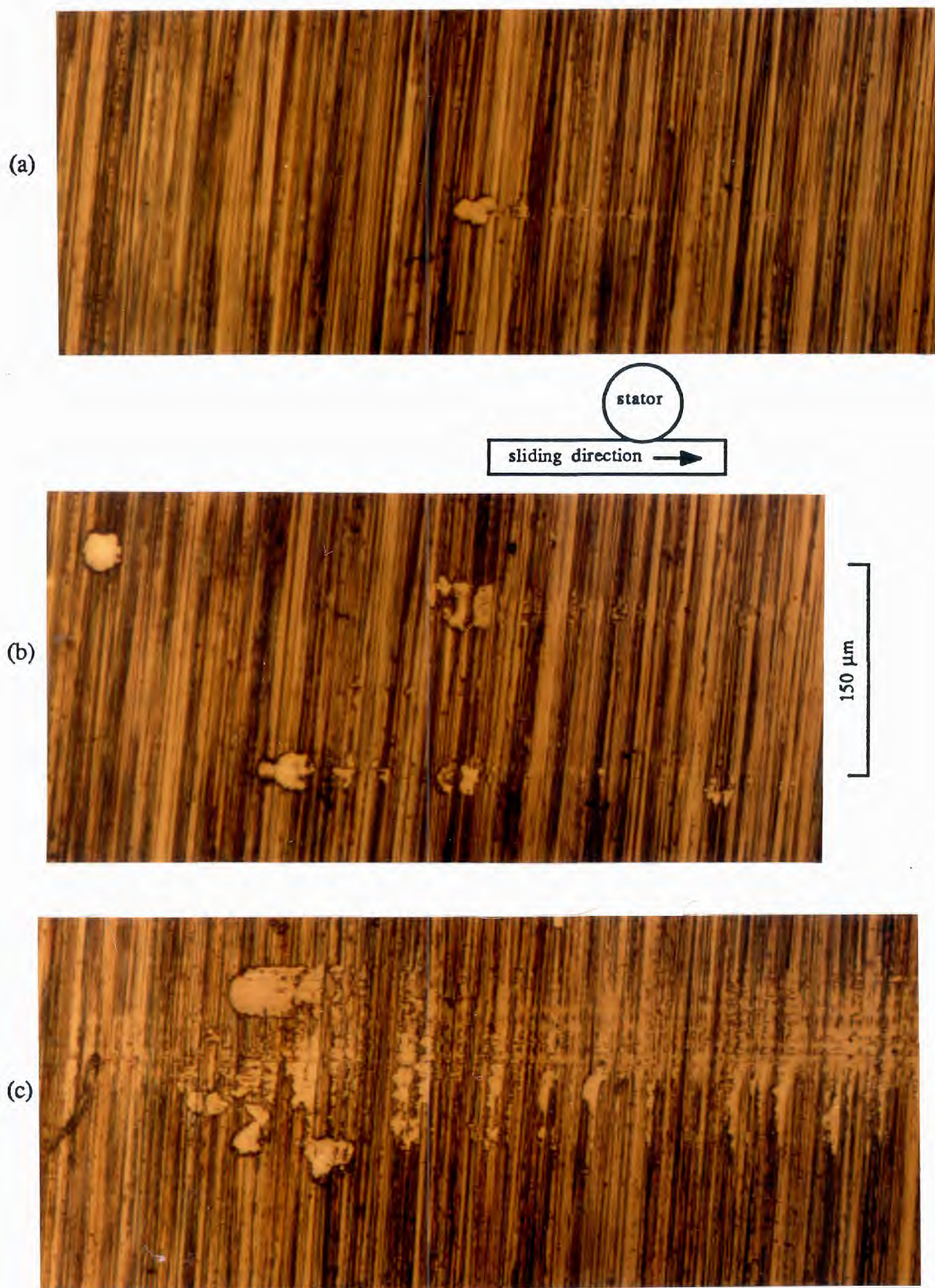


Figure 7.15 Micrographs of wear tracks, showing scuffing increases with temperature. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

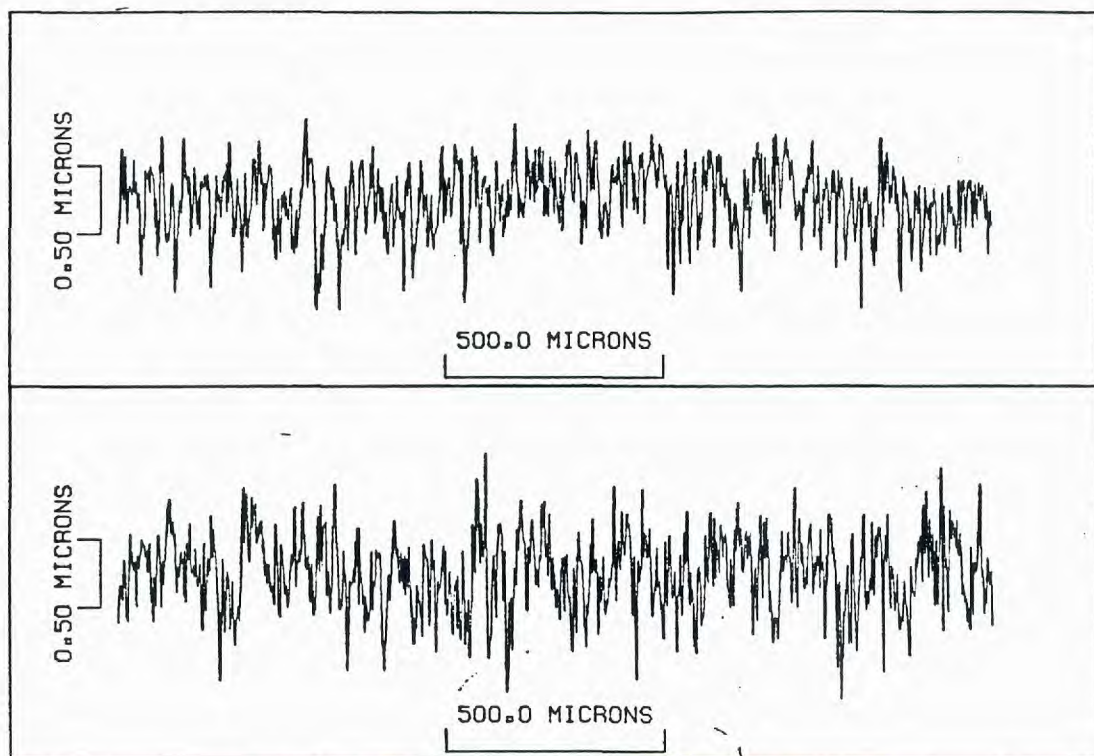


Figure 7.16 Surface profile of wear track of the surface shown in Figure 7.15(b). Lower: before sliding, Upper : after sliding.

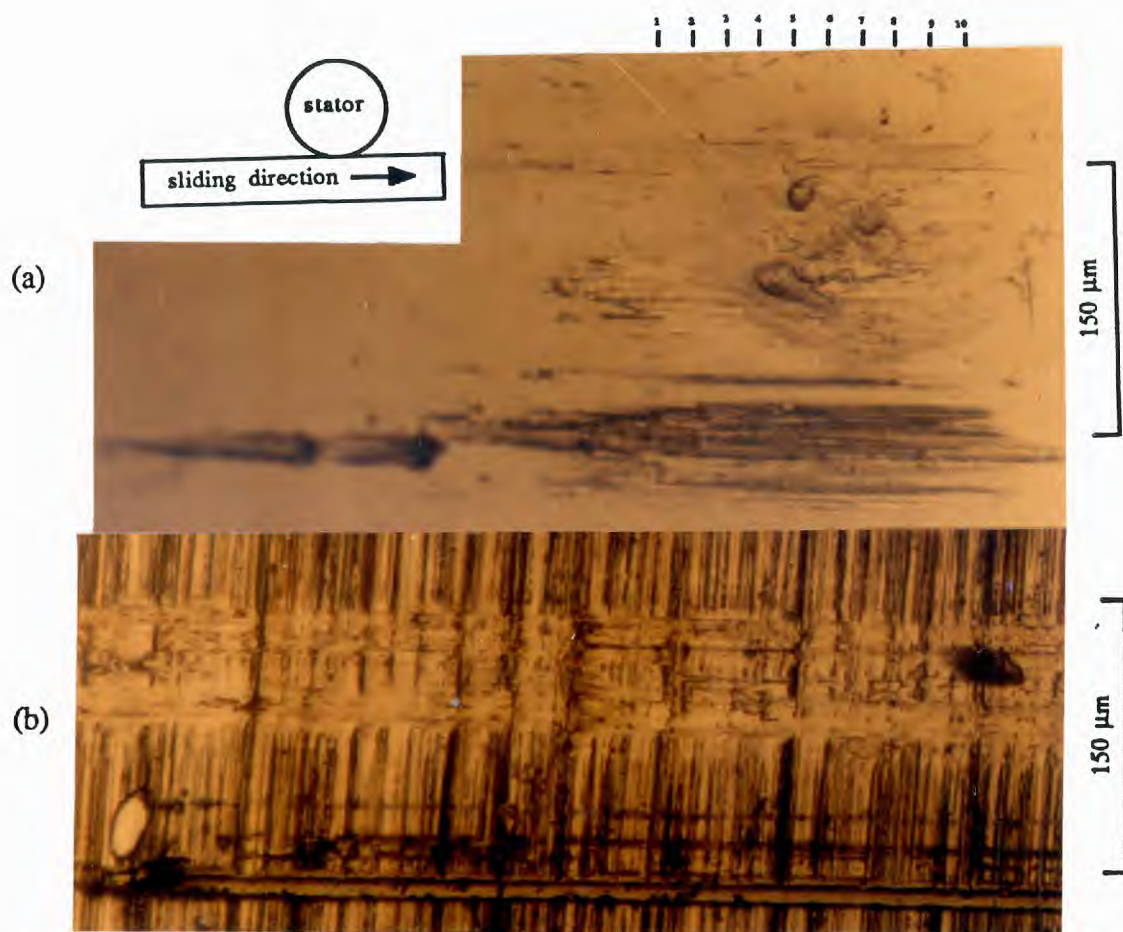


Figure 7.17 (a) Wear scar on ball after test at $T = 100^{\circ}\text{C}$.
(b) Corresponding wear track on the counter-surface near the end of sliding.

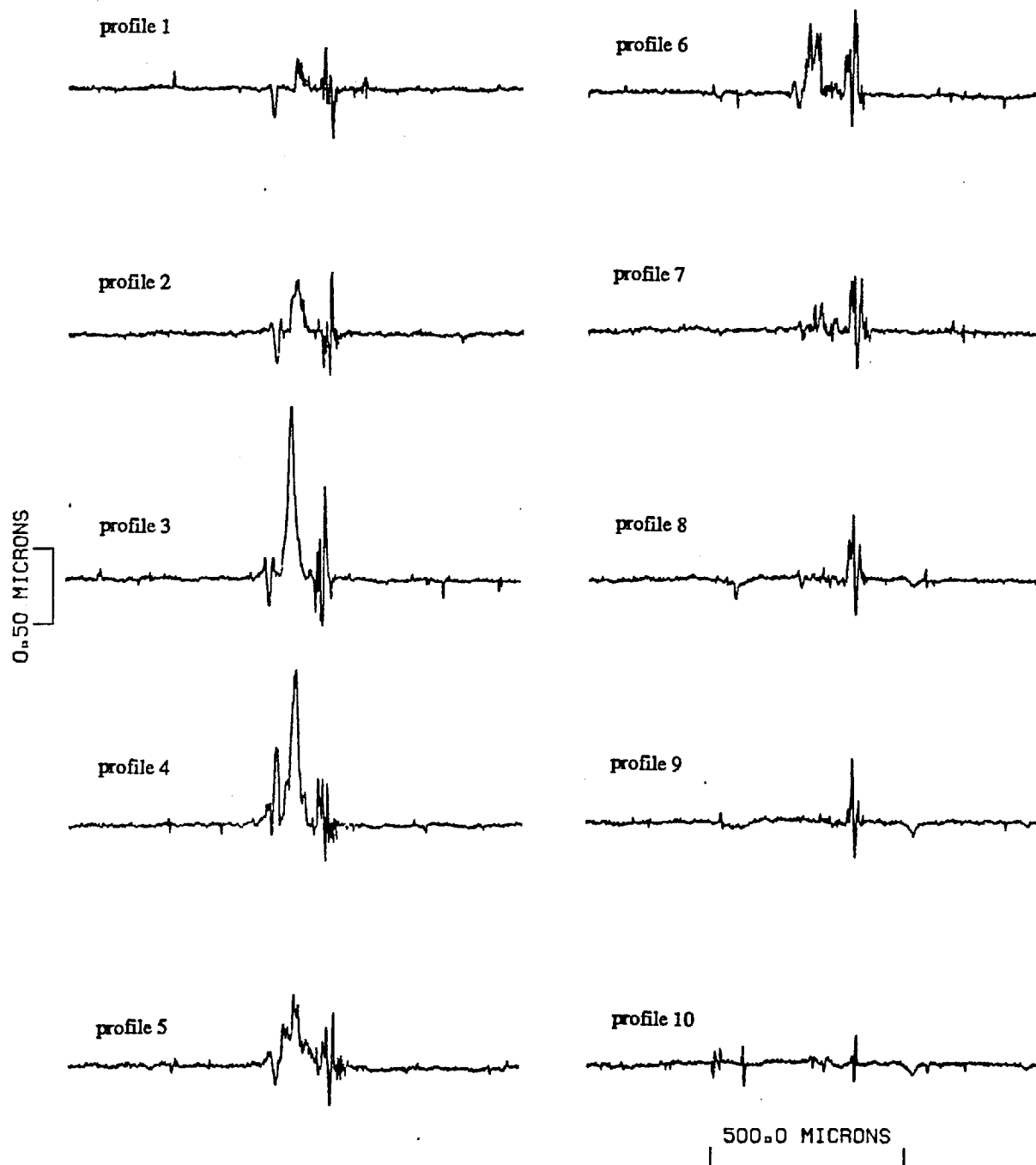


Figure 7.18 Surface profiles over the wear scar in Figure 7.17(a), which indicate the relative position of each profile. Each profile is 20 μm apart.

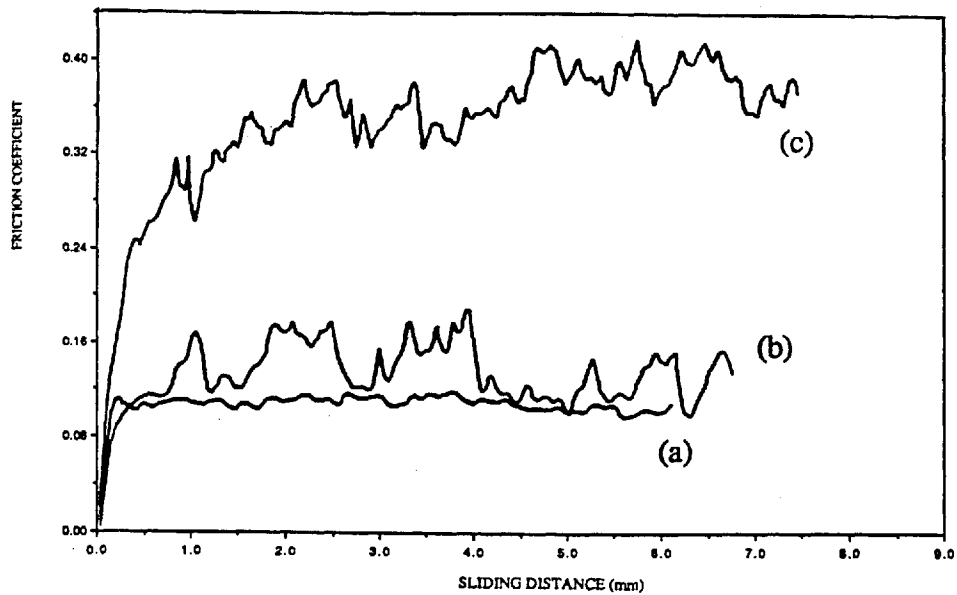


Figure 7.19 Friction traces of a steel ball on a smooth gear steel surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

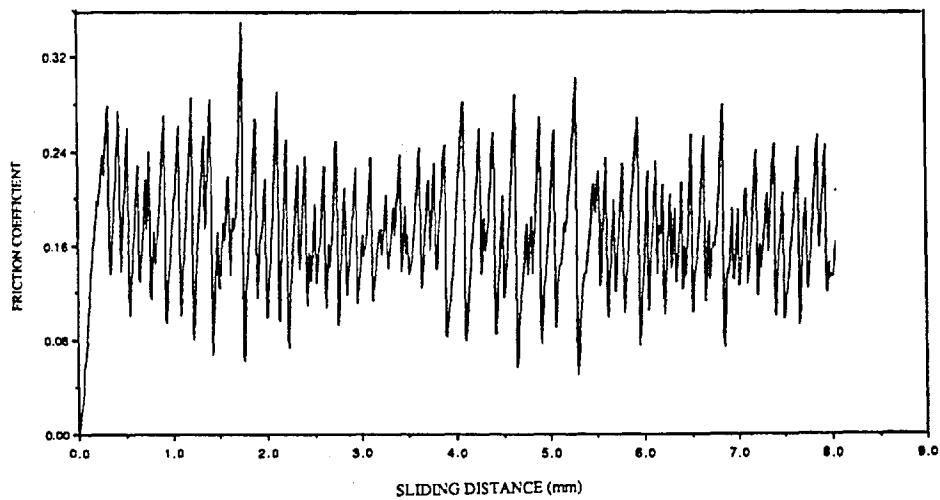


Figure 7.20a Friction traces of a steel ball on a very smooth gear steel surface, $T = 50^{\circ}\text{C}$.

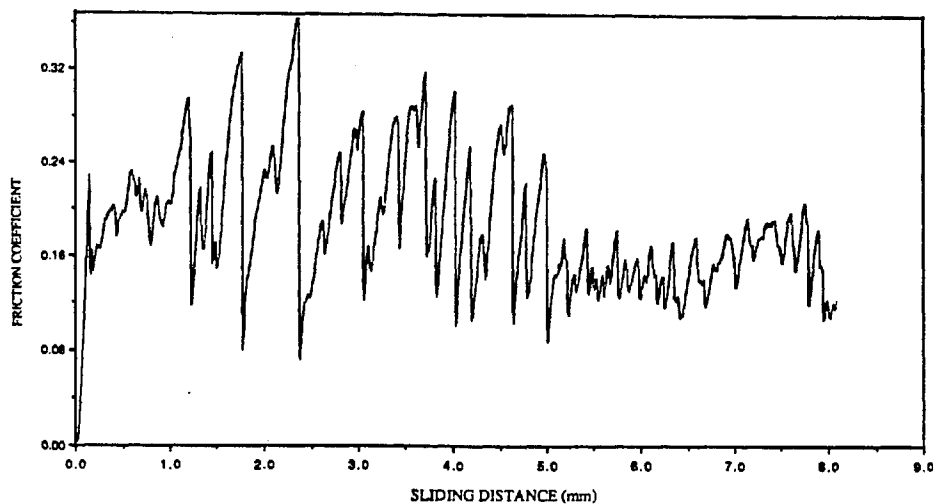


Figure 7.20b As Figure 7.20a but with $T = 100^{\circ}\text{C}$.

$\beta^* = 34.0 \mu\text{m}$. The traces show a similar stick-slip pattern to that observed with the steel ball sliding on a very smooth stainless steel surface. Figure 7.21(b) shows the wear track accompanied with a series of stick-slip patterns. Figure 7.21(a) shows the corresponding wear scar on the ball indicating the real contact area covered almost the entire region of nominal contact.

A different frictional behaviour was observed on a similar surface, the friction traces are shown in Figure 7.22. At $T=50^\circ\text{C}$, the friction remained steady but stayed at a relatively high level. The friction also increased slightly with the sliding distance. A few fine, parallel, thin scratches were found on the corresponding wear track, towards the end of sliding. This was apparently due to the third body particles participating in a 'micro-machining' process. This process continued to plough the surface due to a lack of interruptions in the surface which would have prevented further growth. At $T=100^\circ\text{C}$, the trace fluctuated with a stick slip pattern similar to that of the surface shown in Figure 7.21a. At $T=150^\circ\text{C}$, the overall frictional level increased, although no regular pattern was observed.

7.3.3.4 Discussion

The friction behaviour observed with a steel ball sliding on rough and smooth gear steel surfaces was found to be similar. This resulted from the elastic mode of deformation on both surfaces. The sliding friction and surface damage observed were generally small. However, third body abrasion was evident during the sliding process.

For a very smooth surface, the friction and surface damage were both high. This could be explained, as before, by the conformity of the two bodies in contact. However, there were two types of frictional behaviour detected on two such surfaces. One

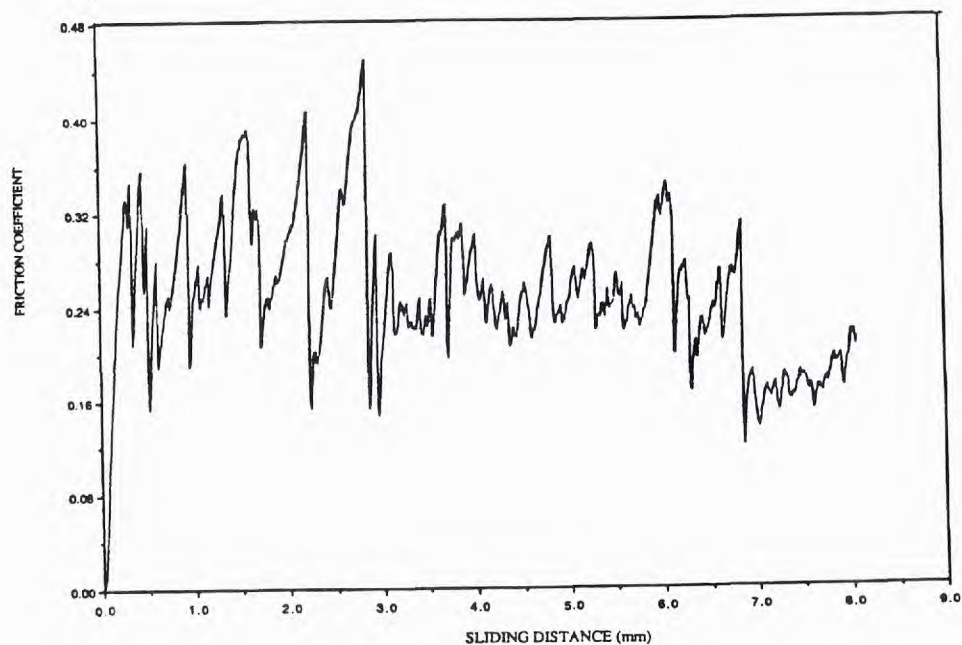


Figure 7.20c As Fig. 4.19a but with $T = 150^{\circ}\text{C}$.



Figure 7.21 (a) Wear scar on ball at $T = 100^{\circ}\text{C}$.
(b) Corresponding wear track on the counter-surface.

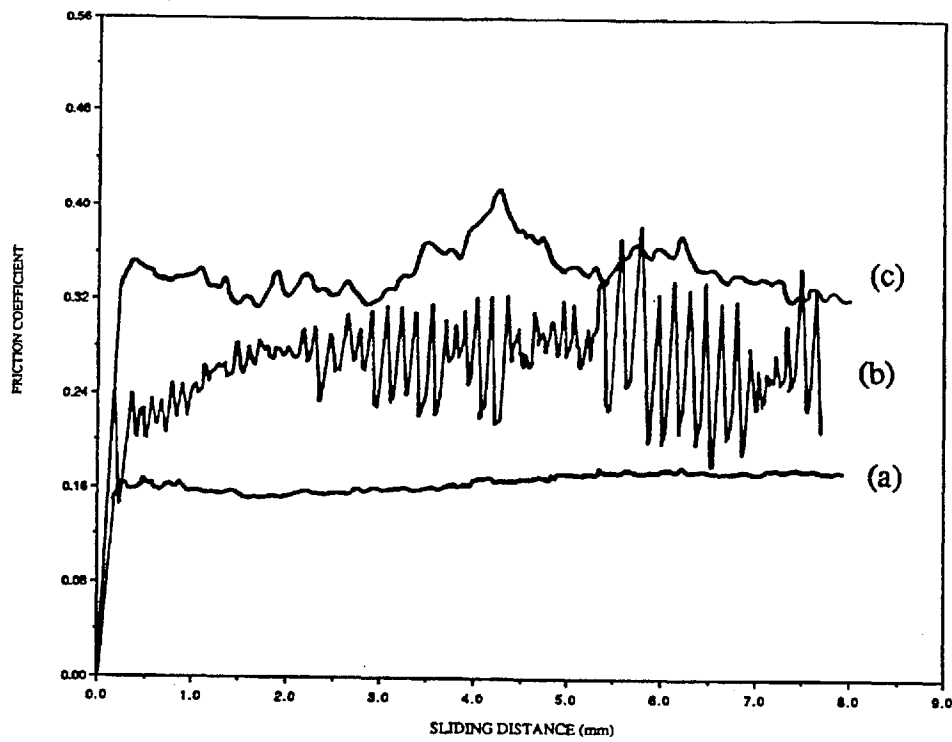


Figure 7.22 Friction traces of a steel ball on a similar surface as that in Figure 7.20 (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

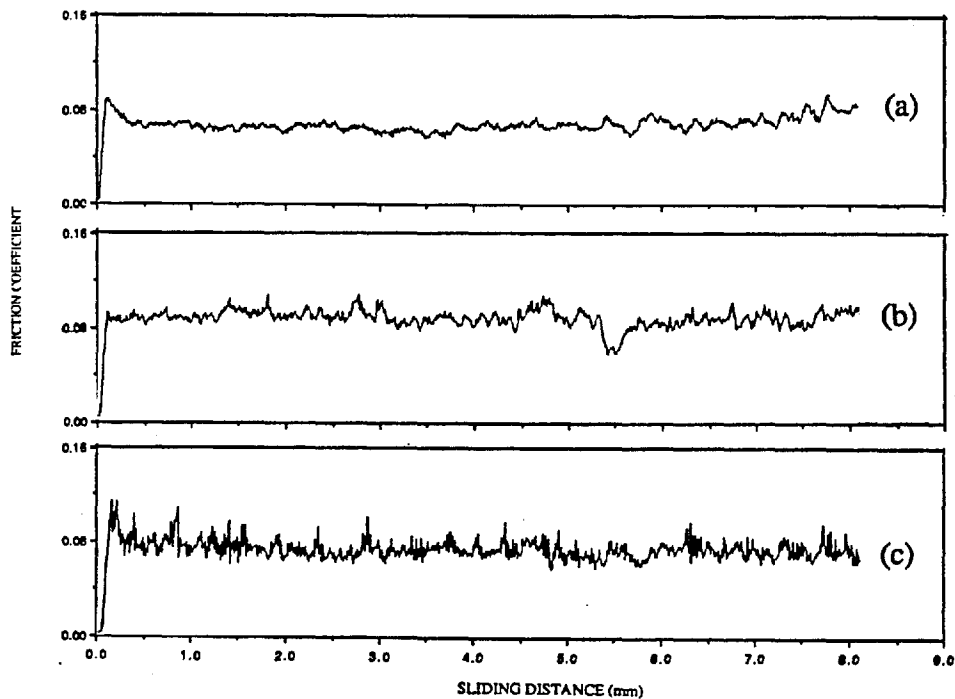


Figure 7.23 Friction traces of a steel ball on a rough copper surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$

demonstrated a stick-slip friction trace, the corresponding wear scar showed a similar stick-slip pattern. The stick-slip motion apparently resulted from breakage of the adhesive junctions formed at the interface. The other shows a steady, but high friction trace. The corresponding wear scar consisted of a series of continuing scratches. The scratches appeared to be produced by ploughing of the wear particles. Nonetheless, both surfaces have high friction for the same reason: there are no interruptions in the interface to prevent the growth of a junction once it is formed.

7.3.4 Steel Ball on Copper

7.3.4.1 Rough surfaces

Figure 7.23 shows the friction traces for a surface with $\sigma=0.5\mu\text{m}$ and $\beta^*=7.5\mu\text{m}$. The friction remained low and relatively steady at low temperature. At higher temperature ($T=150^\circ\text{C}$), the friction traces fluctuated, with a much shorter wavelength, but the mean friction remained about the same as that observed at lower temperature. Figure 7.24 shows the surface profile along the wear track at $T=100^\circ\text{C}$ indicating that the peaks were undergoing plastic deformation, but the valleys remained unaltered. The corresponding wear track was consistent with plastic deformation, but no significant surface seizure. At $T=150^\circ\text{C}$, the wear track was wider and showed a larger area obliterated by plastic deformation, but no distinguishable abrasive lines were evident.

7.3.4.2 Smooth surfaces

Figure 7.25 shows the friction traces of a surface with $\sigma=0.08\mu\text{m}$ and $\beta^*=5.2\mu\text{m}$. The frictional behaviour and surface damage were similar to that of the rough surface. The friction remained low at all temperatures and the surface damage observed was not

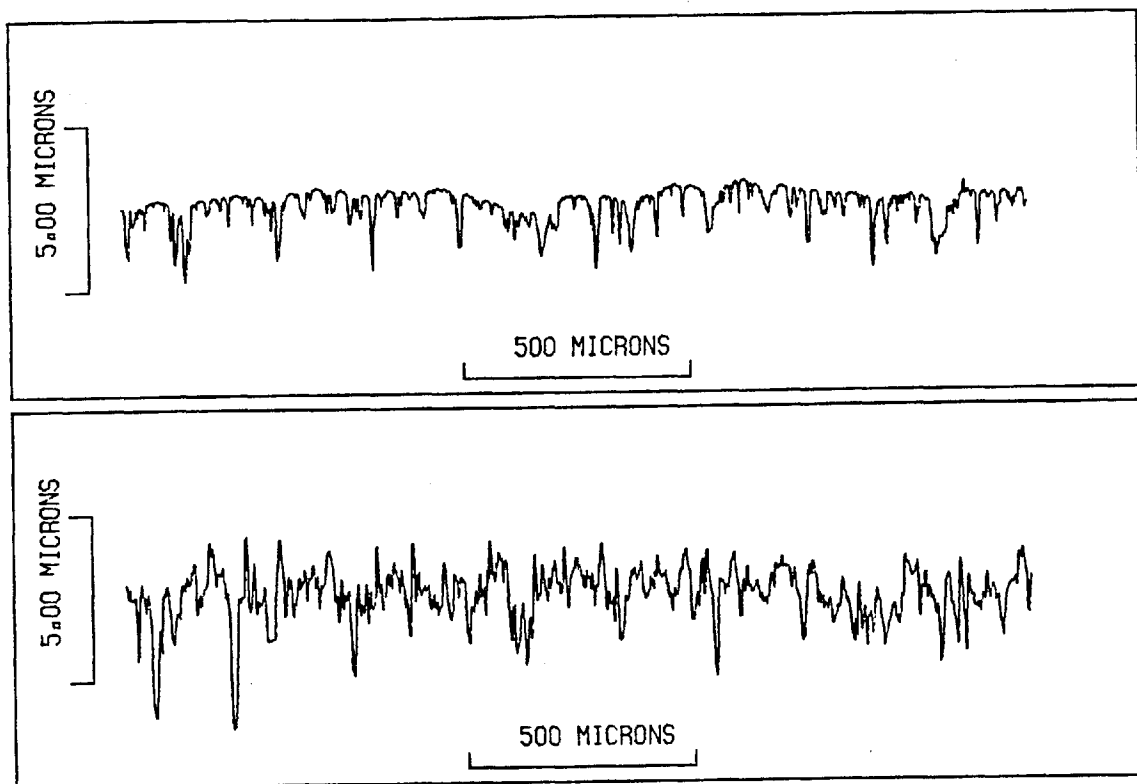


Figure 7.24 Surface profile of the wear track.
Lower: before sliding, Upper: after sliding

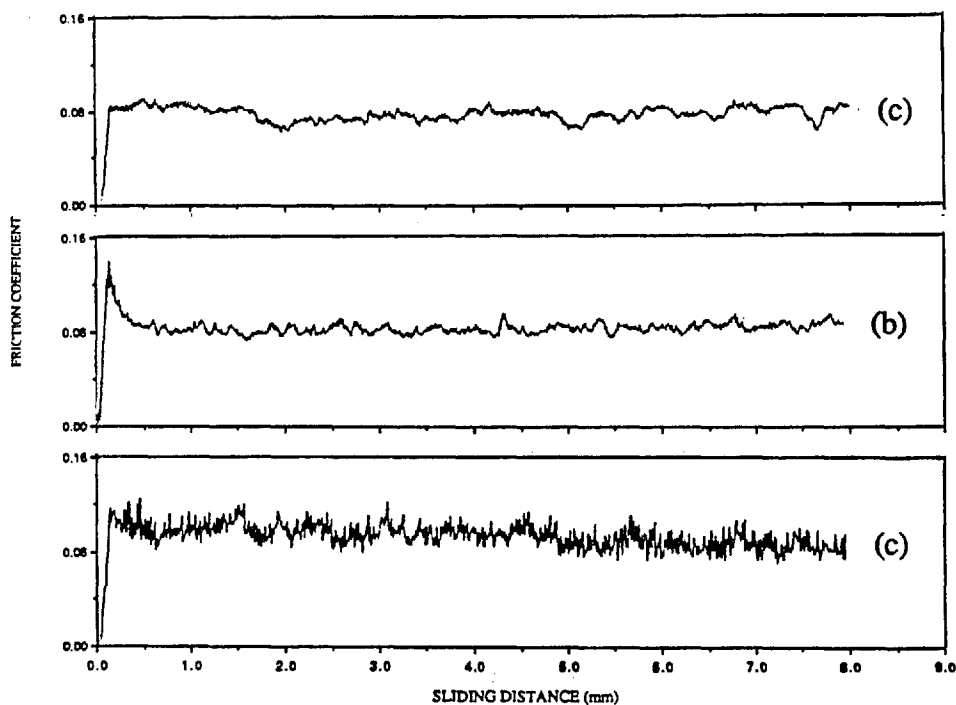


Figure 7.25 Friction traces of a steel ball on a smooth copper surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

significant.

7.3.4.3 Very smooth surface

Figure 7.26 shows the friction traces of a surface with $\sigma=0.02\mu\text{m}$ and $\beta^*=13.6\mu\text{m}$. The friction remained low at all temperatures. The surface damage on the wear track at $T=50^\circ\text{C}$ and 100°C was not significant. At $T=150^\circ\text{C}$, the traces fluctuated at a much shorter wavelength, similarly to that observed with the rough surfaces.

7.3.4.4 Discussion

The effect of roughening the surface of copper does not greatly change the frictional behaviour, although the mode of deformation is plastic. It is well known that copper is a reactive metal which means that a layer of oxide film is readily formed under atmospheric conditions. The layer of oxide film may grow with the underlying metal. The growth process often involves the diffusion of metal ions through the oxide film. On certain crystal faces the growth may be far more rapid than on others. Therefore, the thickness of surface film will be different for various materials. Copper is more reactive than stainless steel and gear steel. The oxide film on copper is likely to be thicker than that of stainless steel and gear steel. The oxide film will then react with a lubricant such as paraffin with 1% stearic acid to form a layer of metal-soap and thus provide an effective boundary lubricant. Although the contact of copper is mostly plastic, the boundary lubrication is effective in protecting the underlying metal from seizure.

7.3.5 Ceramic Ball on Stainless Steel

7.3.5.1 Rough surfaces

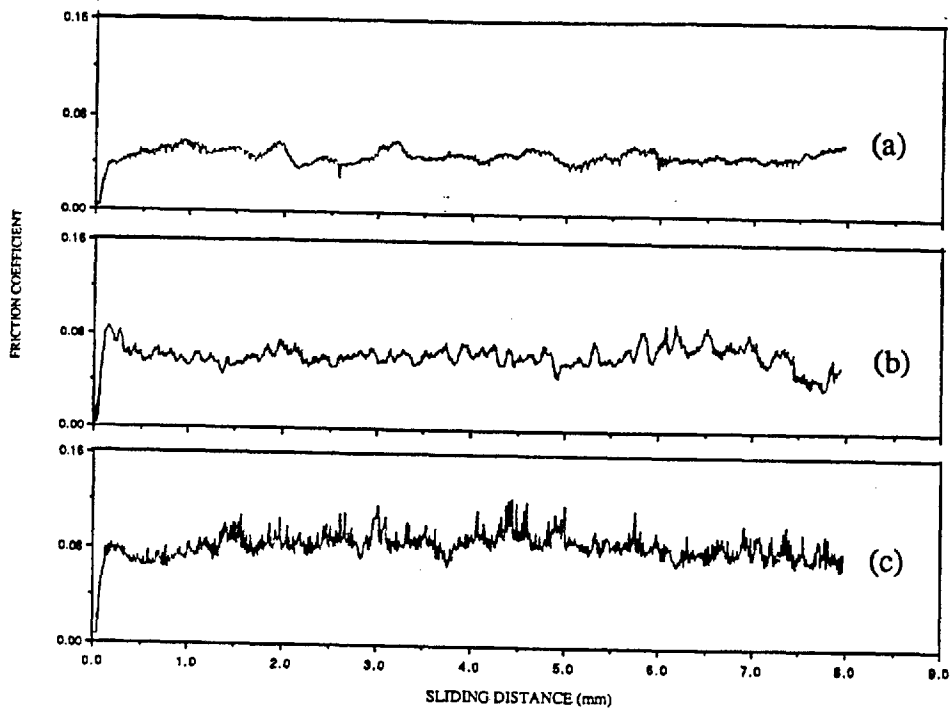


Figure 7.26 Friction traces of a steel ball on a very smooth cooper surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

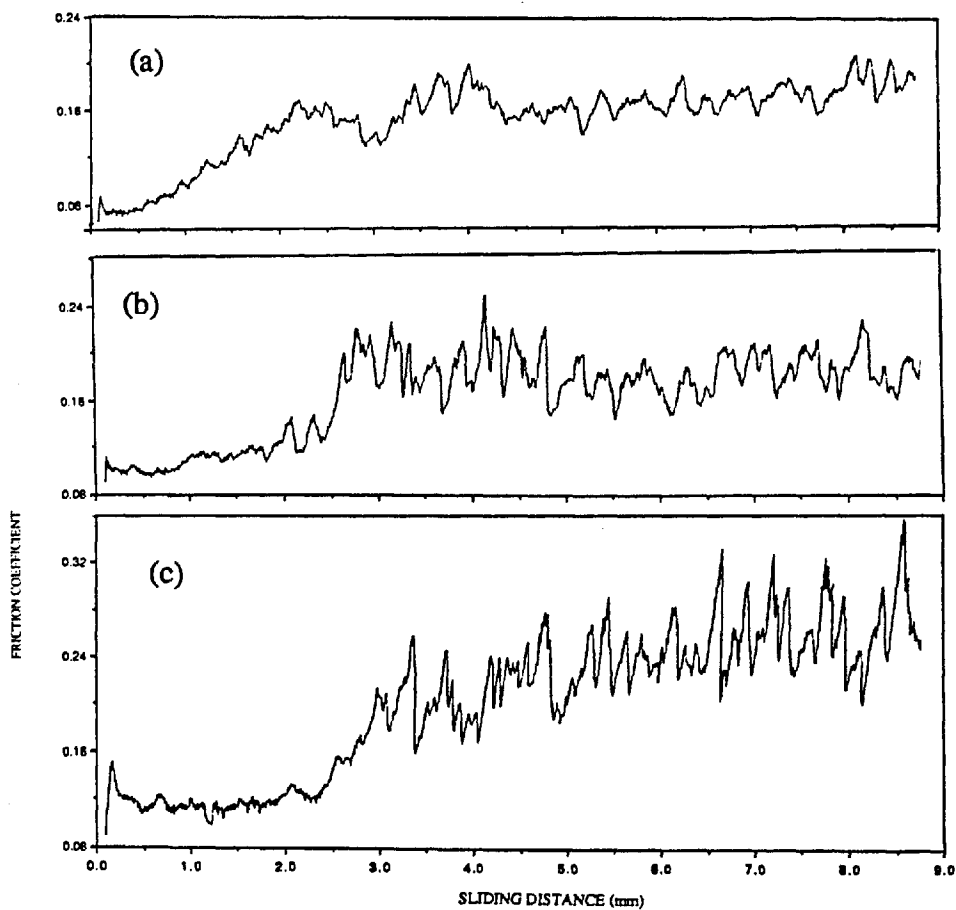


Figure 7.27 Friction traces of a ceramic ball on a rough stainless steel surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$

Figure 7.27 shows the friction traces of a surface with $\sigma=0.2\mu\text{m}$ and $\beta^*=3.5\mu\text{m}$. The friction traces show transitional behaviour from smooth friction to high and variable friction. The general frictional behaviour was similar to that of a steel ball sliding on a rough surface of stainless steel. Adhesive transfer was found on the ball in the contacting region. This was identified by the EDAX facility on the SEM as coming from the countersurface. This leads to two possible suggestions. Firstly, the adhesive junction formed in the ceramic-stainless steel interface may have been stronger than the shear strength of the bulk material, so that the break occurred away from the interface. Secondly, wear particles may have been formed in the sliding process and remained trapped in the interface, thereby taking part in three body abrasion. In both cases, the contact interface may therefore be regarded as stainless-steel on stainless steel. The friction behaviour would therefore be expected to resemble that of the steel ball on stainless steel flat. At the end of the sliding, the highly variable trace may be the result of the contact of stainless steel on stainless steel.

7.3.5.2 Smooth surfaces

Figure 7.28 shows the friction traces of a surface with $\sigma=0.084\mu\text{m}$ and $\beta^*=3.64\mu\text{m}$. It is interesting to note that the friction dropped slightly as the temperature increased. In the wear track, only the ridges were found to be deformed plastically. No abrasive lines or adhesive transfer was detected at any of the temperatures examined.

7.3.5.3 Very smooth surfaces

Figure 7.29 shows the friction traces of a surface with $\sigma=0.017\mu\text{m}$ and $\beta^*=2.2\mu\text{m}$. At $T=25^\circ\text{C}$, the friction went up sharply and remained high towards the end of the sliding.

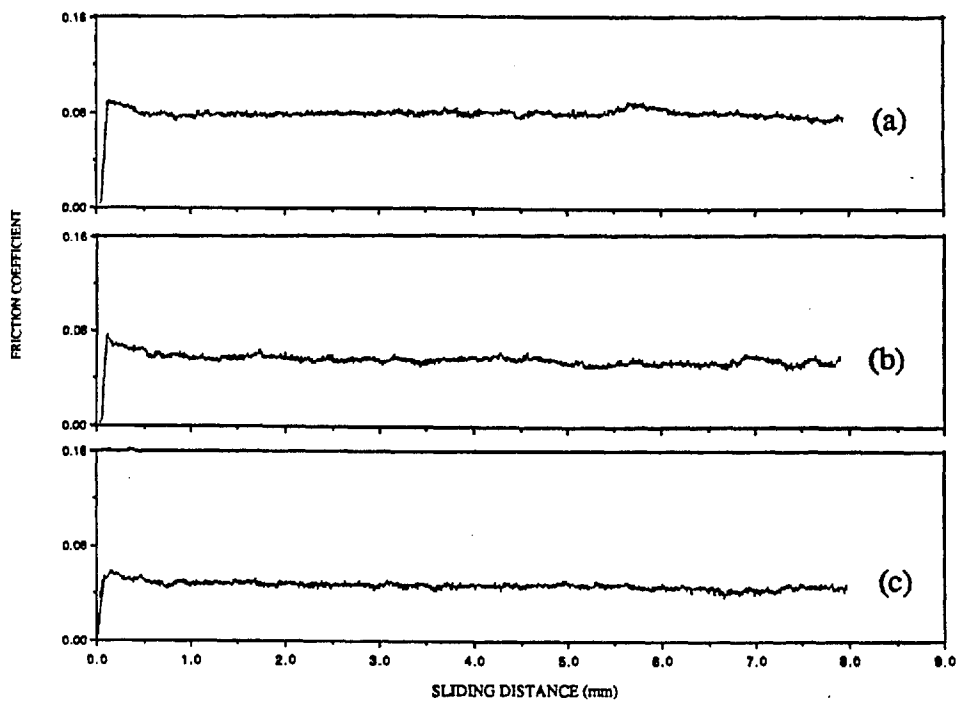


Figure 7.28 Friction traces of a ceramic ball on a smooth stainless steel surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

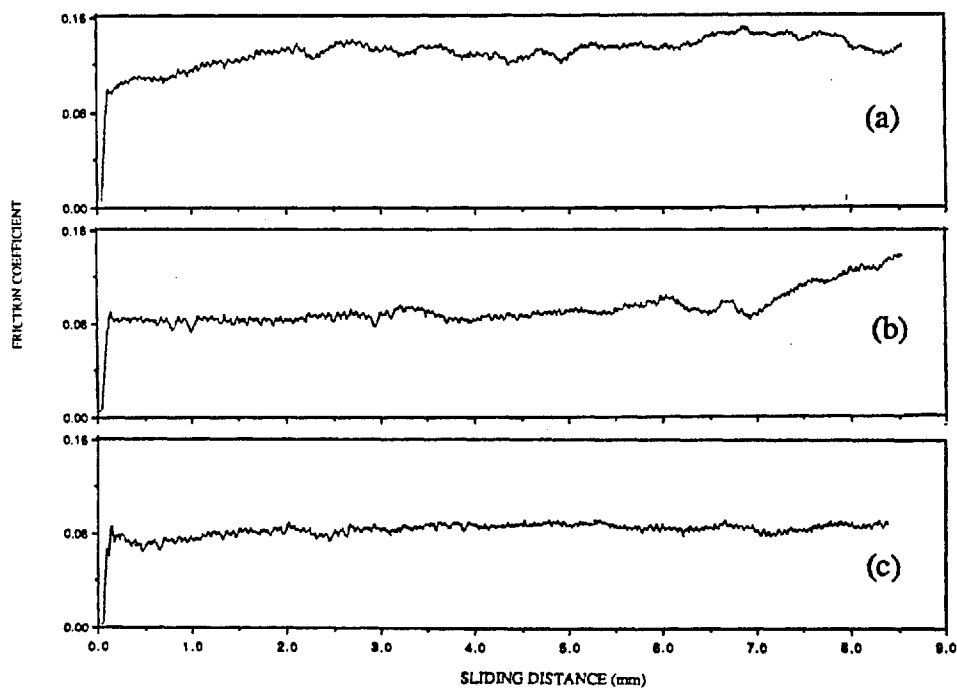


Figure 7.29 Friction traces of a ceramic ball on a very smooth stainless steel surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

At $T=100^{\circ}\text{C}$, it remained low up to a sliding distance of about 7mm after which it rose steadily. At $T=150^{\circ}\text{C}$, it remained low throughout the sliding.

It is interesting to note that the sliding behaviour at increased temperature is different from that observed with the steel ball on stainless steel. In the latter case the sliding friction and surface damage increased with temperature. In the case of a ceramic ball on stainless steel, the friction and surface damage decreased as the temperature increased. Several surfaces with similar finishes have been tested to check the consistency of the results. A similar trend was observed. However, one surface was found to behave a little differently from the other surfaces. This is shown in Figure 7.30. The friction observed at $T=100^{\circ}\text{C}$ was lower than that at $T=25^{\circ}\text{C}$. However, at $T=150^{\circ}\text{C}$ the friction increased to about the level at $T=25^{\circ}\text{C}$, and remained high towards the end of sliding.

7.3.5.4 Discussion

In general, there are three types of friction behaviour depending on the surface roughness, as in the case of a steel ball sliding on a stainless steel flat. The effect of increasing temperature does not necessarily result in increased friction and surface damage. In fact, the observed friction behaviour shows that in the general case the friction decreases as the temperature increases. If the friction force is assumed to break the adhesive junction formed in the interface, then the adhesion between ceramic and stainless steel must decrease with increased temperature. If this is true, it can help to explain the unexpected friction behaviour of a very smooth surface at an increased temperature.

There are two possible contact-types present at the interface. The first is the contact of ceramic on stainless steel. The second is the contact of ceramic on stainless steel with adhered material or 'third bodies' which separate the contacting surfaces. The second contact may be brought about if the interfacial shear strength of the first contact is greater

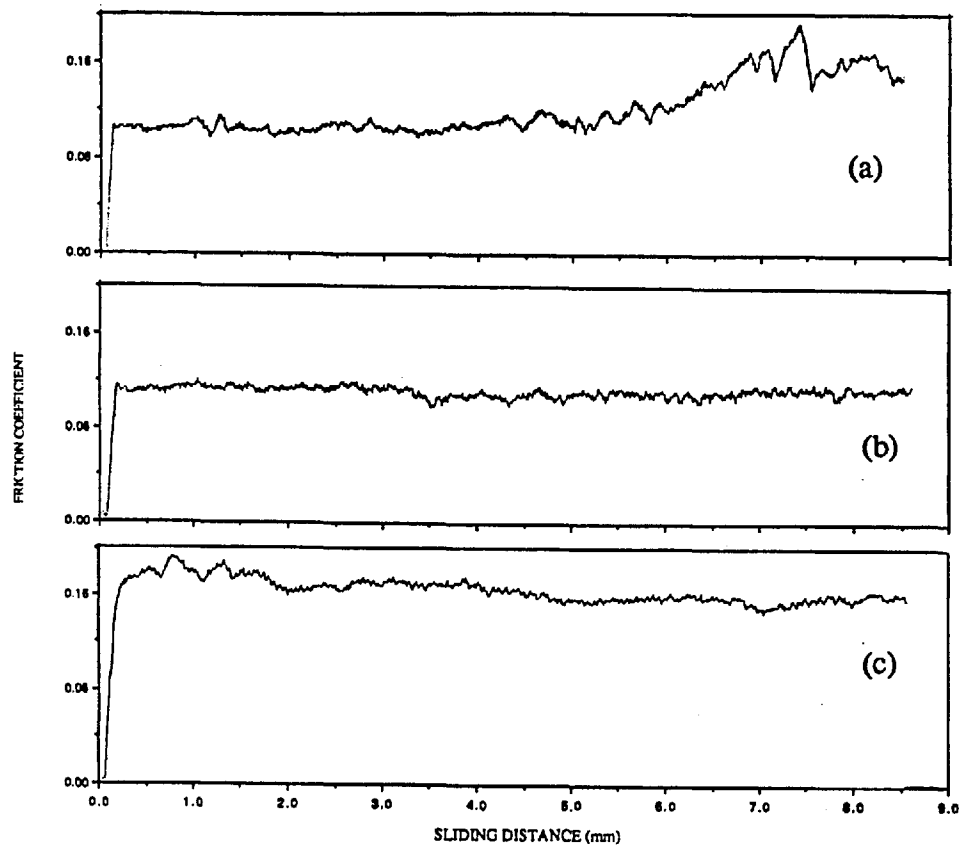


Figure 7.30 Friction traces of a ceramic ball on a similar very smooth surface as that in Figure 7.29. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

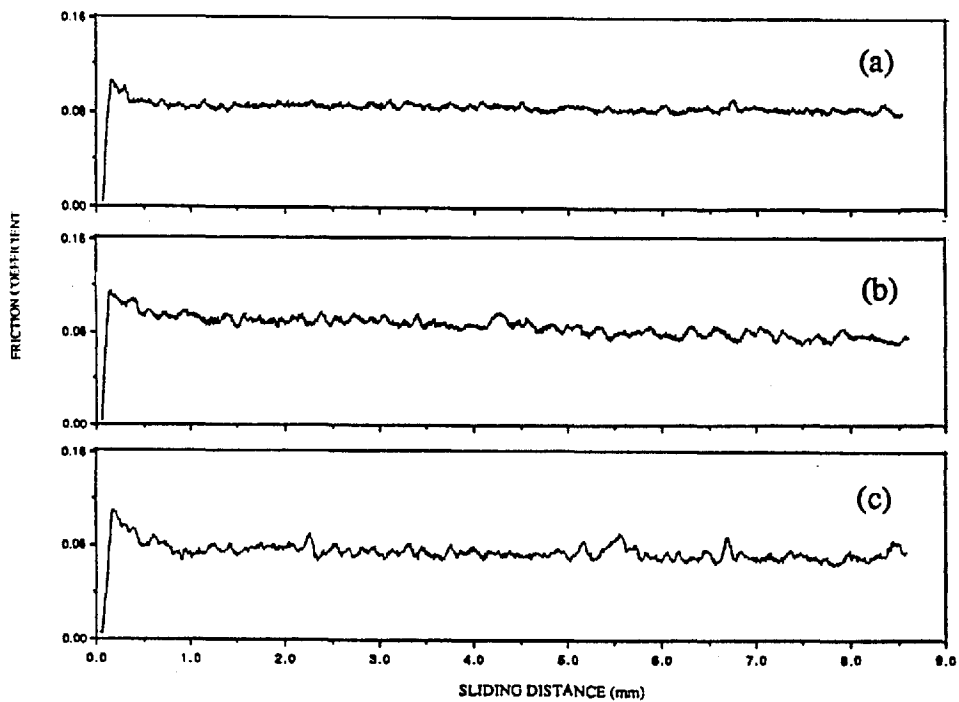


Figure 7.31 Friction traces of a ceramic ball on a rough gear steel surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$

than the shear strength of the bulk material, i.e. the break would occur away from the interface. The sliding friction would therefore decrease with temperature in the first contact and increase with temperature in the second contact.

At $T=25^{\circ}\text{C}$, the high and highly variable friction, and the identification of the adhered material as having come from the counter surface, suggests that the break must have occurred away from the interface. The adhesive junction must have been strong enough to cause the break to occur beneath the surface, and the contact may therefore be regarded as the contact of stainless steel on stainless steel. As the temperature increases, for example at $T=100^{\circ}\text{C}$, the adhesive junctions became weaker, which caused the break to occur in the interface. However in general, the friction increased with sliding distance (as observed in Figure 7.2). As the friction force increased to a value exceeding the shear strength of the bulk metal, then the break would occur away from the interface and the friction was seen to begin to rise. At $T=150^{\circ}\text{C}$, a further reduction in adhesion resulted in less severe surface damage and smooth sliding. However, debris, either from the surrounding environment or generated within from the contacting surfaces could be trapped in the interface. The debris would therefore participate in the abrasion process described earlier resulting in high friction (as shown in Figure 7.30c).

7.3.6 Ceramic Ball on Gear Steel

7.3.6.1 Rough surfaces

Figure 7.31 shows the friction traces of a surface with $\sigma=0.28\text{ }\mu\text{m}$ and $\beta^*=5.60\text{ }\mu\text{m}$. The friction coefficient remained low at all temperatures. No distinguishable surface damage was observed in the wear track, with the exception of a few discontinuous abrasive lines.

7.3.6.2 Smooth surfaces

Figure 7.32 shows the friction traces of a surface with $\sigma = 0.05 \mu\text{m}$ and $\beta^* = 4.5 \mu\text{m}$. The overall frictional behaviour was similar to the rough surface. The friction remained low and steady.

7.3.6.3 Very smooth surfaces

Figure 7.33 shows the friction curve of a surface with $\sigma = 0.018 \mu\text{m}$ and $\beta^* = 3.93 \mu\text{m}$. The friction remained high, at all temperatures, throughout sliding. A series of fine parallel scratches were found on the corresponding wear track, indicating third body abrasion had been taking place. Several surfaces of about the same roughness were tested but the 'stick-slip' friction behaviour observed with the steel ball sliding on very smooth surfaces of stainless steel was not observed in the current case.

7.4 Conclusions

When metals slide on one another there is profound evidence to show that junctions are formed and sheared during sliding. This was shown, for example, in the case of a steel or ceramic ball on both stainless steel and gear steel. Adhesive transfer was evident. This was identified as coming from the countersurface. It should be noted that adhesive transfer occurred in certain conditions, with surfaces that were either rough or very smooth. The effect of roughening the surface is to facilitate the breakdown of the oxide film, by plastic squeezing or smearing of surface asperities, which makes metal-metal contact possible and

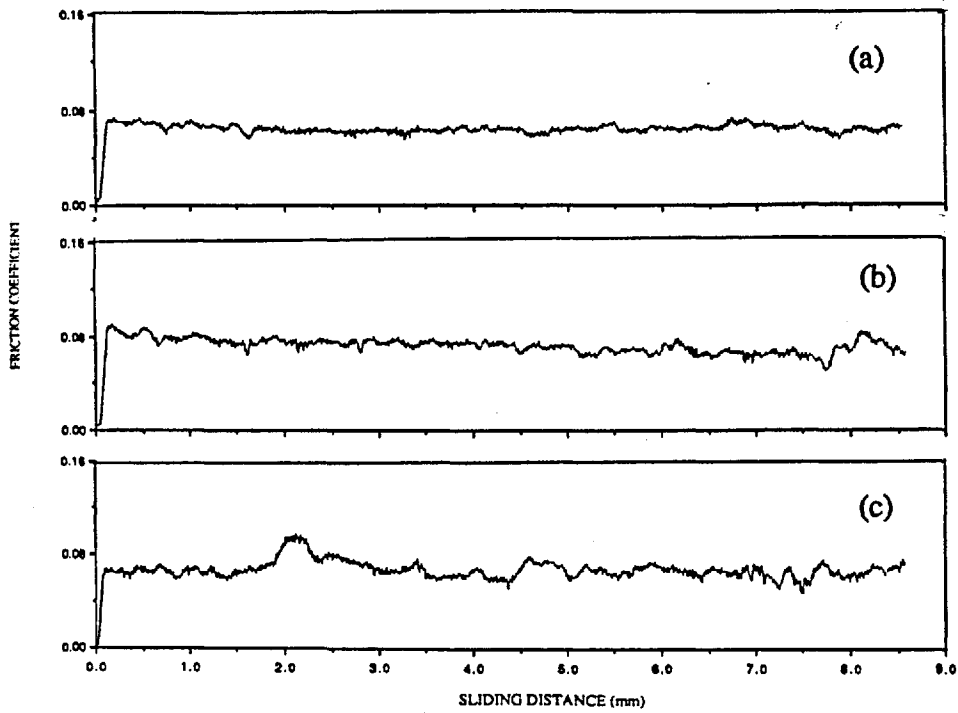


Figure 7.32 Friction traces of a ceramic ball on a smooth gear steel surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

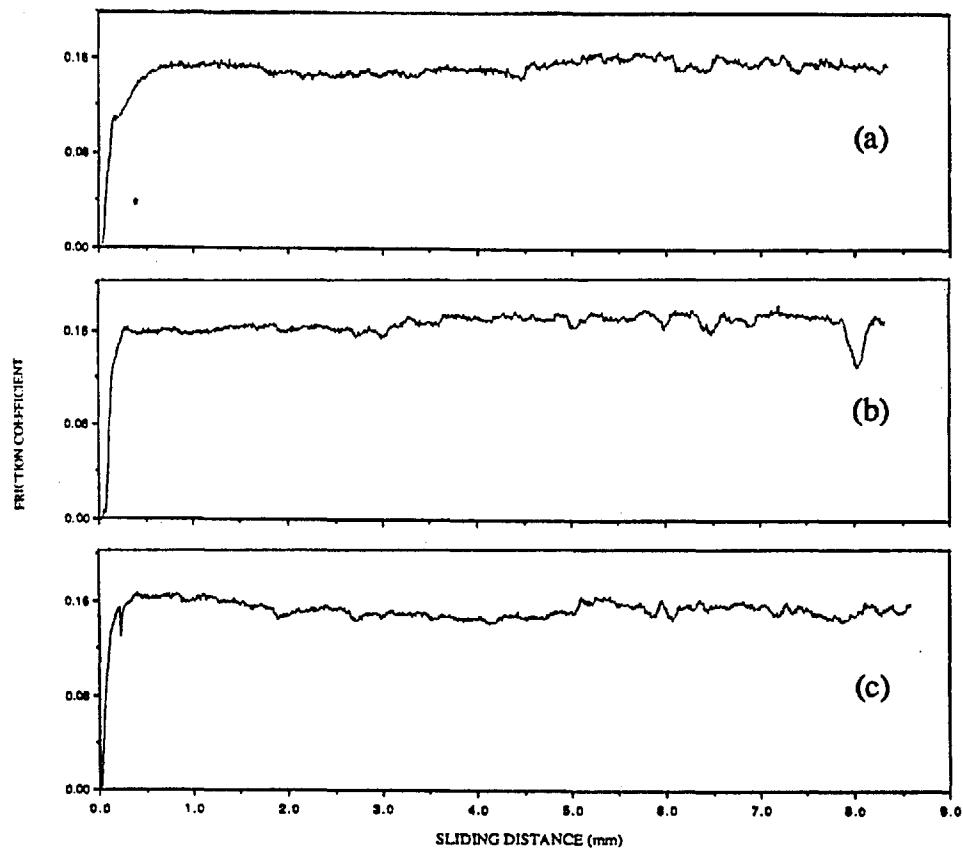


Figure 7.33 Friction traces of a ceramic ball on a very smooth gear steel surface. (a) $T = 50^{\circ}\text{C}$, (b) $T = 100^{\circ}\text{C}$, (c) $T = 150^{\circ}\text{C}$.

results in the formation of strong adhesive junctions. With rough surfaces, the welded junction will be broken by subsequent sliding due to the spacing between contacts. In the case of very smooth surfaces, the welded junction will grow, due to the lack of surface irregularities, until the energy stored up in the elastic deflection arm exceeds the strength of the interface. Then the excess energy will pull the slider against the direction of translation, resulting in a 'slip'. This process produced the stick-slip patterns observed in the wear tracks. Thus, the friction of a very smooth surface is usually higher than that of a rough surface.

With reactive metals, a surface layer of metal oxide is formed. The thickness of surface films will depend on the metal's reactivity. Copper is more reactive than stainless steel and gear steel, thus the oxide film on copper is likely to be thicker than that of either stainless steel or gear steel. The oxide film will react with the acid-containing lubricant to form a layer of metal-soap. This provides an effective boundary lubricant to protect the underlying metal from seizure.

The friction and surface damage of smooth surfaces are usually small. This could be explained in two ways. Firstly, the contact of a smooth surface is predominantly elastic. The layer of oxide film is effective in protecting the underlying metal from gross seizure. Secondly, should any adhesive junction form, the space between contacts would break the junction and prohibit it from being propagated.

The ploughing effect of third bodies is significant, for example, in the case of bearing-steel/gear-steel or ceramic/gear-steel contact. The relative hardness of these materials is of the same order and the effect of plowing by large, hard asperities can be equal or greater than adhesion in the friction process. As a result, a ridge of abrasive wear particles may be formed. This debris can interact further in the rubbing process. It may also result from detachment of material directly by 'micromachining'. Loose wear particles may remain trapped between the two rubbing surfaces for a while resulting in 'third-body'

abrasion before they become free.

In conclusion, the friction force between sliding surfaces results from the combined effects of asperity ploughing, adhesive and deformation, and the relative contribution of the sliding interface, which in turn is affected by the history of sliding, the specific materials used, the surface topography and the environment. Gross seizure is often associated with the rupture of oxide films and film breakdown due to increased plasticity and conformity.

CHAPTER 8

THE INFLUENCE OF SURFACE TOPOGRAPHY ON SLIDING DAMAGE

8.1 Introduction

Hirst and Stafford (1972) examined the factors influencing the failure of the lubricant film in boundary lubrication. They used the transition temperature, from an adequate to an inadequate state of lubricant, as a criterion to determine the effectiveness of lubricant film. A surface with high transition temperature is said to be effectively lubricated. On the other hand, a surface with low transition temperature is said to be ineffectively lubricated and is attributed to the breakdown of lubricant film. They showed that the point of breakdown of lubricant film is determined by the bulk plastic deformation of the contacting surface which is in turn affected by the surface finish as well as load.

Hirst and Hollander (1974) carried on the theme of this research, and related the surface damage to the surface parameters σ and β^* . The results showed that surfaces can be divided by the use of σ - β^* maps into those likely to fail catastrophically and those likely to remain undamaged during sliding. The surface damage for rough surfaces is related to σ/β^* due to plasticity. They also identified that surfaces can be 'too smooth' for safe sliding, but the range of σ and β^* involved was not well defined.

Smooth surface damage is shown in this thesis to be associated with contact conformity, and it is proposed here that this is due to there no longer being any

interruptions within the sliding interface to prevent the growth of small welded junctions. If this is true, then it is important to define the regions of conformity when the surface contact conditions are such that little or no part of the nominal contact area remains out of contact to prevent junction growth.

Rough surface contact conformity has been studied in Chapter 6. It was shown that surface conformity is related to both vertical and spatial parameters. Therefore, if the proposition that the surface damage is due to the conformity of surfaces, then the boundary which divides the safe and unsafe region for α very smooth surface is likely to be dependent on σ and β^* .

In this chapter, friction tests were performed to confirm Hirst and Hollander's work and extend it to several more materials. The range of σ - β^* for safe and unsafe sliding were defined by means of friction tests combined with the contact analysis results of Chapter 5.

8.2 Criterion for Safe and Unsafe Sliding

Frewing (1943) and Hirst & Stafford (1972) both used a transition temperature as a criterion for film breakdown. By Frewing's criterion, the transition temperature is the temperature at which the friction curve changes from a steady state of sliding to a stick-slip sliding. In practice, there is usually no sharp line of demarcation between the steady sliding and the stick-slip sliding. Furthermore, the stick-slip motion is also dependent on the dynamic characteristics of the system.

By Hirst & Stafford's criterion, the transition temperature is the temperature at which the friction level is above a typical value of the boundary state. Unfortunately, by using this criterion with a Bowden-Leben test, a period of about 15 minutes is required for

the heating system of the rig to increase the temperature from ambient to $T=200\text{ }^{\circ}\text{C}$. Thus at a sliding speed of 0.04 mm/sec , then the specimen must be at least 36 mm long. Also the system usually takes about two or three hours to cool down sufficiently for the next test. Thus for a large number of friction tests, this criterion would require a considerable period of testing time. Therefore, an alternative criterion was considered which enables the tests to be carried out at a constant temperature.

Frewing and Hirst & Stafford considered that the transition temperature was dependent on the roughness of the surfaces. For a rough surface interface, transition temperatures were low, and for a smooth surface the transition was at a higher temperature. The transition temperature was established by observing the friction level with temperature. In other words, at a particular temperature, the friction of a surface with high transition temperature should be lower than a surface with low transition temperature. Therefore, an alternative criterion can be based upon the friction level at a certain temperature, i.e. a value typical of the boundary state can be set as a transition friction such that a surface with friction above this value can be described as 'unsafe'.

The transition values established for this work are described in section 8.7.1. The adoption of this criterion has several advantages; it allows the friction tests to be carried out at a constant temperature, which leads to a reduction of the testing time and the specimen size for each test; the length of the specimen can be reduced to 10 mm , which allows a greater consistency of surface parameters over the surface; and the carriage, which holds the specimens, can accommodate more specimens such that four or five tests can be carried out one after the other within one traverse.

8.3 Plasticity Index

The plasticity index, was derived by Greenwood and Williamson (1966) and later

expressed in terms of the correlation length β^* by Whitehouse and Archard (1970)¹, as

$$\psi_w = 0.6 \frac{E' \sigma}{H \beta^*} \quad (8.1)$$

where H is the hardness of the softer material, and

$$\frac{1}{E'} = \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \quad (8.2)$$

ν_1, ν_2, E_1, E_2 are the poisson's ratios and Young's moduli respectively.

8.4 Conformity Index

The conformity index ψ_c is a measure of the real contact area A_r to the nominal area A_n , i.e.

$$\psi_c = \frac{A_r}{A_n} \quad (8.3)$$

The conformity of rough surface contact has been investigated in Chapter 6. For a value of ψ_c , it is given by equations (6.15) to (6.18), i.e.

$$\frac{P_n}{\pi E'} = c \cdot \frac{\sigma}{\beta^*} \quad (8.4)$$

where P_n is the nominal pressure and c is a constant depending on ψ_c . For $\psi_c=1.0, 0.9$ and 0.8 , $c=0.7, 0.38$ and 0.3 respectively.

¹(see Hollander's these (1976) for the derivation of ψ , part of the theory has been reworked, and he found some of the expressions not always to agree with Whitehouse and Archard's)

8.5 Adhesion Index

It is evident that the surface damage for conforming surfaces is largely due to adhesion. Fuller and Tabor (1975) have investigated the effect of roughness on adhesion between two bodies. They showed the adhesion force between the interfaces is significant if

$$\psi_a = \frac{E' \sigma^{3/2}}{R^{1/2} \Delta\gamma} < 10 \quad (8.5)$$

where ψ_a is the adhesion index, E' the reduced elastic modulus of two bodies, σ the surface roughness, R the radius of asperity curvature and $\Delta\gamma$ the surface energy.

It is instructive to compare the adhesion index with the friction test results for smooth surface failure. According to Onions and Archard (1973), the radius of asperity curvature can be related to σ and β^* , i.e.

$$R = 2.08 \frac{\beta^{*2}}{\sigma} \quad (8.6)$$

Substitution of equation (8.6) into equation (8.5), that the adhesion force between a rough surface interface is significant if

$$\psi_a = \frac{0.69 E' \sigma^2}{\beta^* \Delta\gamma} < 10 \quad (8.7)$$

8.6 Experimental Details

The apparatus used to investigate the transition state of frictional behaviour has been described in Chapter 7. The preparation of specimens is also the same but the surfaces with intermediate σ and with large β^* were obtained by an additional procedure of lapping the abraded surface with $1\mu\text{m}$ diamond paste for a short time.

The surface profiles were subjected to a high-pass envelope system or E-system filter. The radius of envelope was taken as that of the steel ball, thus giving the effective surface roughness of the ball relative to the nominal profile. The preference of using an E-system filter to a cubic-spline filter or moving average filter has been discussed in Chapter 2.

The frictional force of each test was analysed on a BBC microcomputer. The mean value was obtained from 1000 digitised points over a sliding distance of about 8mm. The experimental conditions are summarised as follows:

- (a) Upper specimen: En31 bearing steel ball (1/2 inch diameter, 810 Hv, $E=207\text{ GPa}$) or Ceramic ball (3/4 inch diameter, 1100 Hv, $E=310\text{ GPa}$)
- (b) Lower specimen: En58b stainless steel (160 Hv, $E=207\text{ GPa}$) or Gear steel, Brit Standard Aerospace series S156 (700 Hv, $E=207\text{ GPa}$) or Copper (88 Hv, $E=130\text{ GPa}$)
- (c) Lubricant: Solution of stearic acid (1%) in Paraffin.
- (d) Sliding Speed: 0.04 mm/sec
- (e) Load : 10 N
- (f) Temperature: 50 °C , 100 °C , 150 °C

8.7 Results and Discussions

Friction test data is compared with plasticity, conformity and adhesion indices. The plasticity index is given by equation (8.2). In the case of a steel ball on stainless steel, i.e. $H=1.6$ Gpa. $E_1=E_2=207$ Gpa, and $\nu_1=\nu_2=0.3$, the plasticity index becomes $\psi_w=42\sigma/\beta^*$. Similarly, for a steel ball on gear steel $\psi_w=9.7\sigma/\beta^*$. For a ceramic ball on stainless steel $\psi_w=51\sigma/\beta^*$ and for a ceramic ball on gear steel $\psi_w=11.7\sigma/\beta^*$.

The conformity index is given by equation (8.4). In the case of a steel ball on stainless steel or gear steel, the nominal pressure is the load divided by the nominal area derived from the Hertzian contact which gives $p_n/\pi E'=0.0016$. Similarly, for the contact of a ceramic ball on stainless steel or gear steel, $p_n/\pi E'=0.001143$.

The adhesion index is given by equation (8.7). Unfortunately, determination of the surface energy of solids is both difficult and contentious, however there is general agreement concerning a broad range of materials. For most metals, the range of surface energy lies between 1 J/m² to 3 J/m² and for ceramics between 0.3 J/m² and 0.6 J/m². Therefore, the surface energy for steel can be taken as 2 J/m² and ceramic 0.45 J/m². The surface energy $\Delta\gamma$ of two bodies is given by,

$$\Delta\gamma = \gamma_1 + \gamma_2 - \gamma_{12} \quad (8.8)$$

where γ_1, γ_2 are the surface energies of two surfaces and γ_{12} is the interfacial surface energy. By neglecting the term γ_{12} , we have for steel/steel contact $\Delta\gamma=4$ J/m² and for ceramic/steel contact $\Delta\gamma=2.45$ J/m². From the known value of material moduli and the surface energy of each mating pair, the adhesion force is significant for steel/steel contact if

$$\frac{\sigma^2}{\beta^*} < 5.06 \times 10^{-4} \mu m \quad (8.9)$$

Similarly for ceramic/steel contact, the adhesion is significant when

$$\frac{\sigma^2}{\beta^*} < 2.6 \times 10^{-4} \mu m \quad (8.10)$$

8.7.1 Steel Ball on Stainless Steel

Friction test results are summarised in Table 8.1 to 8.3. Figures 8.1 through 8.3 shows the variation of friction with σ/β^* . The scatter of the results is due to the combined effects of several factors such as the consistency of surface parameters, asperity ploughing, 'three-body' abrasion, difference in the surface film thickness, work hardening, etc. However, there appear two distinct trends; the variation in friction at low values of σ/β^* is several times that seen at higher σ/β^* values. The variation in friction appears to slow down at $\mu \approx 0.13$ and therefore this is taken as a value typical of the boundary state. For convenience, a surface is described as 'safe' when $\mu \leq 0.13$ and 'unsafe' when $\mu > 0.13$.

The friction test results are shown on σ - β^* maps in Figure 8.4 to 8.6. At each temperature, an upper boundary to divide the safe and unsafe region is evident and the slope of the boundary appears to decrease with increased temperature. For small value of σ and β^* , a boundary can be drawn to divide the safe and unsafe regions which is seemingly dependent on σ and β^* . The slope of the boundary appears approximately the same with temperature $T=50^\circ\text{C}$ and $T=100^\circ\text{C}$. But at $T=150^\circ\text{C}$, the slope of the boundary increases significantly.

In Hirst and Hollander's experiments, surfaces with a transition temperature above 125°C are describe as 'safe' and below 75°C as 'unsafe'. Therefore, their results are used as a comparison with the the friction tests reported at $T=100^\circ\text{C}$, as shown in Figure 8.5. It

Surfaces	σ (μm)	β^* (μm)	μ	Surfaces	σ (μm)	β^* (μm)	μ
1	0.014	4.660	0.167	51	0.160	3.160	0.104
2	0.058	3.110	0.070	52	0.032	3.010	0.063
3	0.167	9.230	0.125	53	0.058	4.070	0.061
4	0.097	6.100	0.108	54	0.260	4.140	0.103
5	0.060	3.000	0.071	55	0.304	4.360	0.212
6	0.042	3.800	0.096	56	0.020	10.000	0.190
7	0.144	3.150	0.108	57	0.025	11.500	0.200
8	0.007	1.520	0.141	58	0.060	32.000	0.200
9	0.008	1.660	0.196	59	0.022	3.130	0.200
10	0.172	6.210	0.071	60	0.022	22.000	0.280
11	0.160	5.120	0.126	61	0.050	24.970	0.300
12	0.216	12.800	0.094	62	0.070	21.910	0.115
13	0.218	8.800	0.100	63	0.074	13.510	0.085
14	0.206	6.600	0.088	64	0.173	17.535	0.100
15	0.122	3.960	0.070	65	0.300	22.750	0.092
16	0.210	9.920	0.080	66	0.018	8.300	0.125
17	0.225	6.850	0.070	67	0.019	10.250	0.085
18	0.380	5.500	0.100	68	0.048	12.270	0.112
19	0.224	3.510	0.130	69	0.064	16.220	0.107
20	0.087	3.390	0.076	70	0.197	35.550	0.113
21	0.025	6.620	0.090	71	0.137	40.300	0.227
22	0.026	3.960	0.089	72	0.068	30.040	0.143
23	0.016	4.740	0.065	73	0.013	12.410	0.175
24	0.038	1.970	0.110	74	0.113	43.530	0.178
25	0.045	5.930	0.084	75	0.071	74.500	0.149
26	0.035	5.080	0.073	76	0.011	2.860	0.157
27	0.030	3.430	0.103	77	0.020	5.020	0.118
28	0.023	3.230	0.224	78	0.015	15.750	0.156
29	0.004	3.226	0.300	79	0.013	7.030	0.148
30	0.014	4.110	0.269	80	0.034	7.332	0.109
31	0.032	9.300	0.240	81	0.060	3.900	0.127
32	0.130	14.500	0.110	82	0.020	3.860	0.113
33	0.330	11.700	0.170	83	0.038	13.380	0.135
34	0.250	12.500	0.144	84	0.074	27.400	0.118
35	0.198	7.480	0.050	85	0.097	11.650	0.110
36	0.323	12.700	0.090	86	0.153	14.280	0.088
37	0.130	5.900	0.095	87	0.118	6.336	0.110
38	0.069	3.200	0.100	88	0.047	8.086	0.116
39	0.054	4.200	0.110	89	0.080	6.894	0.110
40	0.160	7.140	0.120	90	0.365	12.750	0.076
41	0.024	4.700	0.120	91	0.052	5.000	0.094
42	0.030	2.350	0.130	92	0.058	9.380	0.109
43	0.019	4.800	0.126	93	0.033	9.350	0.132
44	0.086	2.480	0.130	94	0.094	5.930	0.103
45	0.082	3.520	0.180	95	0.018	10.363	0.178
46	0.097	5.080	0.196	96	0.027	13.160	0.160
47	0.290	8.030	0.150				
48	0.015	6.600	0.086				
49	0.036	4.450	0.088				
50	0.090	3.200	0.077				

Table 8.1 Friction tests results of a steel ball on stainless steel at $T = 50^\circ\text{C}$.

Surfaces	σ (μm)	β^* (μm)	μ	Surfaces	σ (μm)	β^* (μm)	μ
1	0.186	3.420	0.200	61	0.036	4.450	0.070
2	0.206	3.820	0.150	62	0.032	3.010	0.050
3	0.094	3.340	0.240	63	0.058	4.070	0.070
4	0.025	3.930	0.080	64	0.260	4.140	0.140
5	0.044	3.080	0.080	65	0.022	3.130	0.280
6	0.113	3.590	0.196	66	0.022	22.000	0.360
7	0.025	3.390	0.090	67	0.004	5.050	0.426
8	0.018	5.130	0.070	68	0.008	7.220	0.349
9	0.035	2.640	0.080	69	0.011	3.500	0.293
10	0.240	3.650	0.250	70	0.019	5.800	0.308
11	0.307	3.590	0.260	71	0.006	3.760	0.349
12	0.028	2.320	0.051	72	0.035	4.420	0.107
13	0.024	8.240	0.177	73	0.008	1.760	0.132
14	0.036	10.430	0.190	74	0.026	3.420	0.210
15	0.022	2.110	0.097	75	0.064	28.600	0.104
16	0.025	3.160	0.060	76	0.020	2.640	0.335
17	0.041	3.470	0.170	77	0.021	2.680	0.310
18	0.088	1.630	0.200	78	0.016	1.540	0.254
19	0.190	12.600	0.066	79	0.070	21.910	0.129
20	0.060	4.220	0.120	80	0.074	13.510	0.085
21	0.250	5.300	0.255	81	0.173	17.535	0.103
22	0.112	10.340	0.120	82	0.300	22.750	0.098
23	0.310	9.740	0.176	83	0.018	8.300	0.102
24	0.290	9.680	0.174	84	0.021	3.770	0.340
25	0.107	6.400	0.170	85	0.019	10.250	0.100
26	0.033	1.870	0.100	86	0.025	4.500	0.242
27	0.022	3.400	0.100	87	0.048	12.270	0.157
28	0.029	7.420	0.270	88	0.010	6.550	0.240
29	0.019	3.800	0.150	89	0.064	16.220	0.101
30	0.260	18.800	0.130	90	0.020	6.340	0.320
31	0.140	13.000	0.100	91	0.197	35.550	0.092
32	0.620	15.500	0.144	92	0.137	40.300	0.290
33	0.060	3.250	0.050	93	0.068	30.040	0.140
34	0.160	7.600	0.076	94	0.013	12.410	0.140
35	0.070	4.290	0.095	95	0.113	43.530	0.317
36	0.084	4.100	0.046	96	0.071	74.500	0.190
37	0.048	3.600	0.080	97	0.011	2.860	0.168
38	0.030	2.510	0.087	98	0.020	5.020	0.128
39	0.026	3.630	0.078	99	0.015	15.750	0.170
40	0.025	2.040	0.050	100	0.013	7.030	0.193
41	0.011	3.400	0.070	101	0.034	7.332	0.112
42	0.019	2.330	0.100	102	0.060	3.900	0.169
43	0.004	3.226	0.330	103	0.020	3.860	0.135
44	0.014	4.110	0.430	104	0.038	13.380	0.178
45	0.033	8.000	0.350	105	0.074	27.400	0.200
46	0.032	9.300	0.260	106	0.097	11.850	0.116
47	0.330	11.700	0.170	107	0.153	14.280	0.129
48	0.250	12.500	0.180	108	0.118	6.336	0.117
49	0.054	4.200	0.120	109	0.047	8.086	0.124
50	0.037	4.600	0.186	110	0.080	6.894	0.117
51	0.160	7.140	0.140	111	0.052	5.000	0.124
52	0.024	4.700	0.130	112	0.058	9.380	0.118
53	0.030	2.350	0.160	113	0.033	9.350	0.145
54	0.035	3.680	0.155	114	0.094	5.930	0.148
55	0.019	4.800	0.150	115	0.018	10.363	0.240
56	0.086	2.480	0.200	116	0.027	13.160	0.212
57	0.082	3.520	0.186	117	0.001	5.250	0.113
58	0.097	5.080	0.240				
59	0.290	8.030	0.150				
60	0.015	6.600	0.090				

Table 8.2 Friction tests results of a steel ball on stainless steel at $T = 100^\circ\text{C}$.

Surfaces	σ (μm)	β^* (μm)	μ	Surfaces	σ (μm)	β^* (μm)	μ
1	0.032	3.010	0.064	46	0.035	3.680	0.250
2	0.058	3.280	0.071	47	0.097	5.100	0.250
3	0.041	5.750	0.072	48	0.036	4.450	0.255
4	0.038	5.950	0.080	49	0.021	2.550	0.260
5	0.047	8.086	0.090	50	0.113	43.530	0.260
6	0.054	4.200	0.092	51	0.026	6.840	0.267
7	0.028	5.200	0.100	52	0.160	7.140	0.270
8	0.097	11.850	0.103	53	0.020	3.860	0.270
9	0.153	14.280	0.103	54	0.030	2.350	0.274
10	0.058	9.380	0.104	55	0.098	9.040	0.274
11	0.260	18.740	0.104	56	0.019	4.800	0.276
12	0.106	18.540	0.110	57	0.064	16.220	0.276
13	0.052	5.000	0.115	58	0.015	15.750	0.278
14	0.027	4.135	0.120	59	0.048	12.270	0.280
15	0.197	35.550	0.120	60	0.071	74.500	0.281
16	0.118	6.336	0.130	61	0.013	12.410	0.285
17	0.047	3.270	0.140	62	0.020	5.020	0.292
18	0.033	9.350	0.141	63	0.070	21.910	0.295
19	0.126	4.050	0.143	64	0.018	10.363	0.302
20	0.236	8.750	0.145	65	0.013	7.030	0.304
21	0.090	3.200	0.145	66	0.027	13.160	0.304
22	0.094	5.930	0.146	67	0.074	13.510	0.307
23	0.060	3.900	0.146	68	0.038	13.380	0.308
24	0.029	3.300	0.150	69	0.018	1.920	0.310
25	0.160	3.160	0.150	70	0.018	4.960	0.310
26	0.058	4.070	0.150	71	0.173	17.535	0.311
27	0.120	3.560	0.154	72	0.037	4.600	0.313
28	0.156	6.700	0.163	73	0.034	7.332	0.315
29	0.365	12.750	0.164	74	0.304	4.360	0.320
30	0.213	3.710	0.174	75	0.137	40.300	0.325
31	0.022	5.323	0.176	76	0.720	8.160	0.330
32	0.300	22.750	0.177	77	0.060	32.000	0.330
33	0.015	6.600	0.185	78	0.074	27.400	0.333
34	0.290	8.030	0.190	79	0.032	9.300	0.364
35	0.330	11.700	0.210	80	0.004	3.226	0.370
36	0.086	2.480	0.210	81	0.014	4.110	0.370
37	0.068	30.040	0.210	82	0.022	3.130	0.370
38	0.019	10.250	0.216	83	0.033	8.000	0.385
39	0.260	4.140	0.220	84	0.022	22.000	0.410
40	0.022	1.870	0.230	85	0.050	24.970	0.410
41	0.250	12.500	0.240				
42	0.082	3.520	0.240				
43	0.080	6.894	0.240				
44	0.018	8.300	0.246				
45	0.011	2.860	0.248				

Table 8.3 Friction tests results of a steel ball on stainless steel at $T = 150^\circ\text{C}$.

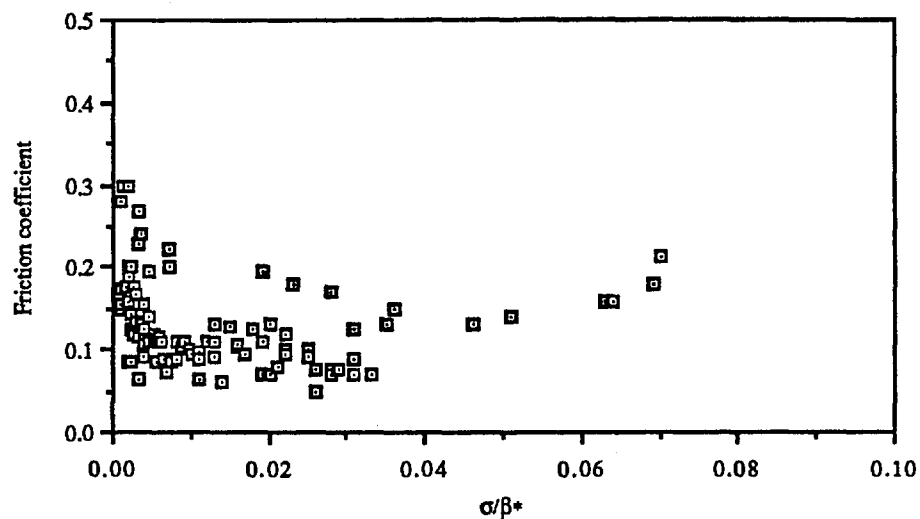


Figure 8.1 Variation of friction with σ/β^* for a steel ball on stainless steel at $T = 50^\circ\text{C}$.

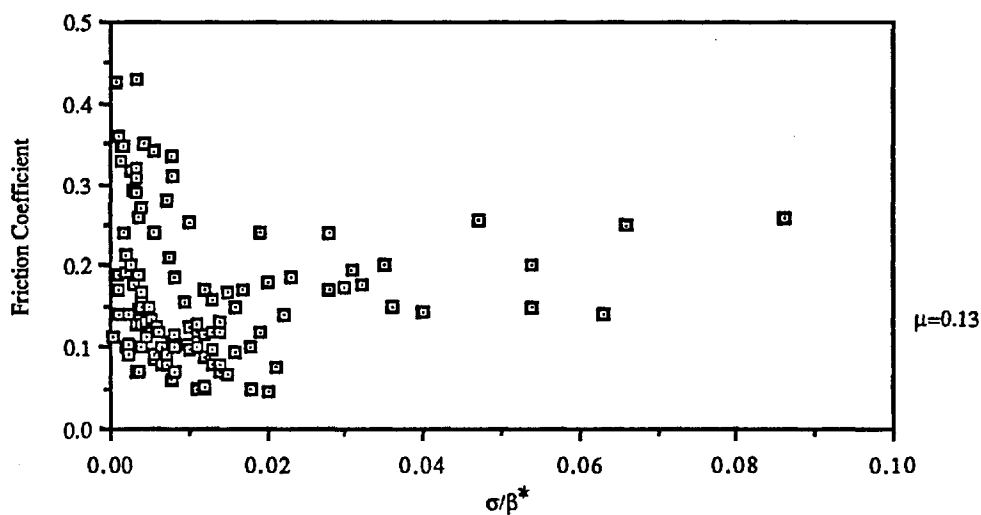


Figure 8.2 Variation of friction with σ/β^* for a steel ball on stainless steel at $T = 100^\circ\text{C}$.

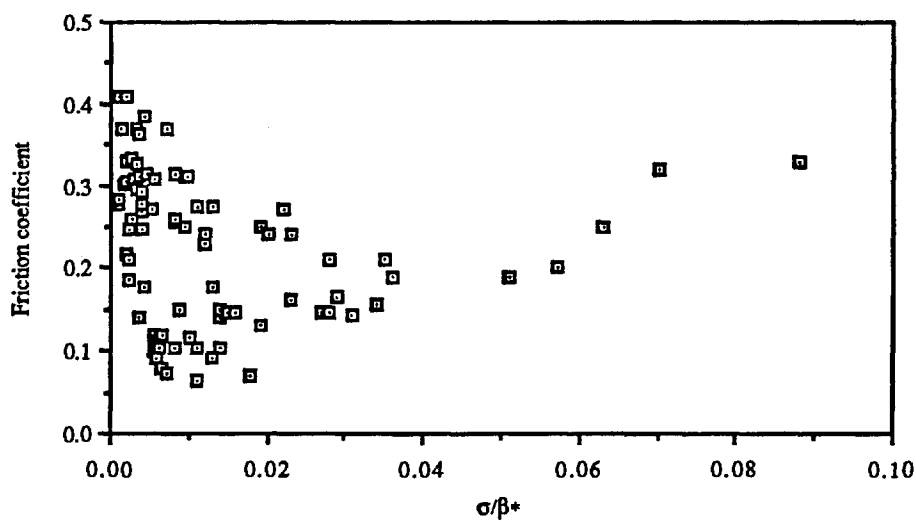


Figure 8.3 Variation of friction with σ/β^* for a steel ball on stainless steel at $T = 150^\circ\text{C}$.

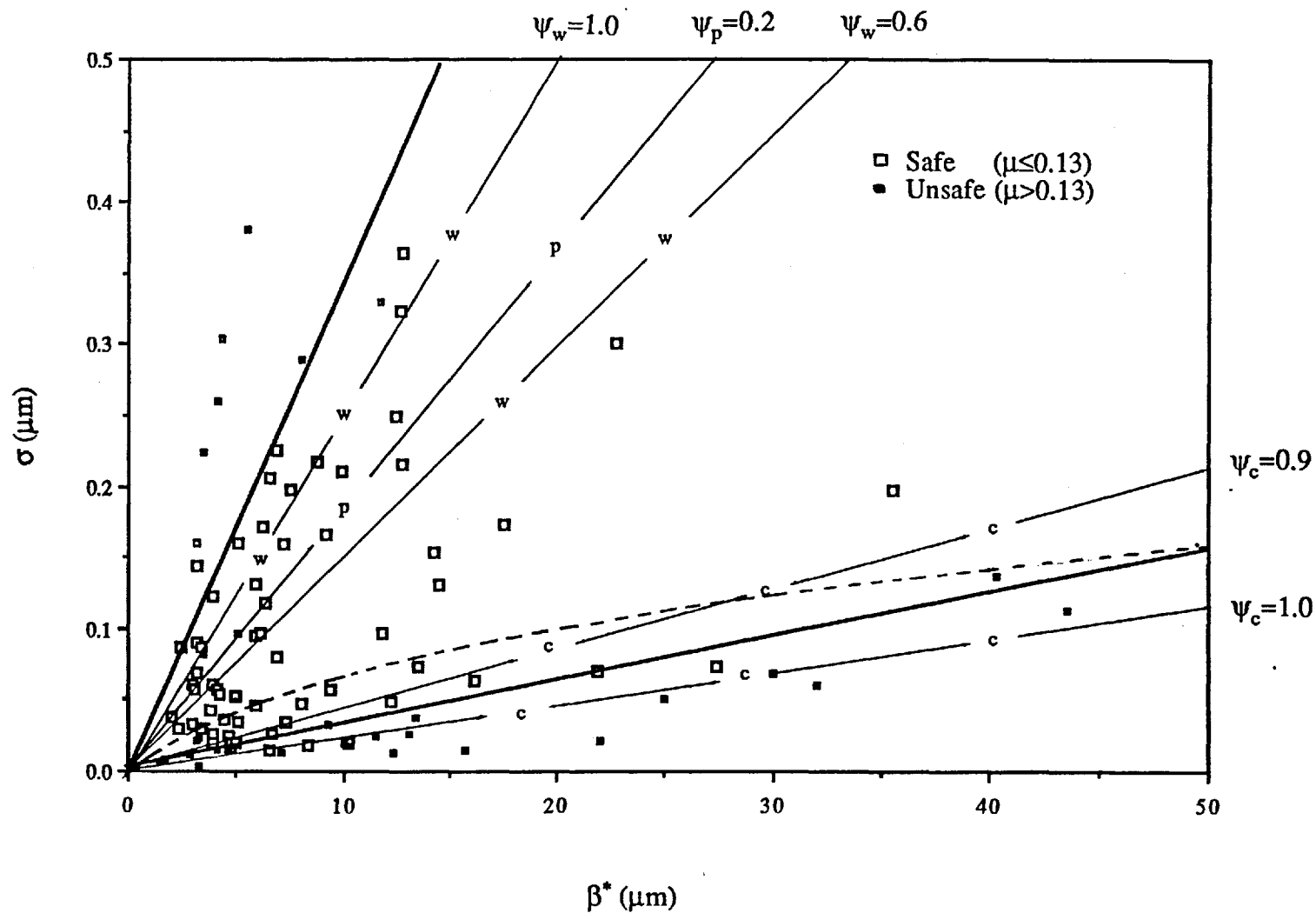


Figure 8.4 σ - β^* map of a steel ball on stainless steel at $T = 50^\circ\text{C}$ showing the boundary between 'unsafe' and 'safe' regions. The lines of $\psi_c = A_r/A_n$ —c— and $\psi_p = A_p/A_n$ —p— are obtained from the contact analysis on rough surfaces. The plasticity lines —w— are given by Whitehouse. The adhesion line --- is given by equation (8.9).

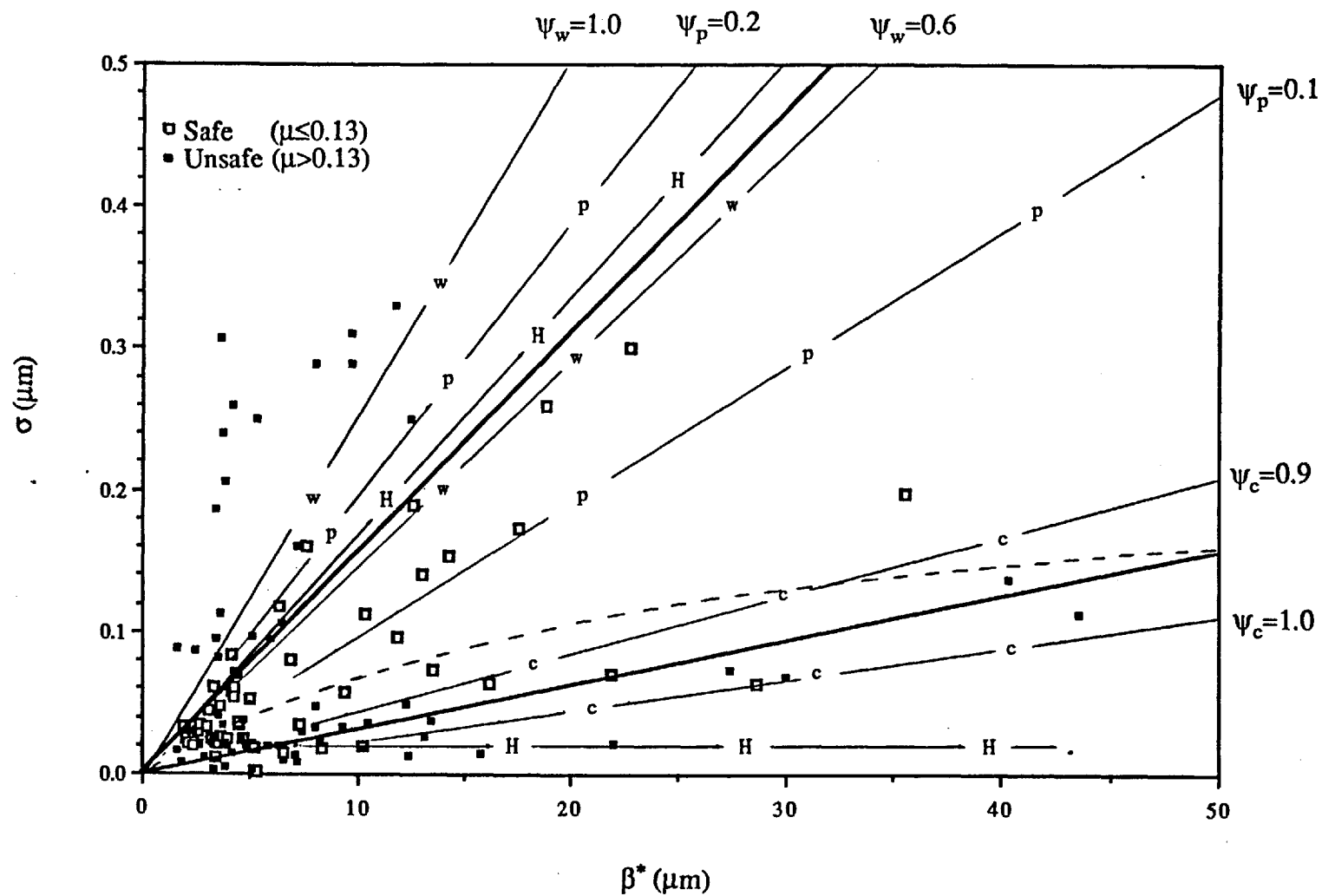


Figure 8.5 As Figure 8.4 but $T = 100^\circ C$ and the results given by Hollander—H—are shown for comparison.

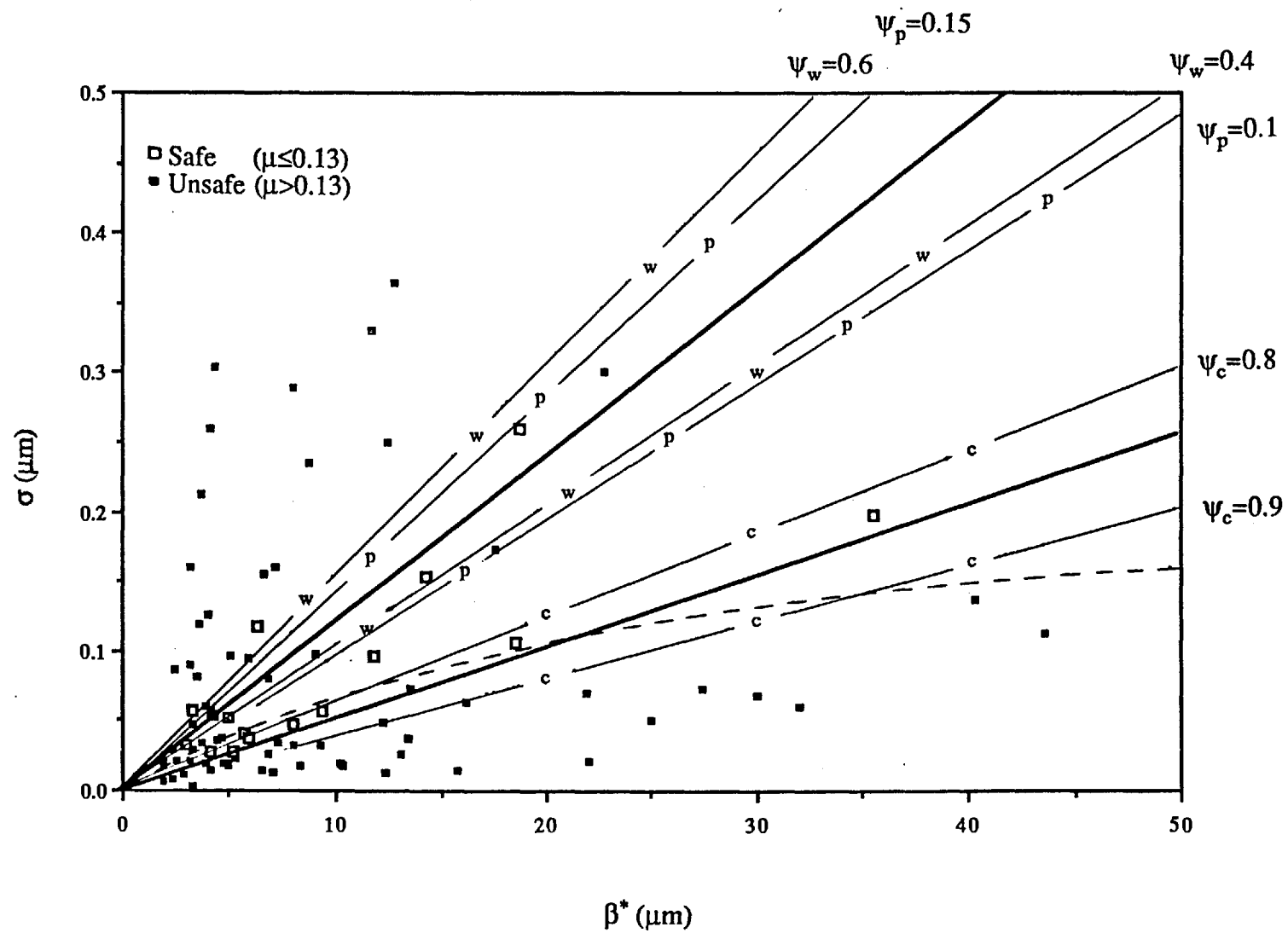


Figure 8.6 As Figure 8.4 but $T = 150^\circ\text{C}$.

can be seen that the upper boundary compares well with Hollander but the lower boundary appears dependent on β^* as well as σ .

The upper boundary has been explained by plasticity and runs approximately parallel to a plasticity line, representing a significant percentage of plastic contact area which is confirmed by numerical contact analysis. The boundary corresponds to about 15% plastic area. The plasticity lines given by Whitehouse and Archard are also shown on the map. The upper boundary corresponds to $\psi_w \approx 0.8$ indicating the contact at the upper boundary is predominately plastic.

The lower boundary appears close to the adhesion line and the regions where the contacts reach almost total conformity. From the mechanistic study of surface damage of a very smooth stainless steel surface in Chapter 7, it is evident that surface failure was due to adhesion, here it is shown to be related to the surface conformity. Therefore, a possible explanation is as follows; The process of a ball sliding on a very smooth surface facilitates the breakdown of the oxide film and the formation of a welded junction which can grow in size through frictional heating due to the lack of interruptions within the interface, normally present with rougher surfaces.

Hirst and Hollander's results also identified that the surfaces failed at $\beta^* < 3 \mu\text{m}$ regardless of σ . However, the results presented show combined 'safe' and 'unsafe' surfaces at small β^* . This may be explained by the difference in experimental techniques and digital sampling errors in measuring the surface profiles. In Chapter 7, it has been shown that the sliding distance at which the transitional friction occurred on rough surfaces increased as surface roughness decreased. Hirst and Hollander have not mentioned the sliding distance in their tests. However, in view of the sliding speed and the time taken to heat up the lubricant from ambient to a temperature of 200 °C, the sliding distance is likely to have been several times than that of the present tests. Thus increasing the tendency for the transitional friction to occur for a less rough surface in their experiments.

The errors in sampling are two-fold. Firstly, the finite size of the stylus will fail to detect the valleys with steeper slopes and acts as a mechanical filter which effectively removes the much shorter wavelength features thereby the underestimating σ and overestimating β^* . Secondly, the sampling interval should be at least twice the wavelength of interest, according to the Nyquist criterion. Failure to do so may result in aliasing or loss of information. Here, the sample interval was set at 2 μm . Errors are likely to occur at β^* less than 4 μm .

The slope of the upper boundary decreases with increasing temperature. This may be due to the increased effectiveness of boundary lubrication at lower temperature, however, it may also be due to the effect of work hardening in the process of preparing the specimens. One of the factors affecting the friction increase with temperature may be the annealing and thermal softening of the surfaces. In order to test this, micro-hardness of a stainless steel surface, prepared in the same way as for the friction test, was measured by using a diamond pyramid indenter at a load of 10N at both ambient and elevated temperatures. Corresponding friction tests were performed at $T=50\text{ }^{\circ}\text{C}$, before and after heating. The friction traces are shown in Figure 8.7, the results of micro-hardness tests are summarised in Table 8.4. It can be seen that the hardness of the material was higher than its bulk hardness initially due to work hardening, and continuously decreased with temperature. When the temperature returned to ambient, the hardness was equal to its bulk hardness. The friction traces also show the irreversible behaviour. Therefore, the friction might be expected to increase with temperature due to annealing effects, and to increase the slope of the upper boundary.

The friction of a surface with an intermediate value of σ and β^* is generally low. This may be due to the surface's ability to trap the lubricant between contacts which effectively reduces adhesion, but it may also be the result of local welded and adhered

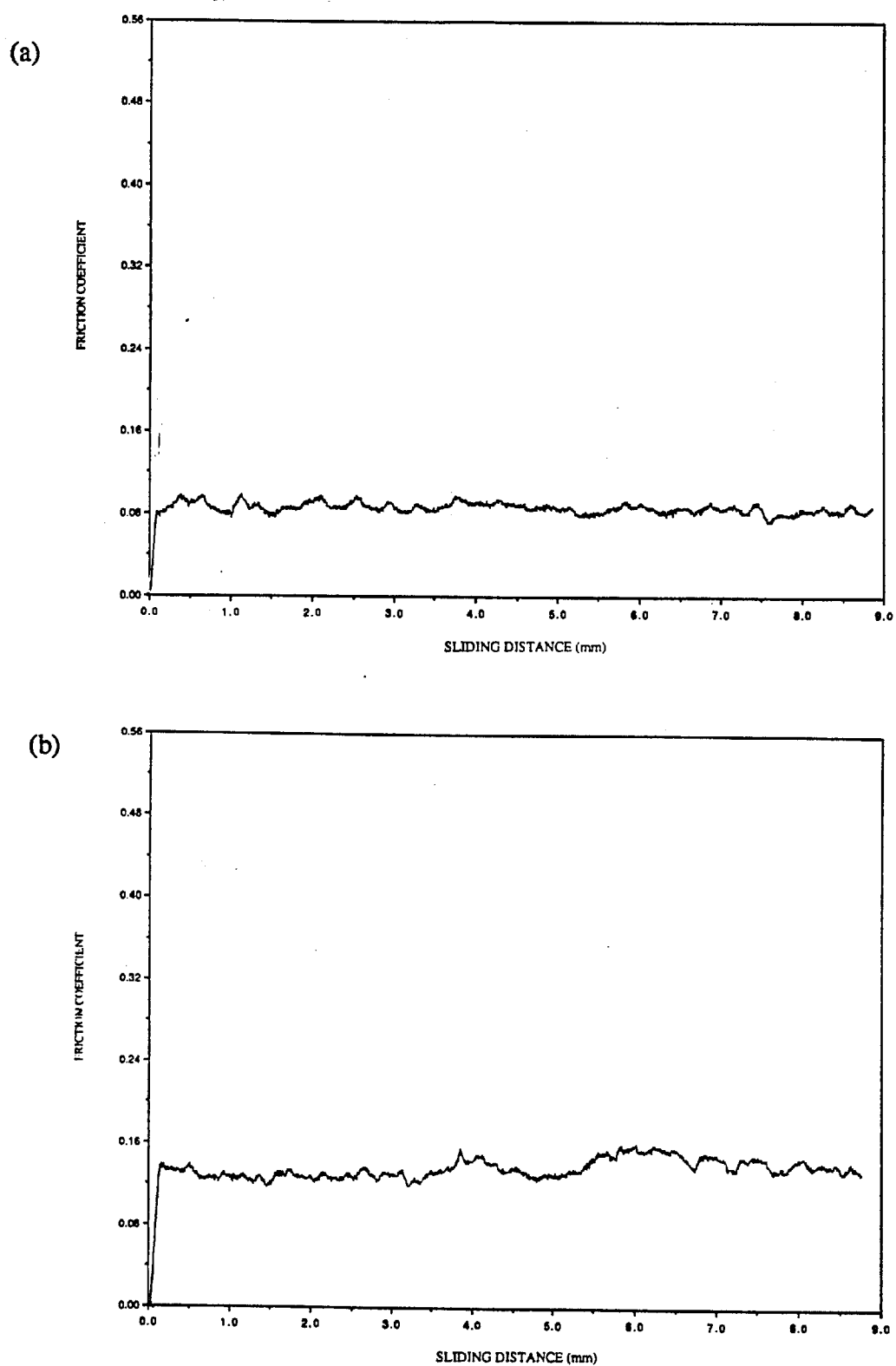


Figure 8.7 Comparison of friction traces.
(a) At $T = 50\text{ }^{\circ}\text{C}$.
(b) At $T = 50\text{ }^{\circ}\text{C}$ after the specimen's temperature has been increased to $T = 200\text{ }^{\circ}\text{C}$.

<u>Temperature (°C)</u>	<u>Micro-hardness (Hv)</u>
22	230
50	194
100	162
150	153
200	129
150	154
100	174
50	162
22	160

Table 8.4

junctions being prevented from growing during sliding because of the gaps in the contact interface.

8.7.2 Ceramic Ball on Stainless Steel

Friction test results are summarised in Tables 8.5 to 8.7. Figures 8.8 to 8.10 show the variation of friction with σ/β^* . The friction curves follow a similar pattern to those of steel balls on stainless steel, although it is of interest to note that (in general) the friction at $T=150\text{ }^{\circ}\text{C}$ drops slightly by comparison with $T=100\text{ }^{\circ}\text{C}$ particularly at the intermediate value of σ/β^* . This could be explained by the types of possible contact occurring in the interface as in Chapter 7 or where contact is induced by the wear particles from the softer material. At a higher temperature, the decrease in adhesion results in fewer wear particles and therefore the first type of contact predominates. Increased roughness facilitates plastic deformation and the rupture of surface asperities to form wear particles whereupon the second type of contact predominates.

A value of friction coefficient typical of the boundary state was obtained for which $\mu=0.09$, the safe and unsafe regions at different temperatures are shown on $\sigma-\beta^*$ maps, Figure 8.11 to 8.13. An upper boundary is evident for each temperature and the boundary at $T=50\text{ }^{\circ}\text{C}$ and $T=100\text{ }^{\circ}\text{C}$ are close to that of a steel ball on stainless steel at the same temperature. However at $T=150\text{ }^{\circ}\text{C}$, the slope of the upper boundary decreases for the reason explained (above).

A lower boundary is also evident, largely independent of temperature, close to the adhesion line and the regions of contact conformity.

Surfaces	$\sigma (\mu\text{m})$	$\beta^* (\mu\text{m})$	μ	Surfaces	$\sigma (\mu\text{m})$	$\beta^* (\mu\text{m})$	μ
1	0.063	3.250	0.072	41	0.070	21.910	0.098
2	0.142	6.490	0.076	42	0.074	13.510	0.080
3	0.100	5.570	0.079	43	0.173	17.535	0.083
4	0.080	3.450	0.076	44	0.300	22.750	0.083
5	0.042	3.800	0.090	45	0.018	8.300	0.130
6	0.154	3.430	0.074	46	0.019	10.250	0.095
7	0.010	6.220	0.107	47	0.048	12.270	0.110
8	0.009	2.310	0.140	48	0.064	16.220	0.090
9	0.191	10.950	0.074	49	0.197	35.550	0.089
10	0.123	3.560	0.122	50	0.137	40.300	0.186
11	0.773	6.540	0.145	51	0.068	30.040	0.087
12	0.245	17.770	0.088	52	0.013	12.410	0.104
13	0.158	7.730	0.082	53	0.113	43.530	0.088
14	0.110	3.540	0.081	54	0.071	74.500	0.100
15	0.110	3.540	0.090	55	0.011	2.860	0.101
16	0.220	8.510	0.068	56	0.020	5.020	0.084
17	0.187	5.740	0.087	57	0.015	15.750	0.118
18	0.410	7.170	0.200	58	0.013	7.030	0.133
19	0.176	3.320	0.156	59	0.034	7.332	0.077
20	0.099	5.760	0.144	60	0.060	3.900	0.086
21	0.027	7.360	0.089	61	0.020	3.860	0.107
22	0.023	3.400	0.088	62	0.038	13.380	0.109
23	0.009	3.800	0.065	63	0.074	27.400	0.105
24	0.029	5.770	0.110	64	0.097	11.850	0.080
25	0.041	5.120	0.084	65	0.153	14.280	0.073
26	0.035	5.080	0.073	66	0.118	6.336	0.079
27	0.029	3.720	0.100	67	0.047	8.086	0.088
28	0.017	5.330	0.100	68	0.080	6.894	0.089
29	0.004	3.226	0.096	69	0.365	12.750	0.063
30	0.014	4.110	0.090	70	0.052	5.000	0.103
31	0.033	8.000	0.100	71	0.058	9.380	0.090
32	0.032	9.300	0.050	72	0.033	9.350	0.092
33	0.106	18.540	0.080	73	0.094	5.930	0.105
34	0.130	14.500	0.070	74	0.018	10.363	0.108
35	0.330	11.700	0.080	75	0.027	13.160	0.087
36	0.250	12.500	0.062				
37	0.086	2.480	0.090				
38	0.082	3.520	0.073				
39	0.097	5.100	0.088				
40	0.290	8.030	0.080				

Table 8.5 Friction tests results of a ceramic ball on stainless steel at T= 50°C.

Surfaces	σ (μm)	β^* (μm)	μ	Surfaces	σ (μm)	β^* (μm)	μ
1	0.162	2.990	0.190	41	0.207	10.750	0.094
2	0.092	2.500	0.130	42	0.115	3.680	0.120
3	0.033	5.210	0.072	43	0.021	6.340	0.165
4	0.048	3.760	0.090	44	0.070	21.910	0.145
5	0.111	3.143	0.124	45	0.074	13.510	0.075
6	0.025	2.675	0.080	46	0.173	17.535	0.087
7	0.092	25.040	0.080	47	0.300	22.750	0.084
8	0.024	9.290	0.080	48	0.018	8.300	0.110
9	0.038	1.980	0.074	49	0.019	10.250	0.106
10	0.243	3.920	0.195	50	0.048	12.270	0.115
11	0.314	3.830	0.172	51	0.064	16.220	0.110
12	0.027	3.390	0.075	52	0.137	40.360	0.094
13	0.026	6.450	0.090	53	0.068	30.040	0.125
14	0.039	3.067	0.110	54	0.013	12.410	0.121
15	0.023	2.715	0.087	55	0.113	43.530	0.100
16	0.023	3.095	0.080	56	0.071	74.500	0.110
17	0.023	2.104	0.080	57	0.011	2.860	0.102
18	0.083	2.080	0.140	58	0.020	5.020	0.077
19	0.156	10.050	0.067	59	0.013	7.030	0.108
20	0.070	5.510	0.090	60	0.034	7.332	0.070
21	0.323	8.100	0.202	61	0.060	3.900	0.076
22	0.055	12.260	0.090	62	0.020	3.860	0.110
23	0.330	11.470	0.110	63	0.098	9.040	0.090
24	0.290	10.130	0.120	64	0.038	13.380	0.123
25	0.090	4.820	0.104	65	0.074	27.400	0.084
26	0.035	1.910	0.070	66	0.097	11.850	0.085
27	0.021	3.038	0.110	67	0.118	6.336	0.074
28	0.047	24.980	0.160	68	0.047	8.086	0.087
29	0.013	2.850	0.097	69	0.080	6.894	0.080
30	0.161	10.370	0.100	70	0.365	12.750	0.055
31	0.162	14.980	0.090	71	0.052	5.000	0.099
32	0.681	13.910	0.100	72	0.058	9.380	0.085
33	0.014	1.755	0.150	73	0.033	9.350	0.088
34	0.060	2.960	0.060	74	0.094	5.930	0.096
35	0.173	9.830	0.120	75	0.018	10.363	0.101
36	0.080	4.370	0.070	76	0.027	13.160	0.108
37	0.086	4.040	0.050				
38	0.045	3.660	0.060				
39	0.153	2.911	0.160				
40	0.007	1.850	0.090				

Table 8.6 Friction tests results of a ceramic ball on stainless steel at T= 100°C.

Surfaces	$\sigma(\mu\text{m})$	$\beta^*(\mu\text{m})$	μ	Surfaces	$\sigma(\mu\text{m})$	$\beta^*(\mu\text{m})$	μ
1	0.042	6.830	0.090	41	0.300	22.750	0.083
2	0.016	2.360	0.088	42	0.018	8.300	0.136
3	0.015	1.360	0.112	43	0.019	10.250	0.110
4	0.019	1.730	0.079	44	0.048	12.270	0.104
5	0.062	3.200	0.056	45	0.064	16.220	0.117
6	0.140	5.320	0.080	46	0.197	35.550	0.090
7	0.079	3.677	0.064	47	0.137	40.300	0.117
8	0.085	3.440	0.045	48	0.068	30.040	0.131
9	0.048	3.140	0.048	49	0.013	12.410	0.124
10	0.140	2.550	0.126	50	0.113	43.530	0.106
11	0.800	12.400	0.204	51	0.071	74.500	0.135
12	0.235	14.300	0.064	52	0.011	2.860	0.105
13	0.154	6.800	0.062	53	0.020	5.020	0.083
14	0.200	6.550	0.080	54	0.015	15.750	0.092
15	0.200	6.550	0.050	55	0.034	7.332	0.082
16	0.260	8.170	0.060	56	0.060	3.900	0.075
17	0.340	5.870	0.188	57	0.020	3.860	0.119
18	0.178	3.400	0.156	58	0.098	9.040	0.073
19	0.093	4.220	0.090	59	0.038	13.380	0.112
20	0.026	7.950	0.080	60	0.074	27.400	0.094
21	0.057	11.600	0.074	61	0.097	11.850	0.084
22	0.013	4.850	0.074	62	0.153	14.280	0.063
23	0.028	4.300	0.078	63	0.118	6.336	0.073
24	0.041	5.120	0.052	64	0.047	8.086	0.090
25	0.033	8.100	0.052	65	0.080	6.894	0.080
26	0.028	2.740	0.074	66	0.365	12.750	0.048
27	0.015	2.840	0.092	67	0.052	5.000	0.090
28	0.004	3.226	0.110	68	0.058	9.380	0.085
29	0.014	4.110	0.100	69	0.033	9.350	0.092
30	0.033	8.000	0.065	70	0.094	5.930	0.075
31	0.032	9.300	0.075	71	0.018	10.363	0.162
32	0.033	11.700	0.070	72	0.027	13.160	0.094
33	0.250	12.500	0.050				
34	0.086	2.480	0.070				
35	0.082	3.520	0.050				
36	0.097	5.100	0.079				
37	0.290	8.030	0.058				
38	0.070	21.910	0.126				
39	0.074	13.510	0.090				
40	0.173	17.535	0.080				

Table 8.7 Friction tests results of a ceramic ball on stainless steel at T= 150°C.

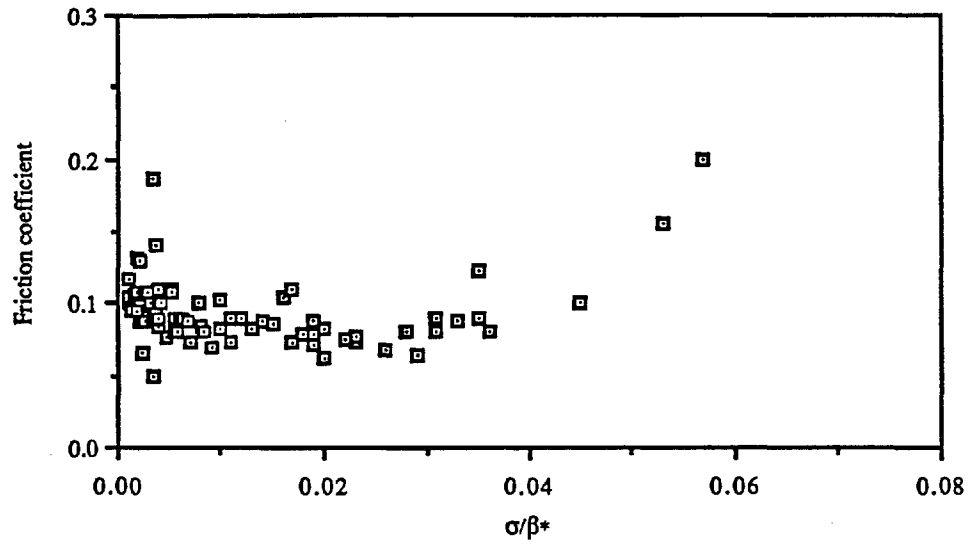


Figure 8.8 Variation of friction with σ/β^* for a ceramic ball on stainless steel at $T = 50^\circ\text{C}$.

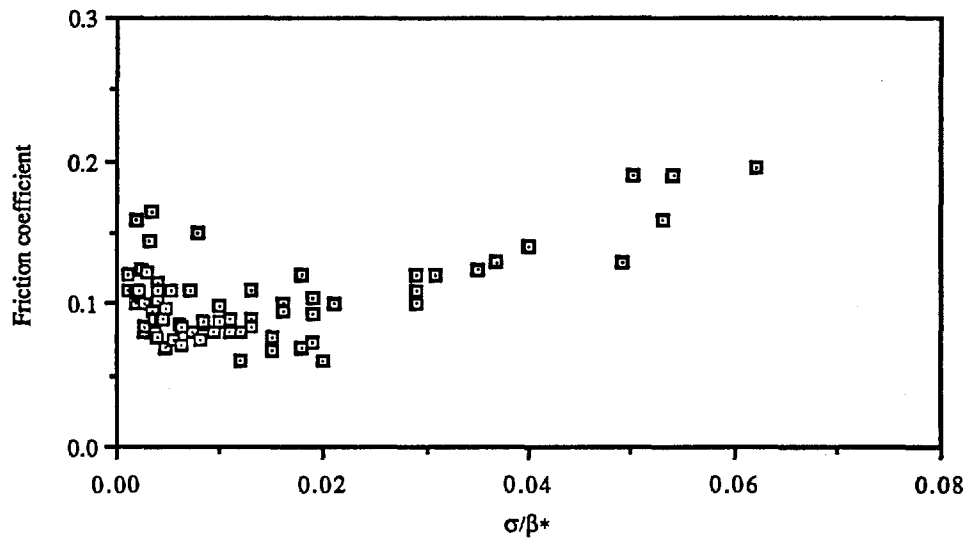


Figure 8.9 Variation of friction with σ/β^* for a ceramic ball on stainless steel at $T = 100^\circ\text{C}$.

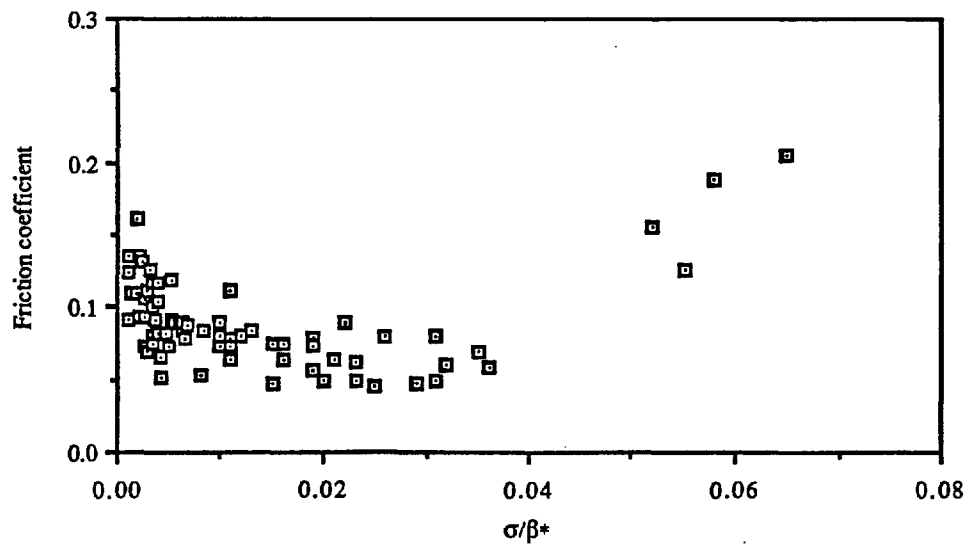


Figure 8.10 Variation of friction with σ/β^* for a ceramic ball on stainless steel at $T = 150^\circ\text{C}$.

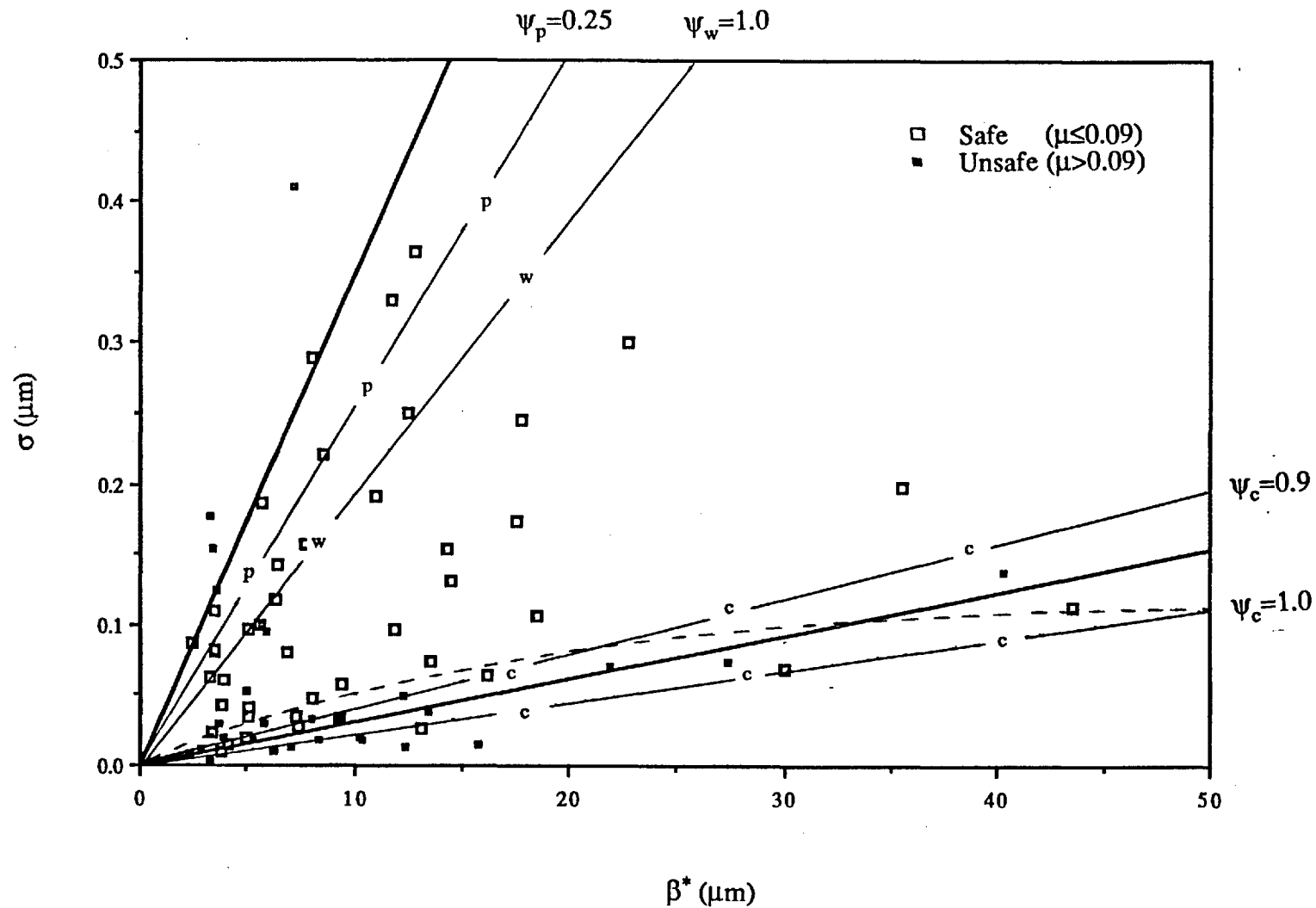


Figure 8.11 σ - β^* map of a ceramic ball on stainless steel at $T = 50^\circ\text{C}$ showing the boundary between 'unsafe' and 'safe' regions. The lines of $\psi_c = A_r/A_n$ —c— are predicted by stochastic approach. The plasticity lines —w— are given by Whitehouse. The adhesion line --- is given by equation (8.10).

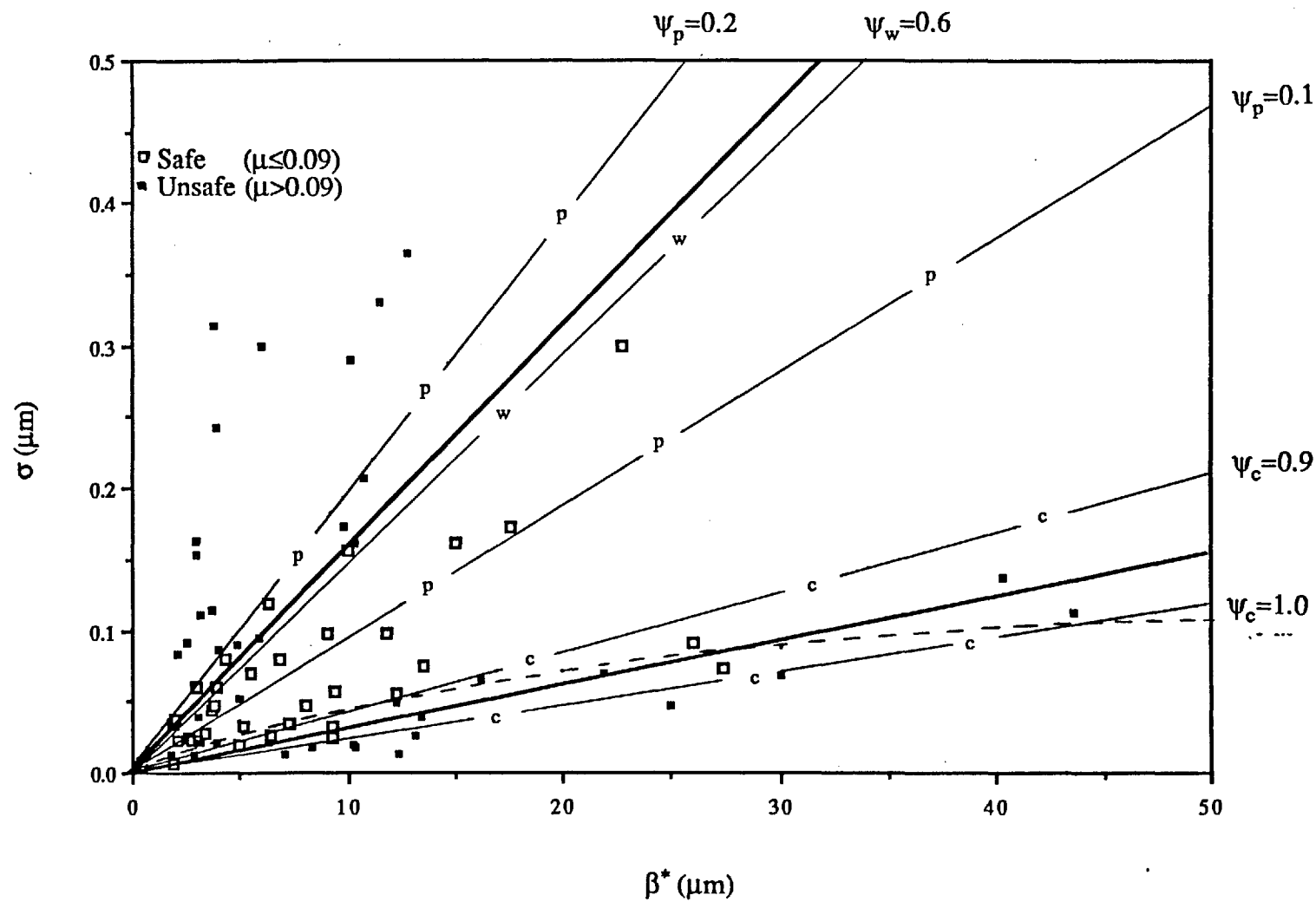


Figure 8.12 As Figure 8.11 but $T = 100^\circ\text{C}$.

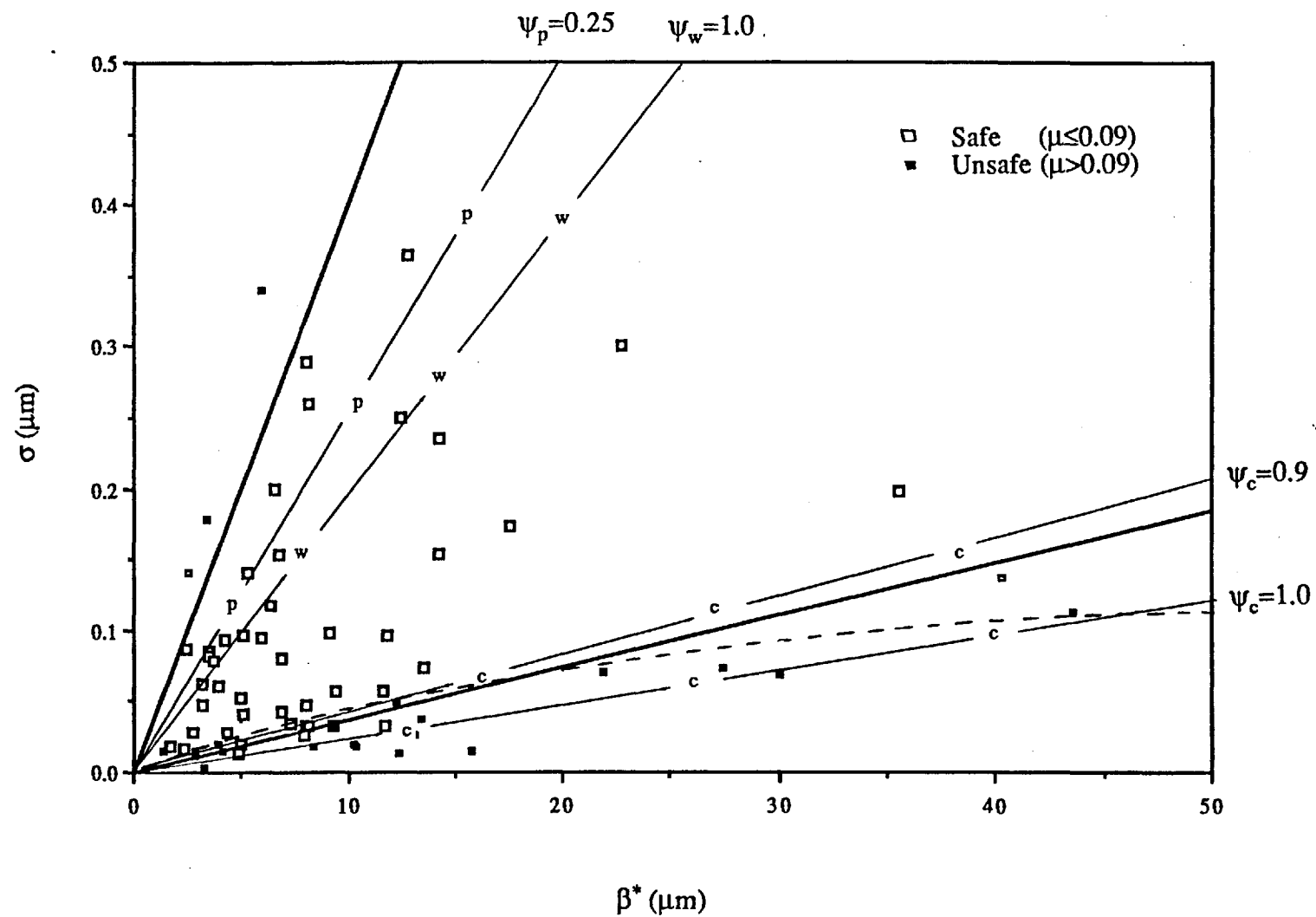


Figure 8.13 As Figure 8.11 but $T = 150^\circ\text{C}$.

8.7.3 Steel Ball on Gear Steel

Friction test results are summarised in Table 8.8. Figure 8.14 to 8.16 show the variation of friction with σ/β^* . The friction generally decreases as σ/β^* increases, and increases with temperature, but the rate of increase appears greater for surfaces with intermediate or smaller values of σ and β^* . This may be explained by the effect of three body abrasion and will be discussed later.

From Figure 8.15, a friction coefficient of $\mu=0.15$ is taken as the value typical of the boundary state. The safe and unsafe regions at different temperature are shown on σ - β^* maps in Figure 8.17 to 8.19. At each temperature whilst there are hints of a possible lower boundary. The upper boundary is not evident, within the range of σ and β^* considered.

The fact that the upper boundary does not appear may be due to the hardness of gear steel, being four times greater than that of stainless steel, reducing the level of plastic deformation. The plasticity lines of Whitehouse and Archard are also shown on the σ - β^* maps. At the regions of high σ and β^* , the plasticity index is less than 0.6, indicating that the contacts are predominately elastic.

The slope of the lower boundary appears dependent on β^* as well as σ . It is also close to the adhesion line and the region where the real and nominal area of contact are equal. In Chapter 7, it was evident that the surface damage of the steel ball sliding on a very smooth surface of gear steel was due to adhesion. Therefore, the same conclusion can be drawn; that the surface failure for a very smooth surface is due to conformity.

The slope of the boundary appears to be about the same for all three temperatures, however, as temperature increases, more unsafe points appear in the intermediate regions of σ - β^* map. This may be explained by the effect of 'three body' abrasion. The relative

Surfaces	σ (μm)	β^* (μm)	Steel ball on gear steel			Ceramic ball on gear steel		
			μ (T=50°C)	μ (T=100°C)	μ (T=150°C)	μ (T=50°C)	μ (T=100°C)	μ (T=150°C)
1	0.100	4.500	0.133	0.157	0.200	0.073	0.050	0.025
2	0.070	5.300	0.137	0.172	0.210	0.080	0.079	0.065
3	0.015	10.200	0.162	0.207	0.450	0.087	0.091	0.140
4	0.020	8.000	0.140	0.173	0.400	0.100	0.085	0.071
5	0.020	5.400	0.127	0.160	0.220	0.072	0.082	0.092
6	0.065	7.500	0.120	0.160	0.285	0.072	0.057	0.074
7	0.050	4.500	0.120	0.162	0.234	0.063	0.067	0.072
8	0.190	4.900	0.100	0.130	0.220	0.040	0.026	0.026
9	0.280	5.600	0.124	0.166	0.153	0.085	0.082	0.070
10	0.100	8.300	0.138	0.200	0.230	0.099	0.110	0.103
11	0.050	3.900	0.137	0.230	0.280	0.107	0.114	0.112
12	0.015	8.000	0.147	0.230	0.280	0.112	0.152	0.155
13	0.023	2.610	0.122	0.140	0.184	0.064	0.059	0.076
14	0.029	7.830	0.122	0.154	0.270	0.069	0.051	0.077
15	0.075	3.690	0.130	0.184	0.380	0.052	0.037	0.046
16	0.405	4.690	0.082	0.130	0.220	0.045	0.031	0.046
17	0.009	3.032	0.127	0.200	0.383	0.110	0.130	0.100
18	0.057	6.640	0.147	0.180	0.290	0.100	0.100	0.100
19	0.150	8.960	0.123	0.180	0.271	0.094	0.087	0.084
20	0.470	11.860	0.132	0.170	0.280	0.090	0.081	0.074
21	0.041	3.320	0.120	0.160	0.400	0.082	0.075	0.044
22	0.345	5.410	0.110	0.130	0.240	0.073	0.068	0.060
23	0.260	11.370	0.140	0.200	0.397	0.076	0.065	0.055
24	0.022	1.746	0.134	0.163	0.190	0.120	0.120	0.160
25	0.055	2.340	0.140	0.173	0.210	0.120	0.110	0.110
26	0.105	2.000	0.122	0.150	0.210	0.092	0.084	0.082
27	0.009	3.330	0.145	0.180	0.270	0.134	0.130	0.146
28	0.017	2.300	0.125	0.200	0.200	0.126	0.120	0.138
29	0.016	6.000	0.150	0.174	0.300	0.140	0.137	0.175
30	0.063	5.800	0.150	0.200	0.320	0.100	0.110	0.122
31	0.014	7.000	0.177	0.192	0.290	0.143	0.164	0.153
32	0.018	3.925	0.220	0.350	0.520	0.152	0.169	0.155
33	0.013	4.676	0.210	0.260	0.430	0.122	0.140	0.147
34	0.005	2.770	0.172	0.260	0.450	0.140	0.145	0.156
35	0.110	4.770	0.100	0.100	0.120	0.088	0.068	0.070
36	0.257	5.027	0.110	0.100	0.135	0.080	0.071	0.064
37	0.400	7.080	0.090	0.088	0.150	0.081	0.070	0.068
38	0.718	9.650	0.098	0.090	0.150	0.085	0.075	0.076
39	0.035	1.800	0.146	0.150	0.250	0.080	0.061	0.058
40	0.018	2.090	0.096	0.100	0.420	0.080	0.068	0.072
41	0.019	6.130	0.110	0.080	0.093	0.086	0.056	0.100
42	0.059	5.620	0.107	0.116	0.128	0.085	0.074	0.087
43	0.045	10.550	0.100	0.093	0.142	0.086	0.085	0.108
44	0.031	6.915	0.110	0.133	0.166	0.100	0.061	0.103
45	0.037	12.890	0.156	0.160	0.193	0.087	0.098	0.094
46	0.010	4.950	0.190	0.310	0.255	0.098	0.100	0.106
47	0.008	6.950	0.197	0.220	0.420	0.097	0.109	0.147
48	0.021	5.580	0.180	0.270	0.460	0.097	0.113	0.095
49	0.096	12.450	0.160	0.184	0.167	0.084	0.098	0.097
50	0.015	10.130	0.098	0.114	0.160	0.095	0.103	0.114
51	0.116	14.360	0.110	0.153	0.180	0.085	0.076	0.088
52	0.080	10.710	0.103	0.140	0.210	0.087	0.096	0.091
53	0.027	36.600	0.207	0.256	0.310	0.110	0.142	0.153
54	0.022	21.330	0.155	0.190	0.200	0.095	0.110	0.123
55	0.095	30.110	0.175	0.183	0.242	0.077	0.089	0.062
56	0.053	23.400	0.155	0.204	0.175	0.074	0.083	0.089
57	0.057	20.160	0.174	0.126	0.215	0.082	0.075	0.058
58	0.155	29.600	0.158	0.102	0.102	0.096	0.074	0.094
59	0.107	16.726	0.164	0.077	0.077	0.090	0.065	0.083
60	0.075	19.445	0.140	0.107	0.107	0.094	0.084	0.063
61	0.042	13.524	0.164	0.097	0.097	0.083	0.073	0.099

Table 8.8 Friction tests results of a steel ball or a ceramic ball on gear steel.

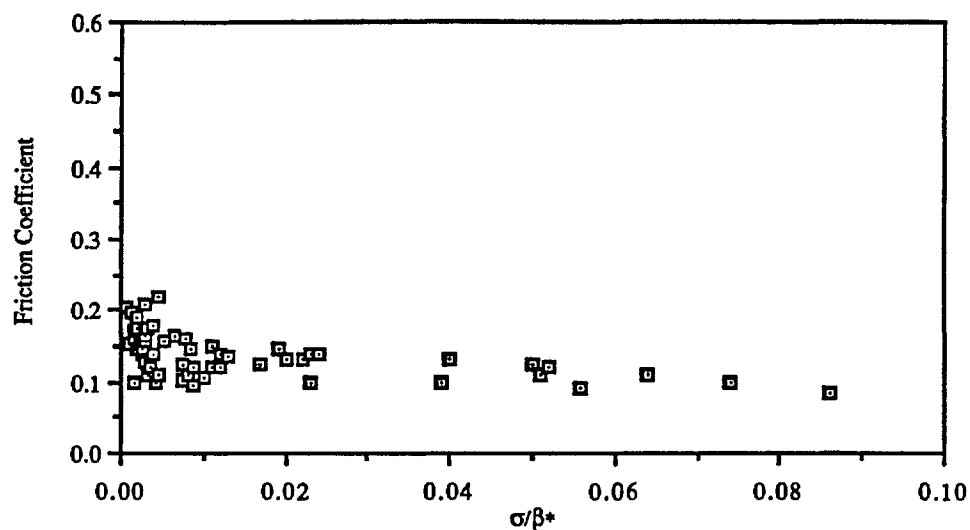


Figure 8.14 Variation of friction with σ/β^* for a steel ball on gear steel at $T = 50^\circ\text{C}$.

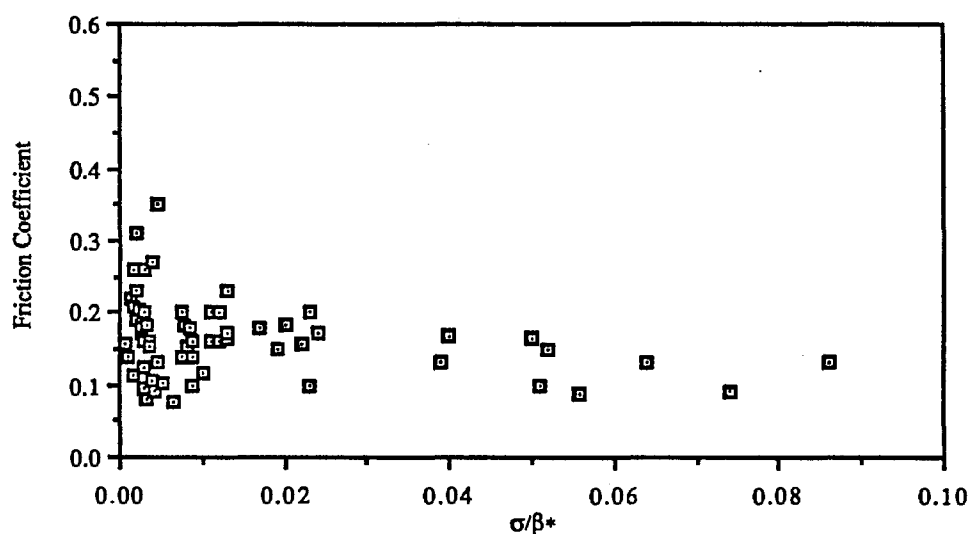


Figure 8.15 Variation of friction with σ/β^* for a steel ball on gear steel at $T = 100^\circ\text{C}$.

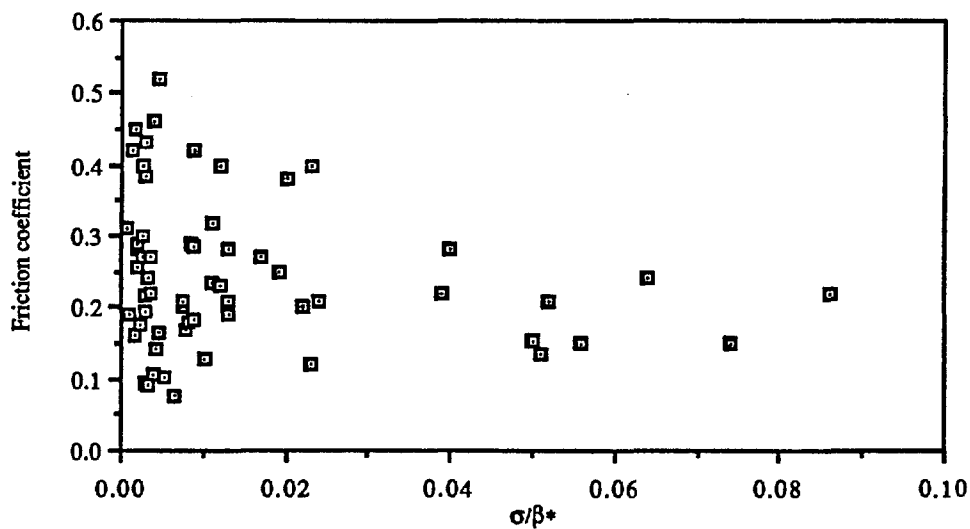


Figure 8.16 Variation of friction with σ/β^* for a steel ball on gear steel at $T = 150^\circ\text{C}$.

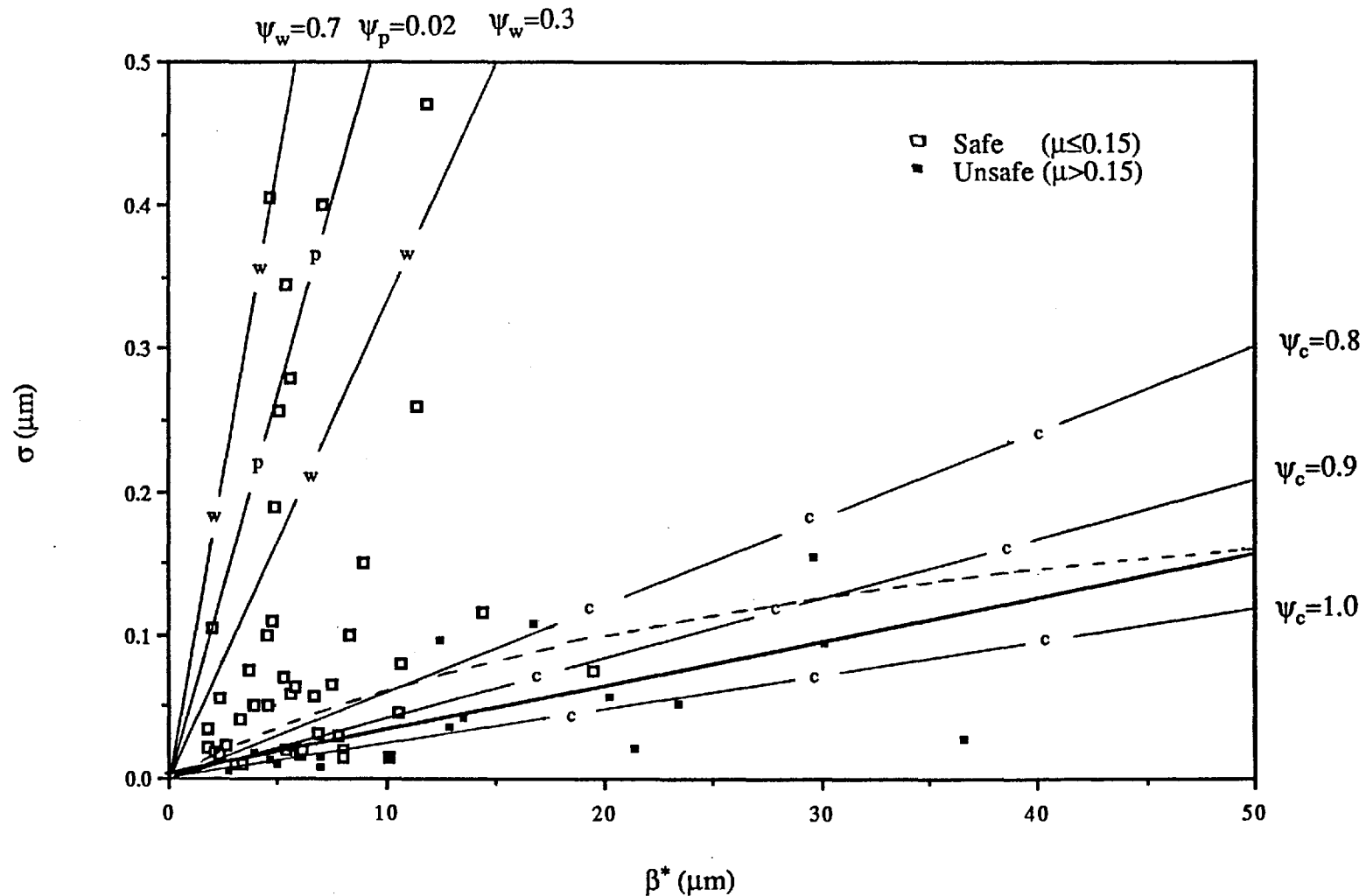


Figure 8.17 σ - β^* map of a steel ball on gear steel at $T = 50^\circ\text{C}$ showing the boundary between 'unsafe' and 'safe' regions. The lines of $\psi_c = A_r/A_n$ —c— are predicted by stochastic approach and $\psi_p = A_p/A_n$ —p— from a triangular wave contact. The plasticity lines --w-- are given by Whitehouse. The adhesion line --- is given by equation (8.9).

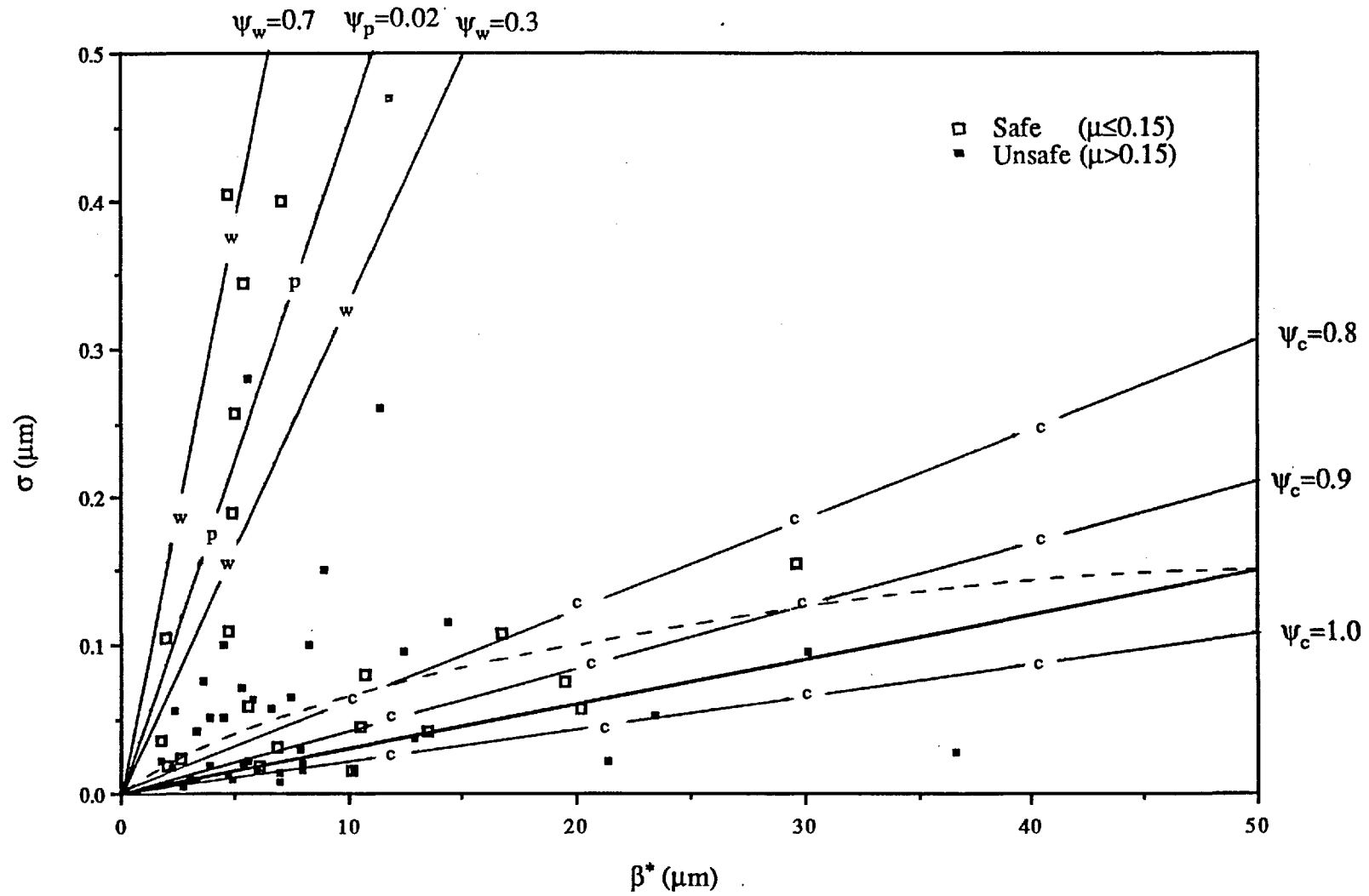


Figure 8.18 As Figure 8.17 but $T = 100^\circ\text{C}$.

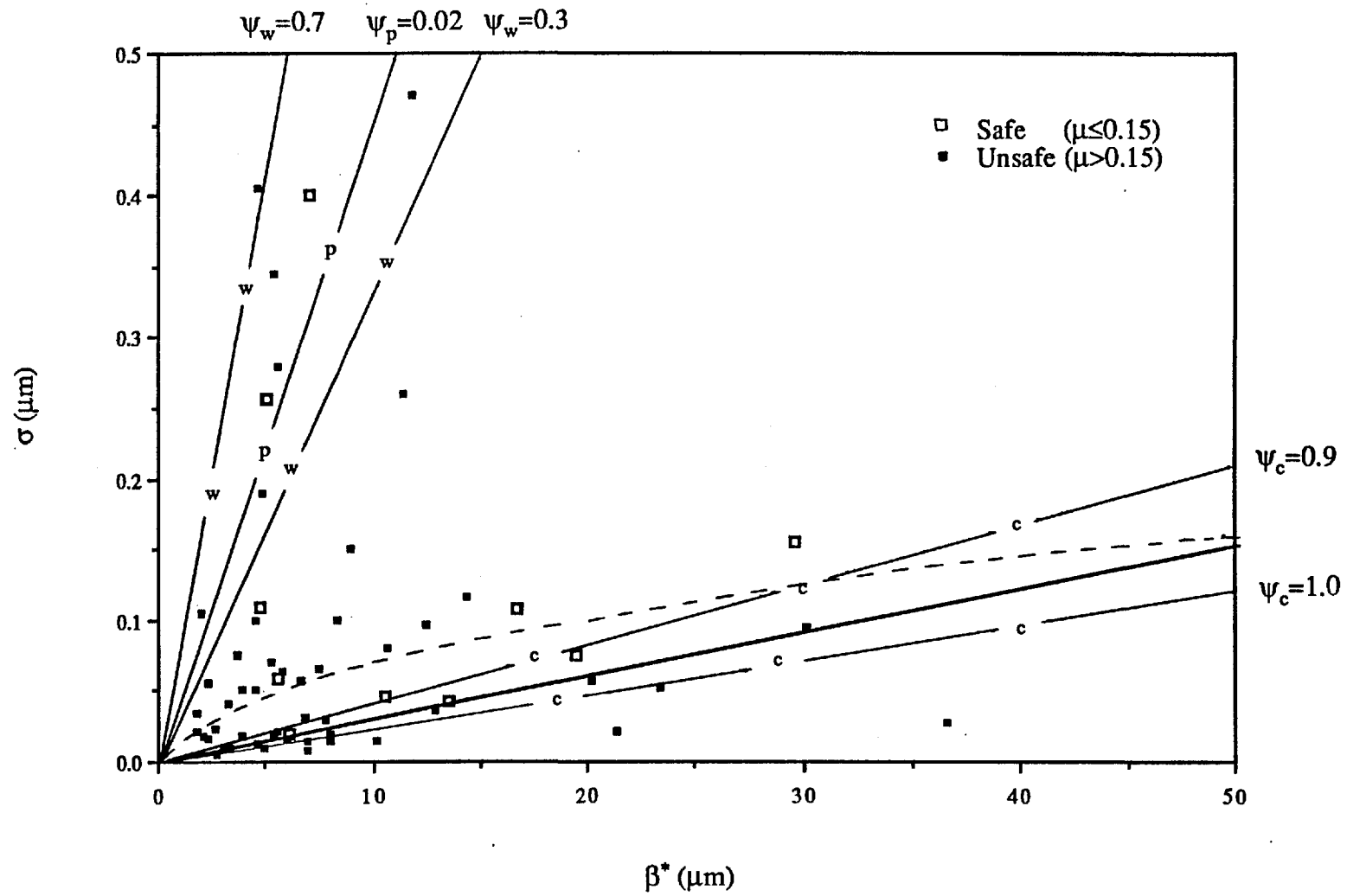


Figure 8.19 As Figure 8.17 but $T = 150^\circ C$.

hardness of the two bodies is of the same order. Loose wear particles will indent the surfaces to the same extent and may be trapped, 'micro-machining' the two rubbing surfaces for a while before they become free. At a higher temperature, the increase in friction facilitates the formation of wear particles which in turn take part in the abrasion process. For a rougher surface, the effect of wear particles is smaller due to entrapment in the scratches between contacts. For a smoother surface the effect is important and helps to explain the occurrence of the unsafe points in the intermediate regions of the σ - β^* map.

8.7.4 Ceramic Ball on Gear Steel

Friction test results are summarised in Table 8.8. Figure 8.20 to 8.22 show the variation of friction with σ/β^* . The friction generally decreases with increased σ/β^* whilst the temperature appears to have little effect on the friction level. The effect of third bodies is not significant which may be due to the fact that the friction between the interfaces is less, resulting in a smaller amount of wear particles taking part in the abrasion process.

Taking $\mu=0.09$ as a value typical of the boundary state, the safe and unsafe regions are shown on σ - β^* maps at different temperature in Figure 8.23 to 8.25. A lower boundary can be drawn to divide the safe and unsafe regions for each temperature. The slope of the boundary is about the same as that of a steel ball on gear steel. Some unsafe points, with a smaller number comparing to Figures 8.17 to 8.19, also appear in the intermediate regions of σ - β^* maps as with the steel ball/gear-steel.

8.7.5 Steel Ball on Copper

Fewer tests were performed on copper because the results, shown in Figure 8.26,

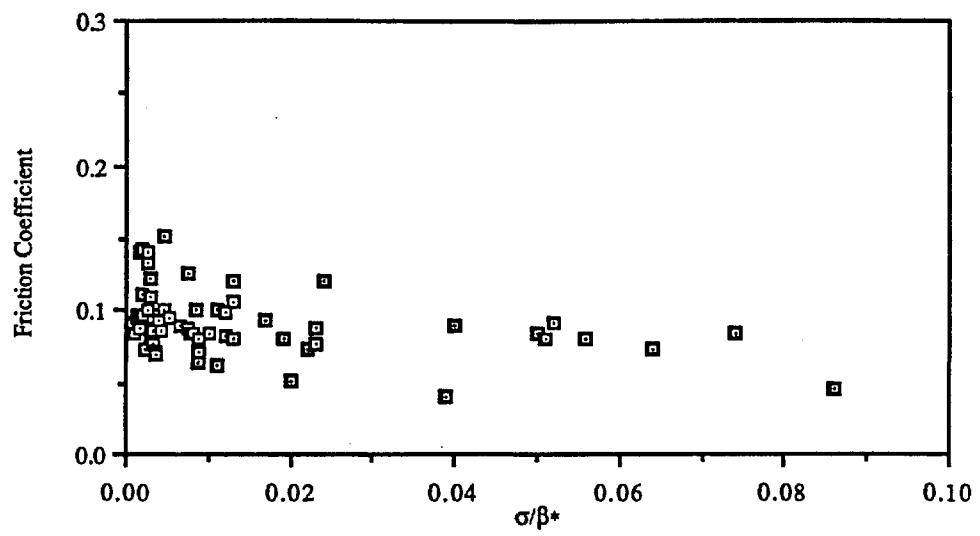


Figure 8.20 Variation of friction with σ/β^* for a ceramic ball on gear steel at $T = 50^\circ\text{C}$.

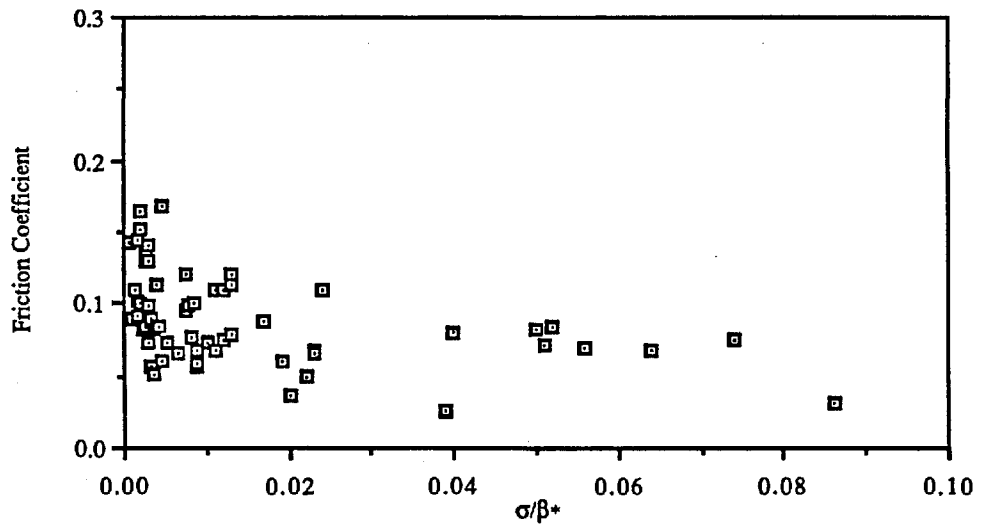


Figure 8.21 Variation of friction with σ/β^* for a ceramic ball on gear steel at $T = 100^\circ\text{C}$.

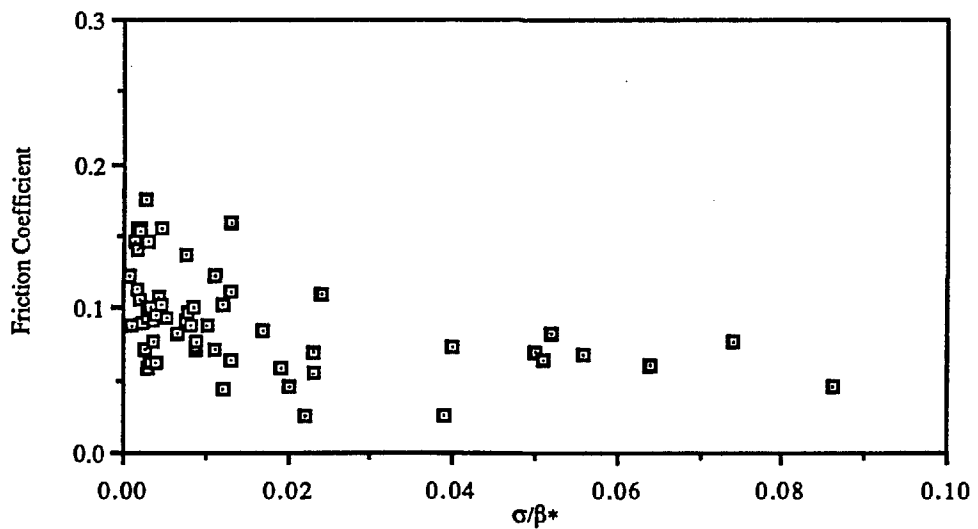


Figure 8.22 Variation of friction with σ/β^* for a ceramic ball on gear steel at $T = 150^\circ\text{C}$.

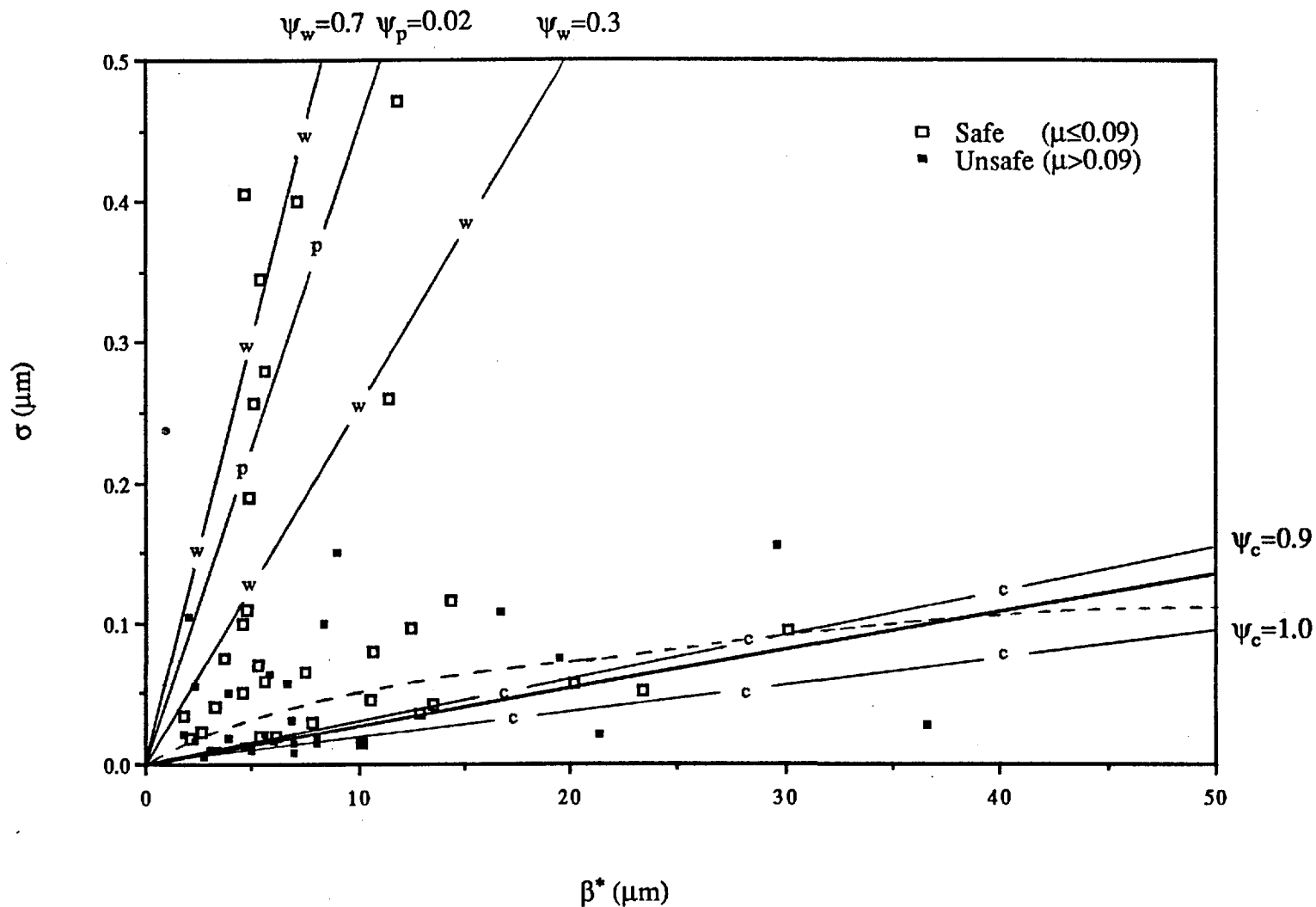


Figure 8.23 σ - β^* map of a ceramic ball on gear steel at $T = 50^\circ\text{C}$ showing the boundary between 'unsafe' and 'safe' regions. The lines of $\psi_c = A_r/A_n$ —c— are predicted by stochastic approach. The plasticity lines —w— are given by Whitehouse. The adhesion line --- is given by equation (8.10).

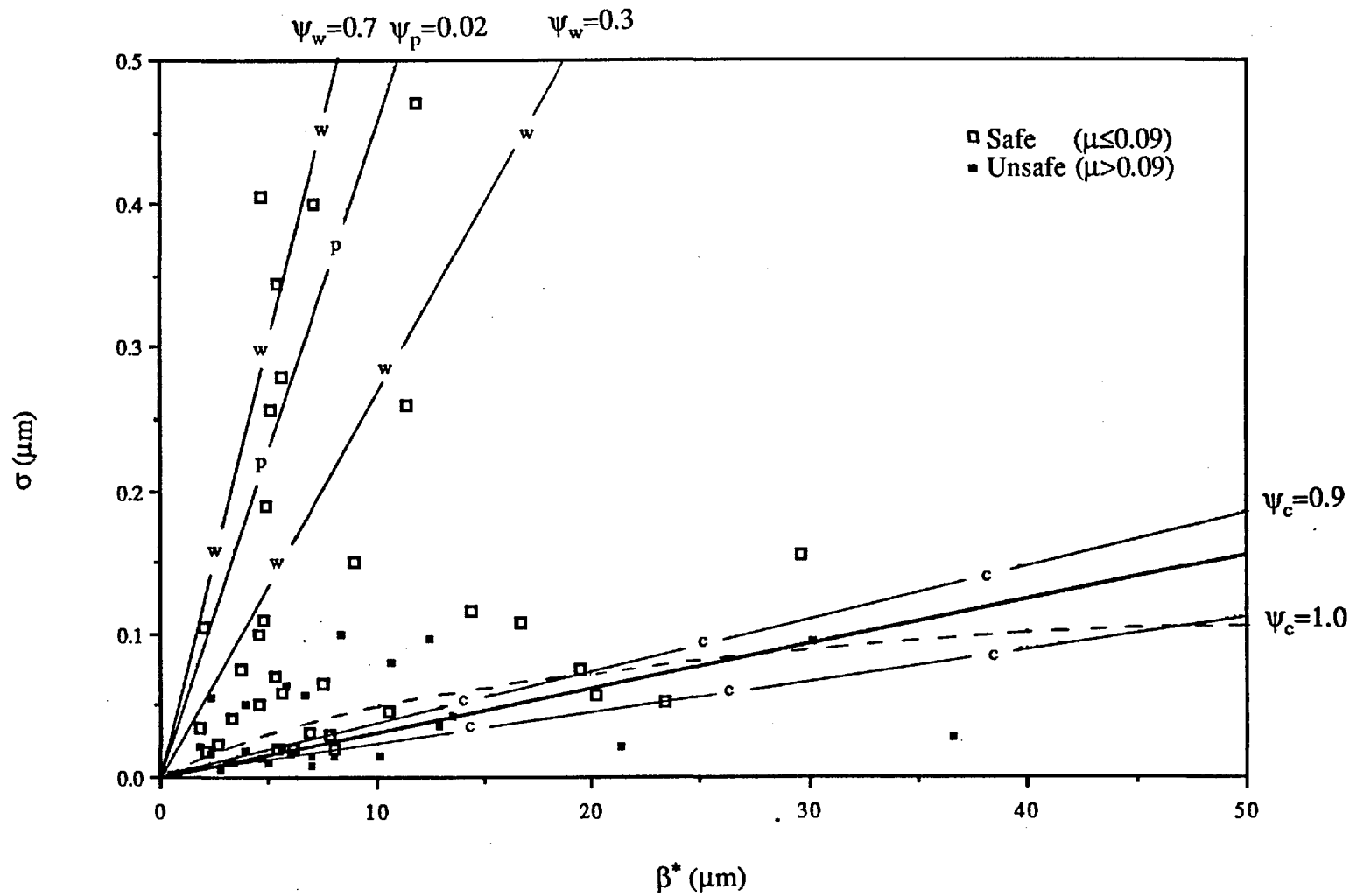


Figure 8.24 As Figure 8.23 but $T = 100^\circ C$.

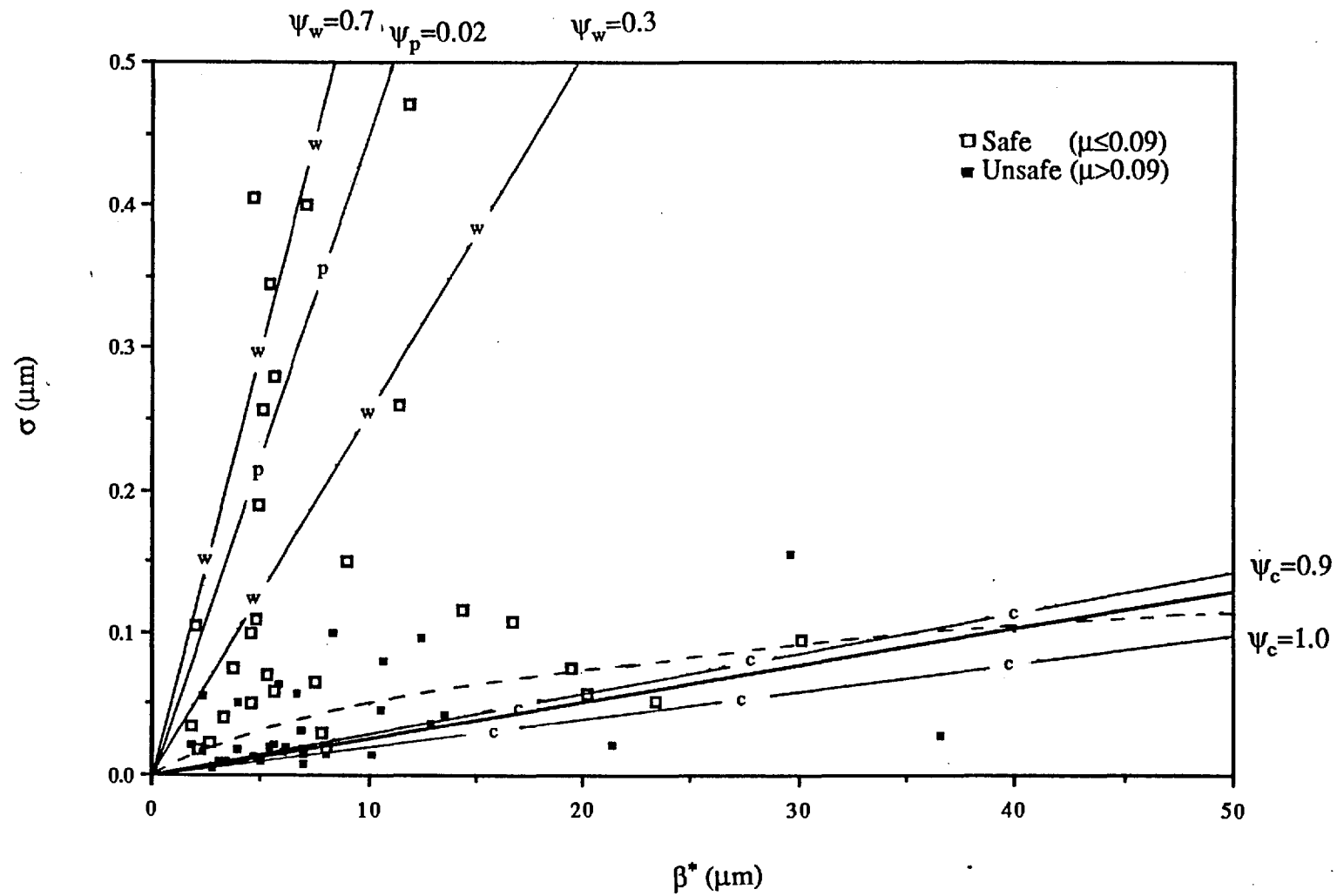


Figure 8.25 As Figure 8.23 but $T = 150^\circ\text{C}$.

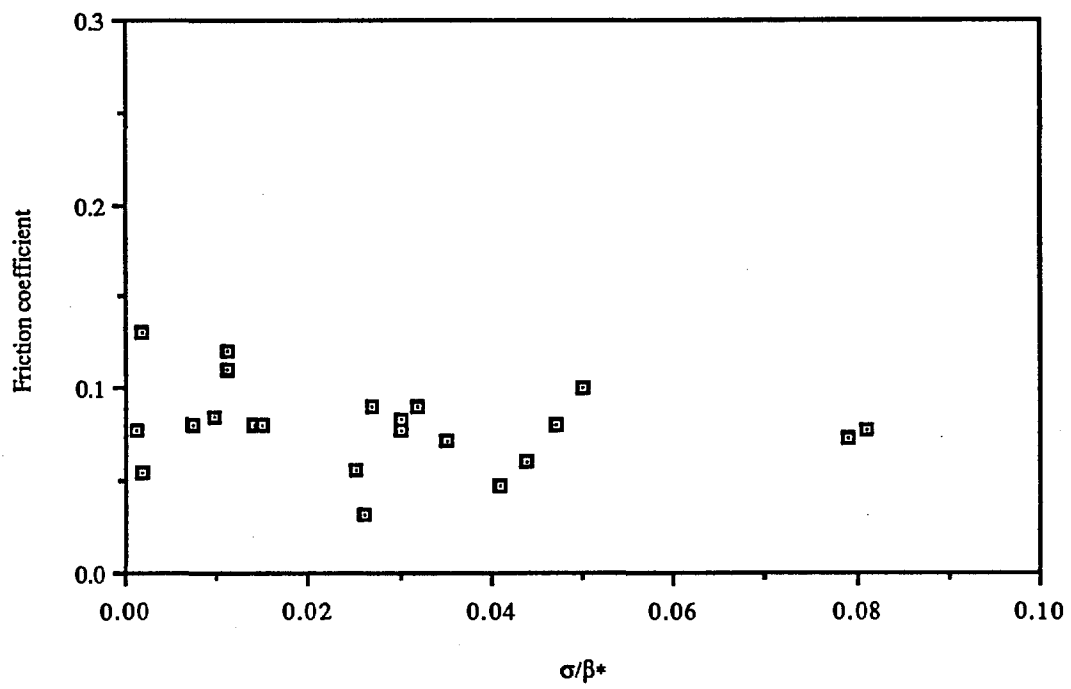


Figure 8.26 Variation of friction with σ/β^* for a steel ball on copper at $T = 100^\circ\text{C}$.

indicate that the friction does not show a positive correlation with σ/β^* and the friction remained low for a number of surfaces. Therefore it is assumed that the effect of surface roughness is not significant for copper.

The friction behaviour of a steel ball on copper in sliding contact is not surprising. In Chapter 7, it has shown that the effect of changing the surface roughness on the surface damage and friction is insignificant due to the chemical reactivity of copper on which a surface film of copper oxide is formed. The oxide film reacts with the acid-containing lubricant, stearic acid in paraffin, to form a metal soap acting as an effective boundary lubricant. In the case of copper, the oxide layer is expected to be considerably thicker and will protect the underlying metal from seizure even though the contacts are predominately plastic.

8.8 Conclusions

For a hard smooth surface sliding on a significantly softer rougher surface such as the contact of a steel ball or a ceramic ball on stainless steel, there exists an upper boundary which divides the safe and unsafe regions for sliding. The upper boundary is related to the ratio σ/β^* due to plasticity. In the case of a hard smooth surface sliding on a very hard rougher surface such as a steel ball or a ceramic ball on gear steel, the mode of contact is predominately elastic. Therefore, the upper boundary is not detected.

A lower boundary exists for the contact of a steel ball or ceramic ball on stainless steel or gear steel. The boundary is close to the region where adhesion is significant which is also related to σ/β^* when the contact almost reaches total conformity. Therefore, contact between very smooth surfaces can be defined such that little or no part of the nominal contact area remains out of contact.

The friction of a surface with an intermediate value of σ and β^* is generally low. However, the effects of third bodies can affect the frictional process. In the case of a steel ball or ceramic ball on gear steel, the effect of third bodies is significant due to the similar hardness of the two bodies. Wear particles indent both surfaces to the same extent and the abrasion process can dominate the frictional behaviour. The effect is less significant for rougher surfaces whereupon the friction is reduced. This may be due to the accommodation of wear particles in surface scratches.

For a steel ball sliding on copper, neither the upper nor the lower boundary can be identified. This can be explained, together with the high degree of plastic deformation expected with any practical surface, by the chemical reactivity of copper; when exposed to the atmosphere, a (considerable) thick layer of oxide film is formed on the surface which reacts with the acid-containing lubricant to form an effective boundary lubricant. Despite the very soft nature of copper, the plastic deformation fails to rupture or penetrate the layer of oxide film.

CHAPTER 9

FRICITION STUDY AND ANALYSIS

9.1 Introduction

Whenever two solid bodies in direct or indirect surface contact are made to slide in relation to one another, there is always a resistance to the motion, which is known as the friction force. In many instances friction is beneficial, in some circumstances we may even try to increase it and undoubtedly some aspects of life would be impossible without it. However, friction is always energy-consuming and so more often than not we endeavor to reduce it.

In 1951 G. Vogelpohl estimated that one-third to one half of the world's energy production was consumed by friction (Fuller, 1956). In addition to this the friction, invariably brings about the wear and surface damage of machine components. Consequently the study of friction is becoming increasingly important and can play an important role in energy conservation.

Friction involves two bodies sliding at their interfaces, which in the case of elasto-hydrodynamic lubrication are separated by a lubricated film. The study of friction has naturally focused on the surface composition and its mechanical, physical and chemical properties. Therefore, a large number of scientists from different fields of chemistry, physics, engineering, metallurgy etc, have been involved in the study of friction and have

made their own contribution to the theory of friction. Investigations range from microscopic studies of the molecular interaction between the ideal surfaces of two solids, to quantitative interpretation of the observed macroscopic behaviour such as found in friction induced fracture, fatigue, wear etc. Unfortunately, these theories often differs from one another.

Nevertheless most of the present theories of friction can be broadly divided into two schools of thought. The first theory claims that the physical cause of the friction force was the deformation or breaking of asperities which had become interlocked at the surface interface.

The other theory of friction claims that friction originated from breaking the interfacial adhesion junctions when sliding takes place (Bowden & Tabor 1950).

Although there are many theory of friction surrounding us, unfortunately, there is little experimental evidence to support any particular frictional model. This is probably due to the simplicity of most models where the expressions are based upon just one particular frictional process such as shearing of adhesion junctions, interlocking or ploughing of asperities. In reality, the mechanistic process of friction may involve several components, such as rupture of adhesion, interlocking by wear particles or surface asperities, displacement of material by ploughing and shearing, detachment of displaced material, fatigue and cracking, etc, either sequentially or simultaneously.

Without the knowledge of the mechanistic behaviour of one surface sliding on the other, it is difficult to predict the friction force of certain pairs of material. However, as Henri Poincare (1905) said, " It is better to predict without certainty than never to have predicted at all.' Although friction is a very complex phenomenon, some of the observable trends in frictional behaviour may be comprised of orderly and predictable combinations

and sequences of basic mechanisms which in turn depend upon the conditions of the sliding interface, the specific material used, the surface topography and the environment and the lubricant regime.

The lubricant regimes may be divided into five different types, namely, in the order of decreasing lubricant thickness,

- (1) Hydrodynamic
- (2) Hydrostatic
- (3) Elasto-hydrodynamic
- (4) Boundary lubrication
- (5) Dry friction

The first three regimes usually have extremely low friction and wear because the lubricant film is sufficiently thick to completely separate the two surfaces. The friction force is then dependent only on lubricant properties, such as viscosity, compressibility, density and specific heat capacity. The last two regimes involve the contact of two bodies. The friction force is then dependent upon the surface topography, interfacial film properties, materials properties etc. In many engineering processes, contacts operate for part of the time under boundary lubrication. It is well known that in a reciprocating engine, the piston velocity approaches zero near the top dead centre where boundary lubrication is at work and this is where the surface damage is most severe. In order to improve engineering surface performance, an understanding of factors affecting friction behaviour in boundary lubrication is essential.

In this chapter, factors affecting friction behaviour in boundary lubrication are discussed. Friction theories are reviewed and an attempt is made to correlate results obtained from friction tests of a steel or ceramic balls rubbing against a stainless steel, gear steel or copper counterface.

9.2 General Consideration of Friction in Boundary Lubrication

9.2.1 Adhesion

When one body slides on another body, adhesive transfer sometimes takes place particularly when surfaces are clean. There is considerable evidence that adhesion is one of the processes in friction (Bowden and Tabor 1950) as has been shown in Chapter 7.

However, it is a common experience that no adhesion force is observed when two metals are pushed together. Tabor suggested two reasons why the normal adhesion is not detected. Firstly, the surface are covered with adsorbed films even when the surfaces have been carefully cleaned in the atmosphere. Secondly, which is more important, the adhesion junctions formed between the lower asperities on loading are elbowed apart on unloading by the compression between the higher asperities. In this way adhesion developed at the points of real contact is progressively broken down. For materials with high modulus, the compression is more significant than the adhesion and therefore adhesion is not detected.

Johnson et al (1971) investigated the contact between a glass sphere and flats of rubber and gelatine which are low modulus materials. They found good agreement between the experimental values of adhesion force and the predictions of their theory. Fuller and Tabor (1975) later combined Johnson's theory with the Greenwood and Williamson (1966) model of a rough surface in a study of the adhesion between elastic solids and the effect of roughness in reducing the adhesion. They showed the adhesion force between the interfaces is not significant unless:

$$\psi_a = \frac{E'\sigma^{\frac{3}{2}}}{R^{\frac{1}{2}}\Delta\gamma} < 10 \quad (9.1)$$

where ψ_a is the adhesion index, E' the reduced elastic modulus of two bodies, σ the surface roughness, R the radius of asperity curvature and $\Delta\gamma$ the surface energy.

9.2.2 Junction Growth

If the frictional force is assumed to break the asperity adhesions each adhesion is equal to the shear strength of the bulk metal times the contact area A , so the friction force is

$$F = A \cdot \tau_m \quad (9.2)$$

For plastic deformation, the area is given by the load W divided by the flow pressure p_m of the material. Then equation (9.2) becomes,

$$F = \frac{W}{P_m} \cdot \tau_m \quad (9.3)$$

The friction coefficient is

$$\mu = \frac{F}{W} = \frac{\tau_m}{P_m} \quad (9.4)$$

For an ideal plastic material, p_m is at least five times τ_m , so the coefficient of friction should fall in the vicinity of $\mu=0.2$. But in fact the coefficient of friction observed in the unlubricated sliding of most ordinary metals in air falls in the range 0.5 to 1.6 (Dorinson & Ludema, 1985).

To avoid the restrictions imposed by a limiting value of 0.2 for the ratio τ_m/p_m , Tabor (1959) developed a model for the friction of solid bodies based on the theory of

plastic flow under combined normal and tangential stresses using Huber-von Mises criterion for idealised plastic flow in the form,

$$\sigma_n^2 + \alpha \tau_i^2 = p_m^2 \quad (9.5)$$

where σ_n is the normal stress, τ_i is the interfacial shear stress, α is an empirical factor and p_m is the uniaxial flow stress of the bulk metal. Tabor argued that p_m is a constant characteristic property of the metal, so σ_n must decrease when τ_i increases. Therefore if the total load W remains fixed throughout the deformation process, a decrease in σ_n means that the real area of contact must increase resulting in junction growth.

Tabor suggested that $\alpha=9$ is reasonably representative of the metals usually encountered and setting $p_m=3\tau_m$, he obtained,

$$\sigma_n^2 + 9 \tau_i^2 = 9 \tau_m^2 \quad (9.6)$$

Writing $\tau_i=k\tau_m$, where k lies between 0 and 1, the friction coefficient is

$$\mu = \frac{\tau_i}{\sigma_n} = \frac{1}{3\sqrt{k^{-2} - 1}} \quad (9.7)$$

It can be seen that as k approaches 1, μ approaches infinity. When $k=0.95$, $\mu=1.014$ which is about the value obtained for clean metal-metal contact.

The significant increase of friction is because the combined stresses causes the asperity to slump and the junction to grow thus necessitating the breakage of a much larger contact than a simple normal stress would produce. Tabor also provided experimental evidence which agreed with the junction growth theory.

9.2.3 Effect of Contaminant Films

Practically all published work agrees that extremely small amounts of contaminants, especially oxygen, can greatly reduce the adhesion between metals. (Buckley 1981; Bowden & Tabor 1950). The extent to which contaminated films reduce the adhesion depends on the nature of the films, the geometry of the surface and whether surface deformation is sufficient to cause penetration of the film. Tabor has studied the effect of contaminated films on friction by performing tests in which the adsorbed film of oxygen and other contaminants are removed from metal surfaces such as nickel/tungsten and copper/copper. He concluded that the coefficient of friction of the surfaces at the beginning of the experiments lies around $\mu=0.5$ and after prolonged heating in vacuo at bright red heat and subsequent cooling, the friction increases to values lying between $\mu=4.5$ and $\mu=6$. The friction was reversible after a trace of oxygen was admitted to the surface. The results show that the removal of the surface films leads to a very great increase in the friction. This is due largely to the ease with which the metals surfaces can weld together in the absence of contaminant films.

This surface film effect has a parallel in the way interfacial films are modified by the diffusion of minor constituents or of adventitious impurities to the grain boundaries. This often leads to a marked reduction in strength and often to embrittlement.

In the case of boundary lubrication, the surface layer of metal oxide will react with acid-containing lubricants, such as paraffin with 1% stearic acid, to form a metal soap in the interfaces which protects the underlying metal and reduces friction.

9.2.4 Effect of Roughness

The influence of surface roughness on surface contact has been investigated in detail in Chapter 4 and 5. The experimental study of the effect of roughness on sliding friction and surface damage is also investigated in Chapter 7. Basically, there are three forms of surface contact induced by the surface roughness. For a rough surface, the contact will be predominantly plastic. The friction and surface damage are usually high except where the boundary lubricated film works effectively. For a surface with intermediate surface roughness, the contact will be predominantly elastic. The friction and surface damage are small. For a very smooth surface, the contact will be in a state where the total area approaches the nominal area such that no part of the nominal contact area remains out of contact. The adhesion force in the interface is significant and the friction and surface damage are significantly higher.

9.2.5 Effect of Temperature

When one solid body slides over another most of the work done against the frictional force opposing the motion will be dissipated as heat between the surfaces. The rise in surface temperature is dependent on speed and load for a fixed environmental condition.

The effect of temperature on friction can be looked at from two points of view. Firstly, temperature effects originate as a consequence of the rubbing process. Secondly, temperature is part of the ambient environment. This difference governs the way the influence of temperature is analysed. The ambient temperature in some instances, can be controlled to reduce the damage, but for the rubbing process, the temperature changes are an intrinsic property. The resulting interfacial temperature at equilibrium is a function of

specific parameters such as the coefficient of friction, the loading force, the velocity of sliding, the dimensions of the interface, the properties of the materials etc.

Theoretical (Jaegar 1942; Archard 1958/59; Ling & Pu 1964) and experimental (Loewen & Shaw 1954; Sakurai 1974; Furey 1964; Bowden 1954; Deyber & Godet 1971) studies show that the local surface temperature can reach very high values even under moderate conditions of load and speed. Experiments show that temperatures are confined to a very thin layer of the bodies and are localized at a number of very small points. These high temperatures occur throughout the lubrication film at the points of most intimate concentrated contact. This is of great significance in the study of boundary lubrication since the local high temperatures may produce a marked change in the lubricity properties of the lubrication film at the very points where the surfaces are rubbing. In addition, the high temperature may cause volatilisation and decomposition of the lubricant.

9.3 Friction Theory

9.3.1 Laws of Friction

The two basic laws of friction are:

- (1) Friction is proportional to the normal force between surfaces.
- (2) Friction is independent of the apparent area of contact.

They were first deduced by Leonardo de Vinci (1519) and discussed by Amonton (1699). Coulomb (1785) verified these laws experimentally. The laws of friction have remained unchallenged for more than 400 years. The modern experimental research seems to support them. Even the validity of contacting models are based upon their ability to explain the laws of friction.

Coulomb has discussed the origin of friction and was originally inclined toward the view that friction is produced by molecular adhesion between the interacting surfaces, which is somewhat in line with present day theories. However he later rejected this because he felt that this would make friction proportional to contact area rather than to load. Instead, he favored the view that friction was simply the force required to lift the load over the interlocking asperities.

The early theories of friction, adhesion or interlocking of asperities, are based upon the assumption that the resistance to sliding is due to the surface forces only. Bowden and Tabor have shown experimentally that the friction effects are not confined to the surface of the solid but cause distortion and deformation to a very great depth beneath it. Obviously, the physical processes that occur during sliding are too complex to yield easily to a simple mathematical treatment. However, Bowden and Tabor (1950) have shown that if two metal surfaces are in contact, a localised adhesion and welding of the metal surfaces occurs when sliding takes place. Work is then required to shear these welded junctions.

9.3.2 Friction Models

As mentioned above, there is no "simple" model to predict friction of a given pair of materials. The coefficient of friction between sliding surfaces is due to various combined effects of asperity adhesion, ploughing, plastic deformation, elastic deformation and elastic hysteresis between the surfaces in contact. The relative contribution of these components depends on the conditions of the sliding interface, which in turn is affected by the history of the sliding, the specific materials used, the surface topography and the environment.

9.3.2.1 Adhesion Component of Friction

The adhesion component of friction is due to the formation and rupture of interfacial adhesive bonds. It is assumed that metallic junctions are formed at the regions of intimate contact. The total force F to shear all the junctions is given by,

$$F = A \cdot \tau \quad (9.8)$$

where A is the real area and τ is the mean tangential stress required to shear these junctions. If there are different kinds of asperity junctions, The total friction force is the sum of all the tangential forces involved in the deformation and rupture of the asperity junctions, for which the generalised expression is written as,

$$F = A_1 \tau_1 + A_2 \tau_2 + \dots + A_n \tau_n \quad (9.9)$$

where A_k , τ_k are the area and the shear strength of each kind of asperity junction.

9.3.2.2 Ploughing Component of Friction

If one surface in a sliding process is considerably harder than the other the harder asperities may penetrate into the softer surface. In a tangential motion a force results because of the ploughing resistance. This may contribute considerably to the friction resistance. There are three basic ploughing motions. One is the ploughing by asperities and the other by wear particles penetrating into the interface. The third is by plastic deformation

In the simplest model of a sliding conical asperity, the friction coefficient is given by, (Rabinowicz 1965)

$$\mu = \frac{2}{\pi} \tan \theta \quad (9.10)$$

where θ is the slope of the ploughing asperity. Surface asperities seldom have an effective slope exceeding 5° or 6° , so it follows that the friction coefficient should have a value of $\mu \sim 0.05$. This value may be regarded, however, only as a lower friction coefficient limit for the ploughing component of friction as the model neglects the experimental observations that a pile-up of material ahead of the grooving path occurs in most cases of ploughing during sliding (Landheer 1974).

Zum Gahr (1981) considered the micro-cracking of brittle material as a result of ploughing. He extended the model using fracture mechanics and related the ploughing component to properties of materials like the fracture toughness, K_f , the elastic modulus E and the hardness, H

$$\mu_p = c \frac{k_f^2}{E \cdot (H \cdot W)^{\frac{1}{2}}} \quad (9.11)$$

where c is a constant and W is the normal load.

The ploughing by penetrated wear particles was investigated by Suh and co-workers (1979), the ploughing component is given by,

$$\mu_p = \frac{2}{\pi} \left\{ \left(\frac{2r}{w} \right)^2 \cdot \sin^{-1} \frac{w}{2r} - \left[\left(\frac{w}{2a} \right)^2 - 1 \right]^{\frac{1}{2}} \right\} \quad (9.12)$$

where r is the radius of the wear particle and w is the contact width of the penetrated particle.

Their analysis showed that the contribution of ploughing to friction is very sensitive to the ratio of the radius of curvature of the particle to the depth of penetration. This implies that the materials properties and the geometric properties of the asperities or penetrated wear particles may considerably influence the friction behaviour of sliding surfaces.

Because of the deformation during sliding contact, mechanical energy may be dissipated through plastic deformation or elastic hysteresis effects. Green (1955) analysed the deformation of the surface asperity contact using the slip-line field for a rigid-perfectly plastic material. Since then, slip-line theory (Drescher (1959), Challen and Oxley (1979), Heilmann (1981)) has been used to obtain expressions for friction. Among these models, Challen and Oxley provided a promising relationship of friction due to plastic deformation. They showed that for a hard wedge with an angle α along the sliding direction on a soft surface, the friction coefficient is given by,

$$\mu = \frac{A \sin \alpha + \cos(\cos^{-1} k - \alpha)}{A \cos \alpha + \sin(\cos^{-1} k - \alpha)} \quad (9.13)$$

where

$$A = 1 + \frac{\pi}{2} + \cos^{-1} k - 2\alpha - 2\sin^{-1}\{(1 - k) - \frac{\sin \alpha}{2}\} \quad (9.14)$$

and k is the ratio of the shear strength of the film τ_i to the shear flow stress of the soft material τ_m , i.e. $k = \tau_i / \tau_m$ where $0 \leq k \leq 1$.

Moalic & Torrance(1987) performed friction tests on surfaces with multi-asperities with a range of profile slopes. The results were shown to be in agreement with the theory derived by Challen and Oxley provided a suitable value of k was used.

9.4 Friction Analysis

Friction force is the combined effect of adhesion, ploughing and deformation. The friction coefficient can be written as,

$$\mu = \mu_a + \mu_p + \mu_d \quad (9.15)$$

where the suffix a, p and d represent the components of adhesion, ploughing by either asperities or penetrated wear particles and deformation respectively. The relative contribution of these components depends on the environmental conditions of the sliding interface and materials used. One component might be more important so that the other components can be considered as small.

The adhesion components are thought to have come from the shearing of the contacting junctions. If the type and area of contacting junctions are known, the adhesion component can be calculated. In boundary lubrication, the adhesion can be considered as arising from shearing of the interfacial lubricated film. In the case of plastic deformation, the rupture of the lubricated film makes the contact of the underlying material possible and the adhesion component should then include the shearing of the metal-metal junctions. Thus, the adhesion force F_a can be written as,

$$F_a = A_m \cdot \tau_m + A_i \cdot \tau_i \quad (9.16)$$

where A_m , τ_m are the area and shear stress of the bare material and A_i , τ_i are the area and shear stress of the interfacial lubricated film. μ_a is given by,

$$\mu_a = \frac{F_a}{W} = \frac{A_m \tau_m + A_i \tau_i}{W} \quad (9.17)$$

or

$$\mu_a = \frac{\frac{A_m}{A_n} \tau_m + \frac{A_i}{A_n} \tau_i}{p_n} \quad (9.18)$$

where W , A_n , p_n are the load, nominal area and nominal pressure respectively.

The real area and plastic area of contact at a constant load can be predicted from numerical contact analysis. This has been shown to be dependent on the values of σ and β^* . Therefore, μ_a can be plotted as a function of σ/β^* based upon the results from contact analysis. From friction tests on various materials, the frictional behaviour is found also to be dependent upon the values of σ and β^* . It is instructive to compare the experimental results with those predicted from the adhesion component of friction. This will be presented later.

The ploughing component mainly comes from ploughing by asperities and ploughing by penetrated wear particles. On most engineering surface, the slope of asperities are relatively small and the asperity ploughing force can be neglected. In contrast the ploughing by penetrated wear particles can be catastrophic and may become the main force opposing the motion. Although equation (9.12) gives the friction force for a given size of particle, without the knowledge of the shapes and sizes of wear particles, and how they take part in the frictional process, the determination of ploughing force by wear particles is difficult. However, the interlocking effect of wear particles is probably small if one of the sliding bodies is much softer than the other and the wear particles can penetrate deeply inside the softer material.

The deformation component is due to energy dissipated through plastic deformation and elastic hysteresis. The plastic deformation force is usually calculated by slip-line

theory, in which Challen's expression for friction shows good agreement with experimental results obtained by Challen and Oxley using a single asperity surface, and Moalic using a multi-asperity surface. It is instructive to know if it also applies to random surfaces. Unfortunately, Challen's model can only be applied if one of the surface is much softer than the other, so among the contact pairs in the friction tests, only the results from a steel ball or a ceramic ball on stainless steel or copper can be used to compare with Challen's model.

9.4.1 Steel Ball on Stainless Steel

The adhesion component can be obtained by equation 9.18. The value of τ_m and τ_i must be known. Using von Mises' shear strain-energy criterion, τ_m is $y_s/\sqrt{3}$, where y_s is the yield stress of the material. For stainless steel, $y_s = 0.533$ Gpa and therefore $\tau_m = 0.308$ Gpa. The interfacial shear stress can be calculated from the friction curve in friction tests the value of σ/β^* is such that the contact is predominantly elastic and the adhesion can be assumed to arise from shearing at the junctions of the interfacial lubricated film. Comparing the variation of friction and contact area with σ/β^* shown in Figures 9.2a and 9.1, an appropriate value of σ/β^* can be taken as 0.006, where the value of A_p/A_n close to zero, from which $\mu=0.1$ and $A_r/A_n=0.85$. The nominal pressure of the Hertzian contact is $p_n=0.562$ Gpa at a load of 10 N. Putting $A_i=A_r$ and $A_m=0$ in equation (9.18), the interfacial shear strength is $\tau_i=0.066$ Gpa.

There is now a need to calculate the friction coefficient as a function of σ/β^* . For small σ/β^* , the contact will be predominantly elastic and it is reasonable to assume the friction is due to breaking the interfacial junction. The friction coefficient is

$$\mu_a = \frac{\tau_i A_r}{p_n A_n} \quad (9.19)$$

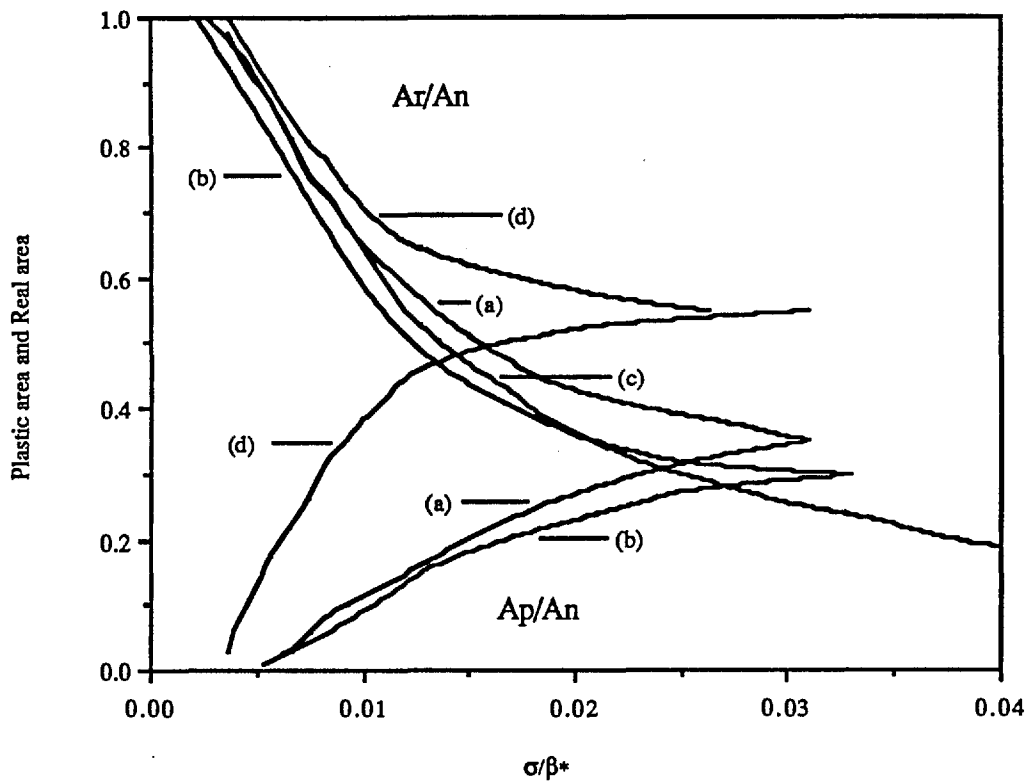


Figure 9.1 Variation of real area and plastic area with σ/β^* predicted by triangular contact. Load=0.56Gpa.
 (a) Steel ball on Stainless steel
 (b) Ceramic ball on stainless steel
 (c) Steel ball on gear steel
 (d) Steel ball on copper

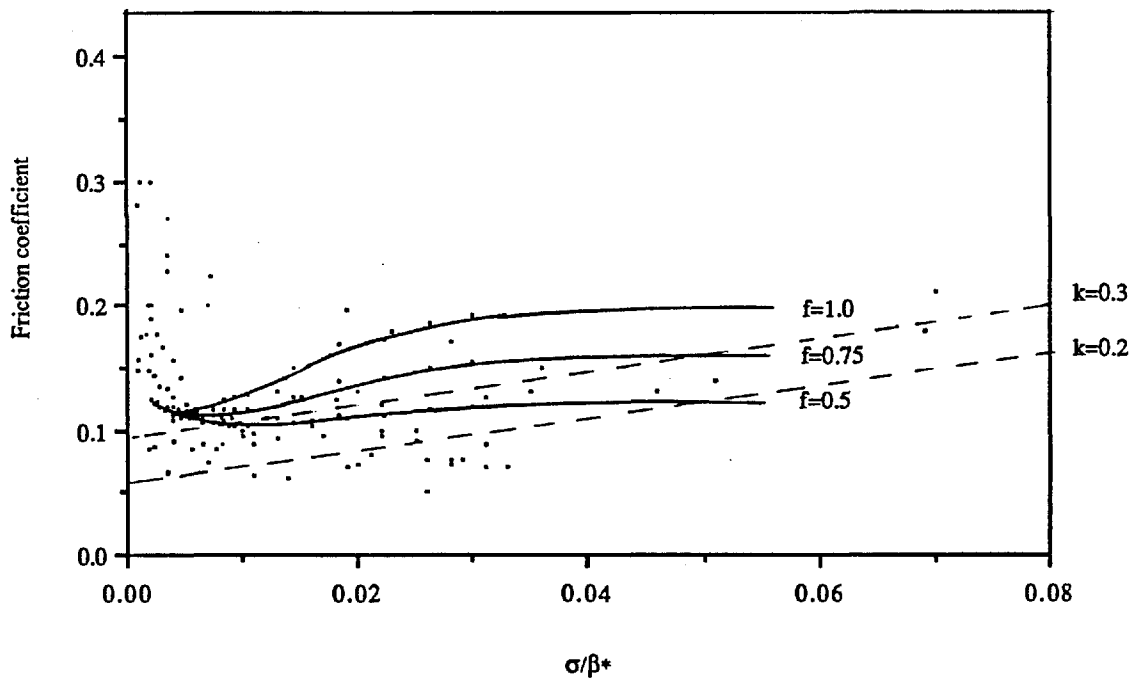


Figure 9.2(a) Variation of friction with σ/β^* for a steel ball on stainless steel at $T = 50^\circ\text{C}$. The lines — are predicted by adhesion friction. The lines --- are due to Challen.

As surface roughness increases, the contact will undergo plastic deformation causing rupture of the oxide film and initiation contact of the bare metals. The extent of this is obviously dependent upon the plasticity and thickness of the oxide film but it is expected that the area of the metal-metal junctions increases with plastic area. Owing to the presence of the oxide film, it is assumed a fraction f of the plastic area will be the area of the metal-metal junction. The remaining area is the contact of the interfacial film. Thus,

$$A_m = f \cdot A_p \quad (9.20)$$

and

$$A_f = A_r - f \cdot A_p \quad (9.21)$$

where A_r and A_p are the total area and plastic area of contact. The variation of A_r/A_n and A_p/A_n with σ/β^* can be obtained from Figure 9.1. Using equation (9.18), the plots of μ with σ/β^* for different values of f are shown together with experimental results in Figures 9.2 and 9.3. For intermediate value of σ/β^* , the prediction follows the general trend of the experimental results provided that a suitable value of f is used. The appropriate value of f is 0.5, 0.75 and 0.9 at $T=50^\circ\text{C}$, 100°C and 150°C respectively.

For small and large σ/β^* , friction values from experimental results are always higher than those predicted by equation (9.18). This could be explained by the assumptions which are made in the derivation of equation (9.18) and a junction-growth element as the area of contact used here is based on the frictionless contact model of two bodies in a static condition. If the frictional force is increased, the contact area will be increased as a result of the combined effect of normal stress and tangential shear stress. In the case of a large σ/β^* , the situation is closely related to the junction-growth theory which predicts friction of solid bodies under plastic-flow. The friction coefficient can be calculated by equation (9.7)

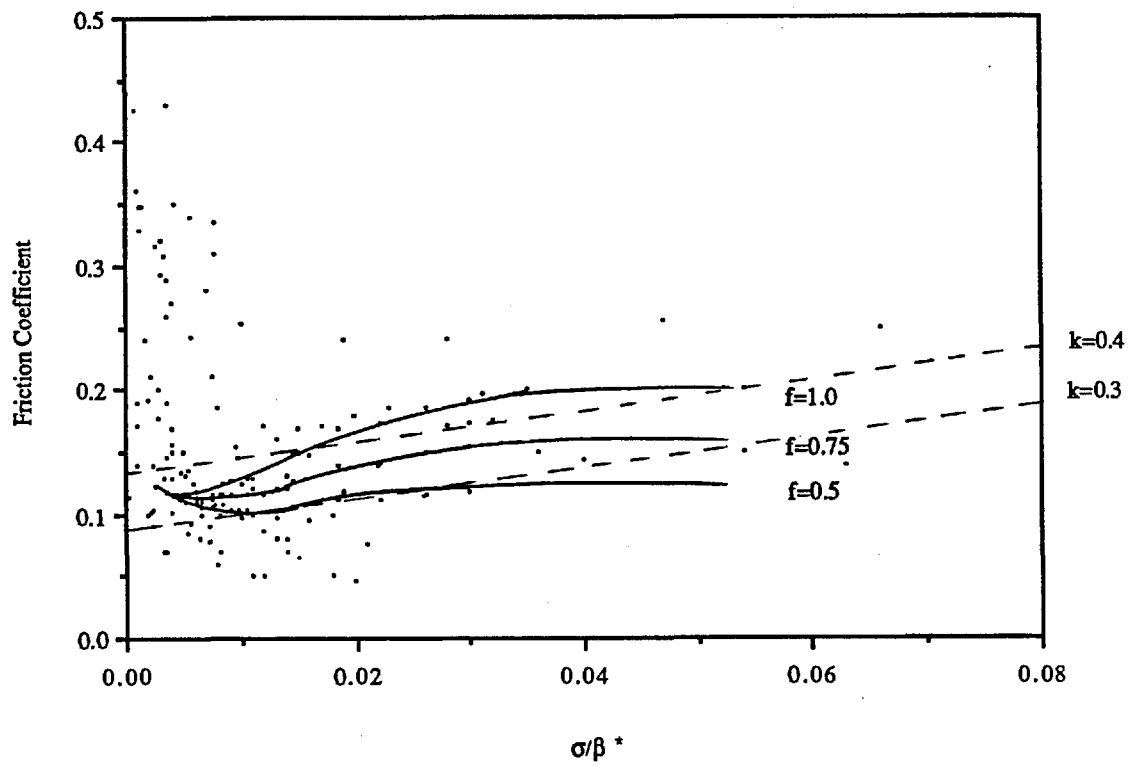


Figure 9.2(b) As Figure 9.2(a) but $T = 100^\circ\text{C}$.

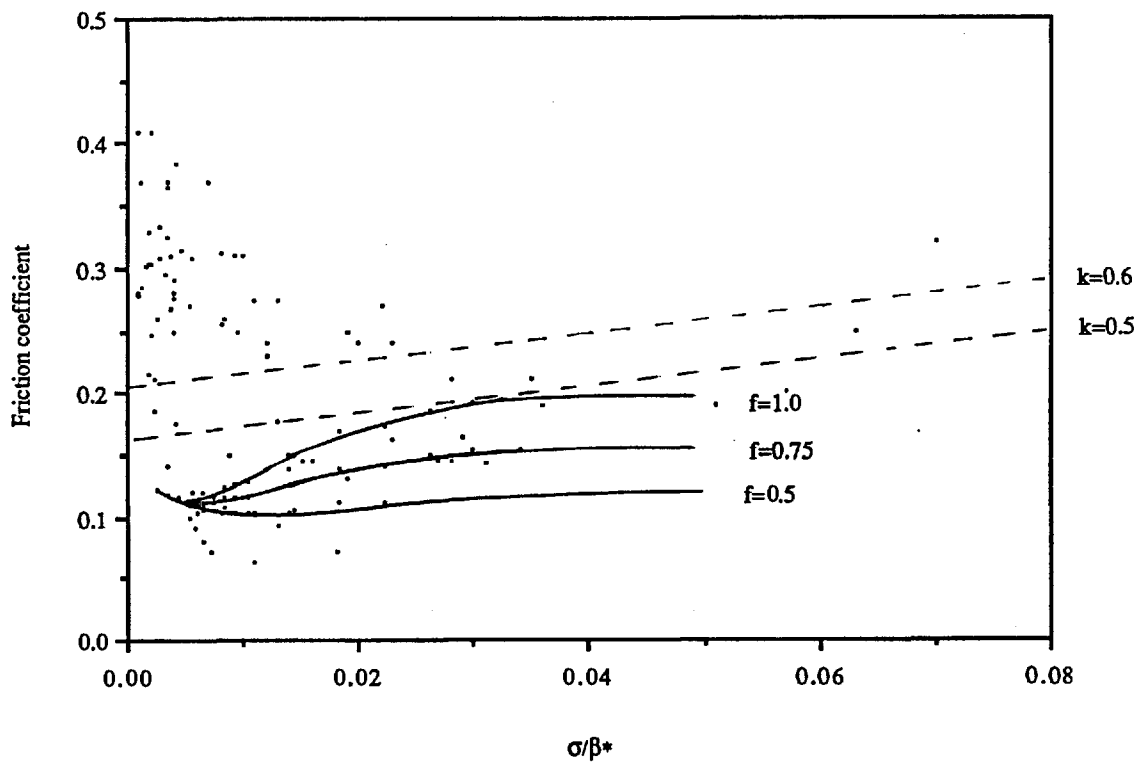


Figure 9.2(c) As Figure 9.2(a) but $T = 150^\circ\text{C}$.

provided the value of k is known. Unfortunately, k cannot be determined directly and must be obtained experimentally.

In the case of small σ/β^* , the contact will be predominantly elastic. The friction can be assumed to arise from shearing of the junctions of the interfacial lubricated film. At very small value of σ/β^* , the value of A_t/A_n approaches unity. Putting $A_i=A_t=A_n$, $A_m=0$, $p_n=0.562$ Gpa and $\tau_i=0.066$ Gpa in Equation (9.18), therefore the friction coefficient is expected to approach 0.12 as σ/β^* decreases. However, at low values of σ/β^* , the variation in friction is 2 ~ 3 times higher than this value. This may be explained by the high conformity of the surfaces at low σ/β^* .

At low values of σ/β^* the variation in friction is several times that seen at higher σ/β^* suggesting that the shear stress at low σ/β^* is higher than that at higher σ/β^* . It may be explained by high surface conformity at low σ/β^* . In the sliding process, the sliding may be interrupted by external sources such as debris to cause ploughing and local hot spots which lead to welded junctions. These junctions may well grow with high conformity to give macroscopic stick-slip. Thus, the localised areas of the materials shear strength begins to add to the friction force. The upper limit of the frictional force is given by the material shear strength times the total contact area which is 0.55 in the present case. Thus the variation is expected to be between 0.12 and 0.55 at very low σ/β^* depending on the interfacial conditions.

The surface contact at large σ/β^* is predominantly plastic. Challen predicted this from a slip line field analysis of asperity deformation in which the frictional force is the force needed to push waves of plastically deformed material ahead of wedge shaped asperities on the harder surface. The friction coefficient is dependent only on the asperity shape and the ratio k of the interfacial film strength to the bulk metal shear strength and is given by equation (9.13). An appropriate value of k can be found by writing the asperity

slope in terms of σ/β^* in the same way as discussed in Chapter 6 for triangular surfaces. The values of k which gave the best comparison with experimental results were found to be 0.22, 0.4 and 0.5 at $T=50^\circ\text{C}$, 100°C and 150°C respectively as shown in Figure 9.2.

9.4.2 Ceramic Ball on Stainless Steel

The calculation of the adhesion friction is similar to the method described earlier. The contact area for various σ/β^* is shown in Figure 9.1. The friction and contact area at $\sigma/\beta^*=0.006$, where the value of A_p/A_n is close to zero, are used to calculate the interfacial shear stress. From Figure 9.3a and Figure 9.1 the friction and the contact area are given by $\mu=0.085$ and $A_t/A_n=0.8$. Substituting these values into equation (9.18), the interfacial shear strength is $\tau_i=0.06$ Gpa. From the variation of contact area and plastic area with σ/β^* , the friction for each σ/β^* can then be calculated for different values of f and the results are shown in Figure 9.3a, b and c.

If the values of f are 0, 0.25 and 0.4 for $T=50^\circ\text{C}$, 100°C and 150°C respectively, the prediction will closely follow the general trend of the experimental results except at very small and large σ/β^* .

The deviation of the friction based on the adhesion prediction from the experimental results can be explained by junction growth theory as described earlier. The adhesion friction due to the simple adhesion prediction is approximately valid for small interfacial shear strength. However, an increase in interfacial shear stress causes adhesion junction growth. So at large σ/β^* , the increase is due to plastic deformation, and at very low σ/β^* , the increase is due to elastic conformity.

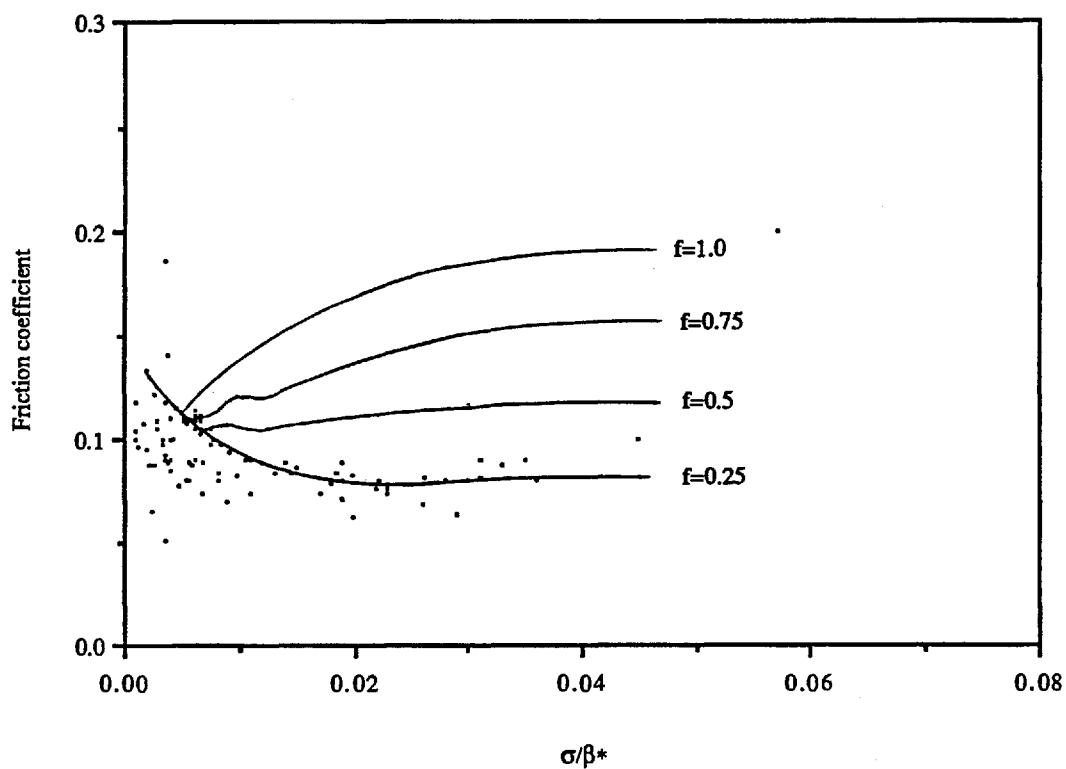


Figure 9.3(a) Variation of friction with σ/β^* for a ceramic ball on stainless steel at $T = 50^\circ\text{C}$. The lines — are predicted by adhesion friction.

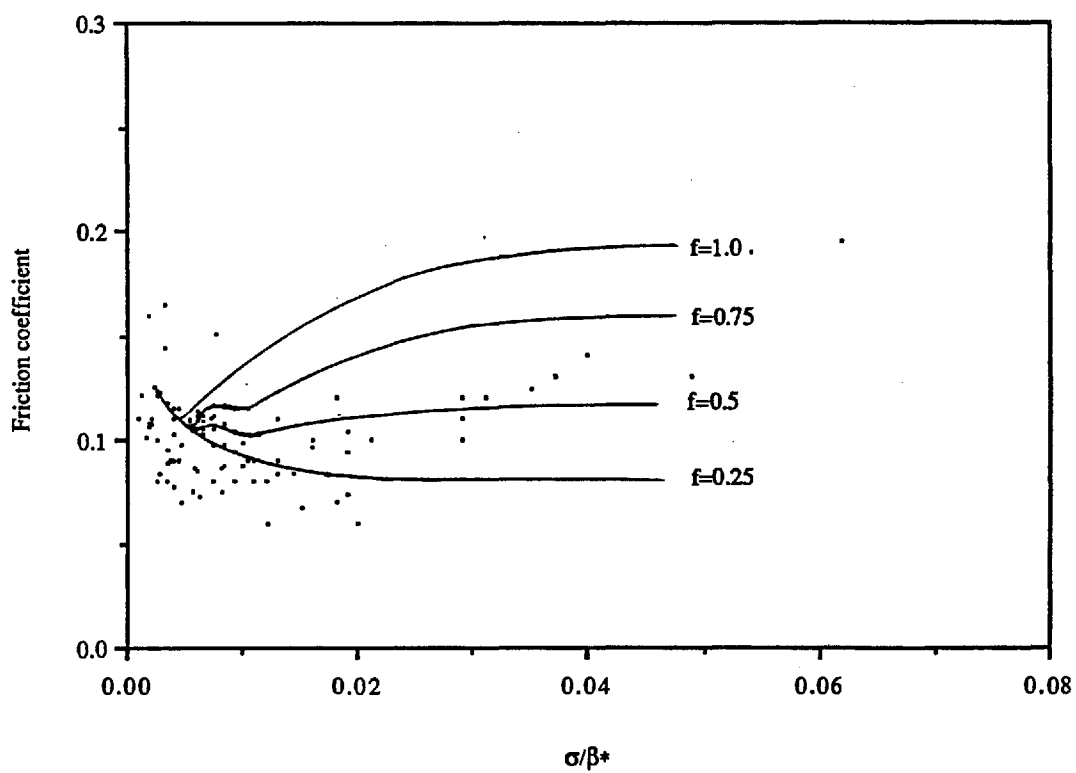


Figure 9.3(b) As Figure 9.3(a) but $T = 100^\circ\text{C}$.

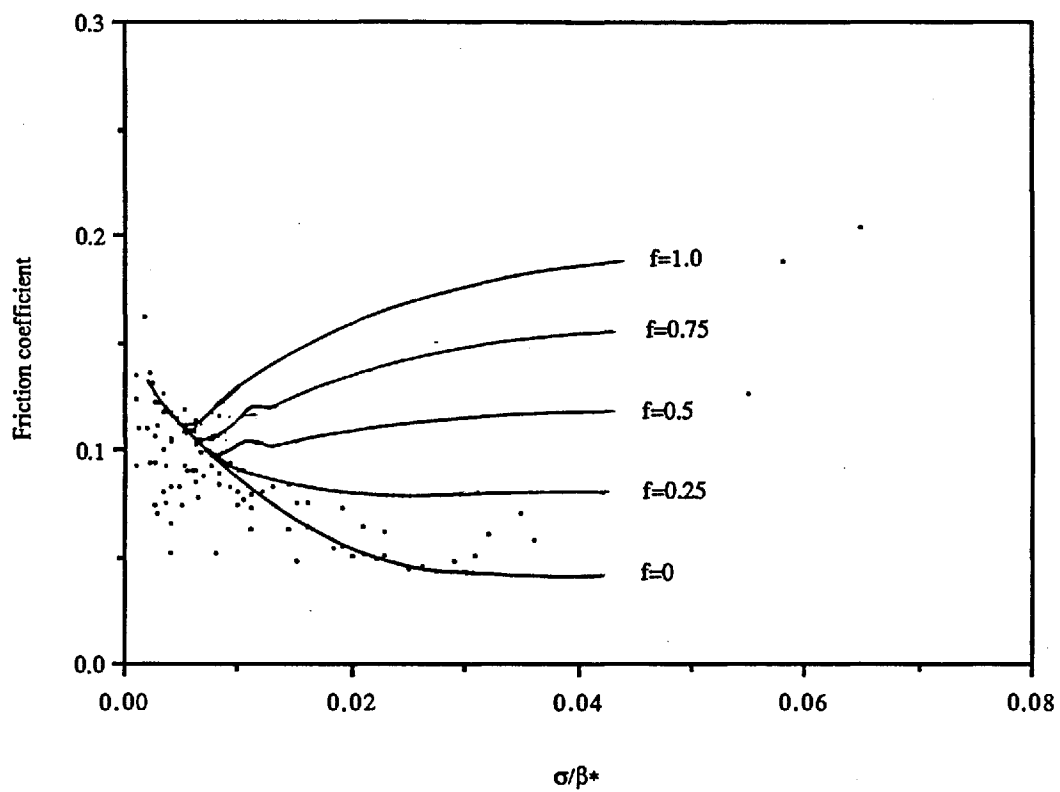


Figure 9.3(c) As Figure 9.3(a) but $T = 150^\circ\text{C}$.

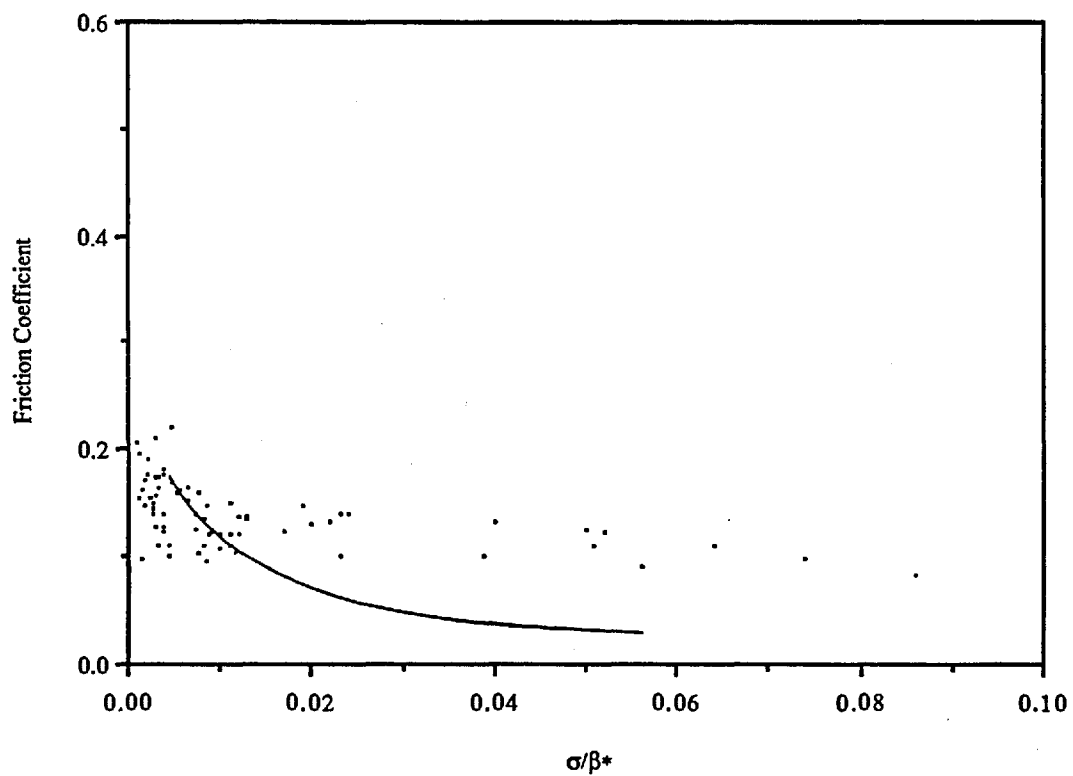


Figure 9.4(a) Variation of friction with σ/β^* for a steel ball on gear steel at $T = 50^\circ\text{C}$. The line; — is predicted by adhesion friction.

9.4.3 Steel Ball on Gear Steel

For the steel ball on gear steel, the contact is always elastic within the range of σ/β^* considered. Since the contact is elastic, the contact condition is not sufficiently severe to rupture the oxide film which separates the surfaces. The friction force can be assumed as arising largely from shearing of the interfacial film. The variations of friction and contact area with σ/β^* are shown in Figure 9.4a and Figure 9.1. As σ/β^* increases, both the friction and contact area decrease.

As before the interfacial shear strength can be calculated for a given value of μ and A_r/A_n at $\sigma/\beta^*=0.006$ which gives $\mu=0.15$ and $A_r/A_n=0.85$. Substituting these values into equation (9.18), the interfacial shear strength is $\tau_i=0.10$ Gpa. Using this value, the variation of μ with σ/β^* can be calculated using equation (9.18). The results are plotted in Figure 9.4 where the friction predicted from this simple model clearly does not fit the results obtained from friction tests.

One further, and very likely component of friction involved here is the ploughing of wear particles which have become interlocked in the interface. In Chapter 7, considerable evidence was presented to suggest that wear particles had been produced in the abrasion process in the contact of a ball on gear steel. The interlocking force can be significantly high due to the similar hardness of the two bodies. It is therefore to be expected that the experimental results might deviate from those predicted by the theory.

9.4.4 Steel Ball on Copper

Although the contact of a steel ball on copper is always predominantly plastic, the variation of friction shown in Figure 9.5 does not appear to be affected by plasticity. It was

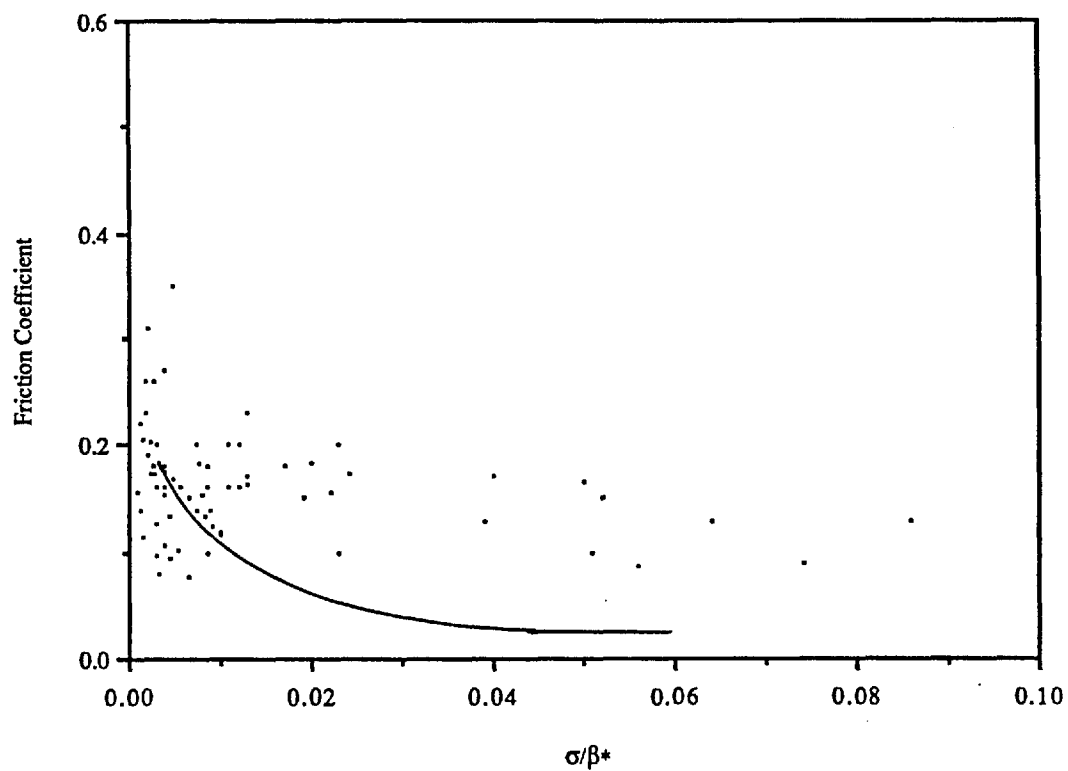


Figure 9.4(b) As Figure 9.4(a) but $T = 100^\circ\text{C}$.

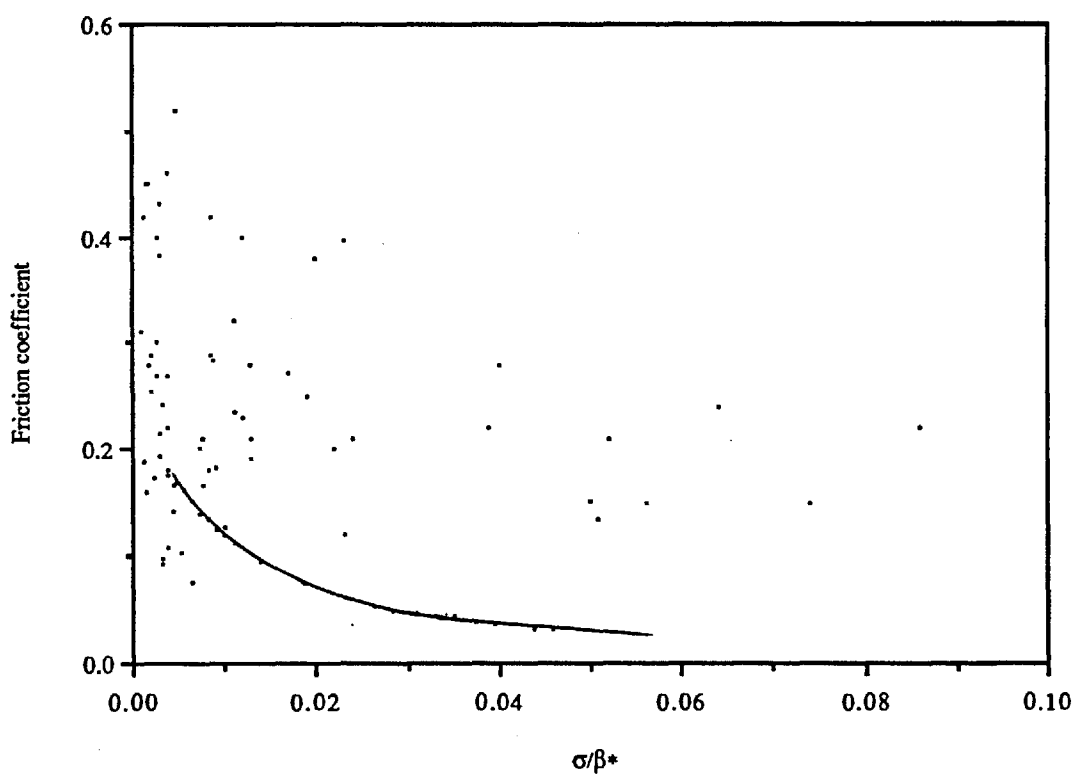


Figure 9.4(c) As Figure 9.4(a) but $T = 150^\circ\text{C}$.

concluded in Chapter 7 that the oxide film is sufficiently thick to prevent the formation of the metal-metal junction and so allow the oxide film react further with the lubricant to form an effective boundary lubricant. It is reasonable to assume that the adhesion friction is given by breaking the interfacial junctions for all σ/β^* .

Take $\sigma/\beta^*=0.03$ as an arbitrary value to obtain the values of μ and A_r/A_n which give $\mu \sim 0.075$ and $A_r/A_n=0.55$ from Figure 9.5 and 9.1 respectively. Using equation (9.18), the interfacial shear strength is $\tau_i=0.0677$ Gpa. The variation of μ with σ/β^* can be obtained as before. Results predicted by equation (9.18) are shown in Figure 9.5 together with experimental results. Although scatter in the results is evident, the experimental results follows the general trend of the theory.

The ploughing friction term due to Challen is also shown in Figure 9.5 where no value of k could be found to satisfactorily fit the experimental results.

9.5 Conclusions

Friction appears to be primarily as combination of adhesion, ploughing and plastic deformation. Adhesion will usually be responsible for the main component of friction. Under mild conditions, the friction force may be predicted by simply summing the shearing force at each junction provided that the area and shear strength of each kind of asperity junction is known.

According to Tabor, if the shear strength of the interface is large there will be junction growth and increase in contact area as a result of the combined effect of normal stress and shear stress. The real contact area obtained from a static contact in normal approach is no longer valid and the predicted friction force will deviate from the

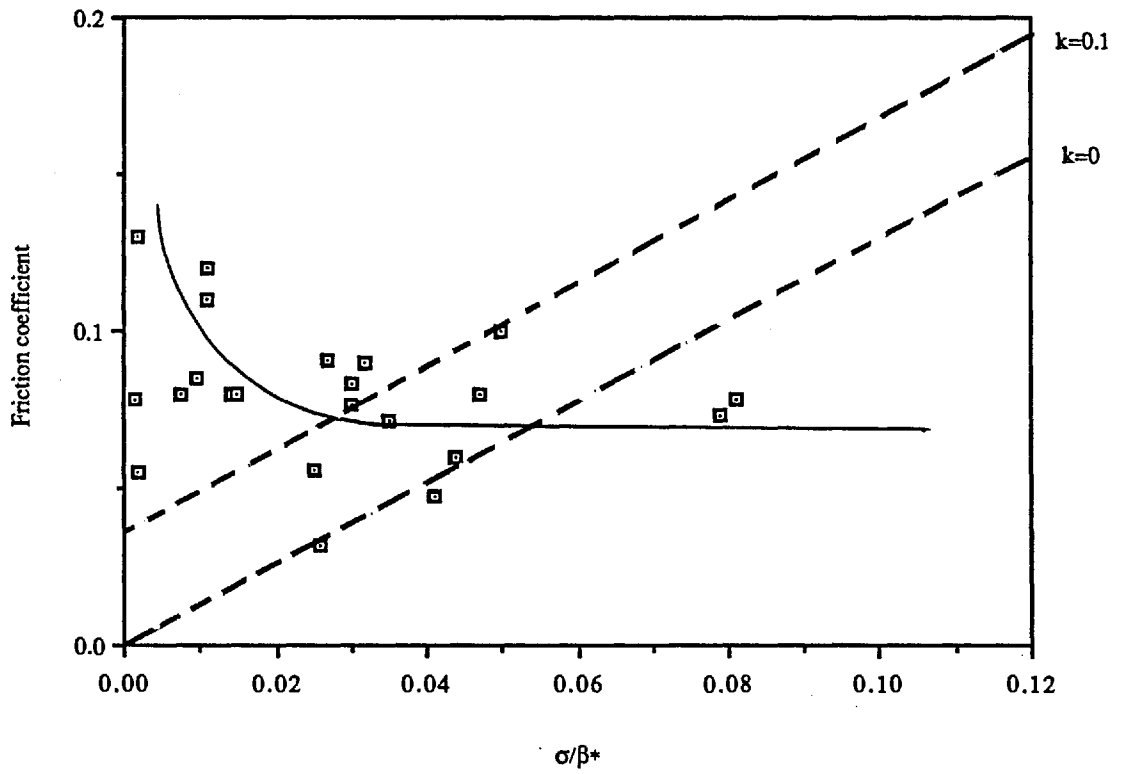


Figure 9.5 Variation of friction with σ/β^* for a steel ball on copper at $T = 50^\circ\text{C}$. The line — is predicted by adhesion friction. The lines---- are due to Challen.

experimental results. For a rough surface, the increase in shear strength is due to plastic deformation causing the rupture of oxide films leading to metal-metal contact. For a very smooth surface, the increase is due to the total conformity. It is therefore to be expected that under these conditions, adhesion will play an important role in defining the friction force.

By ignoring the components of ploughing and plastic deformation, the adhesion component gives good agreement with the experimental results in the case of a steel ball or a ceramic ball on stainless steel or copper provided that the sliding condition is not severe. In the case of a steel ball on gear steel where the ploughing of wear particles is also important, the friction force cannot be assumed as coming from an adhesion component only.

It is interesting to note that the experimental results of a steel ball on stainless steel at high σ/β^* are predicted by both the junction growth theory and the slip line field theory due to Challen. However, the nature of the friction force from which the expressions are derived is different. The first assumes the frictional force is arising from breaking the asperity adhesion and the friction coefficient is only dependent on the interfacial shear strength. The second combines the effects of adhesion and ploughing of the asperities in which case the friction coefficient is expressed in terms of asperity slope and interfacial shear strength.

When both theories were used to predict friction of a steel ball on copper, friction from the adhesion component followed the general trend of the experimental results much better than the slip-line theory.

CHAPTER 10

APPLICATIONS AND SUGGESTIONS FOR FURTHER WORK

In this chapter, implication of the results related to tribological problems are discussed. Suggestions for further work are given.

10.1 Applications

10.1.1 Cylinder Bore Scuffing

The implication of the results from friction tests and numerical contact analysis can be related to the sliding history of a cylinder bore. The surface finish of cylinder bores is one of the significant factors affecting the life of a reciprocating engine. For a honed cylinder bore, the plateau is 1/2-2/3 of the surface as a result of the honing finish with cross hatch angle 22 to 32 degrees off horizontal uniformly cut in both directions. For an unrun engine, a period of running-in, often referred to as 'break-in', is required to remove the asperity crests to promote the surface plateaus. During this initial stage, the cylinder bore wear rate is high and the character of the surface is continually changing. The high wear rate continues until the surface condition will carry the imposed loads.

Before the break-in period, the fine bore roughness is $0.6\text{ }\mu\text{m}$ - $1.0\text{ }\mu\text{m}$ cla ($\sim 0.75\text{ }\mu\text{m}$ - $1.25\text{ }\mu\text{m}$ rms) together with the shorter wavelength features on the plateau. The surface can be considered as lying in or close to the region A of Figure 1.1b. In the break-in period, the removal of the asperity crests effectively reduces the roughness of the surface

and increases the contact area resulting in the relief of the contact stress. Then the surface conditions resemble a surface in region B. The gaps between each plateau provided by honing serve to trap lubricant and act to break any adhesion junction formed in the interfaces. The friction and surface damage is small. In this running period, the removal of the material from the bore surface continues, although on a much smaller scale. The bore surface roughness continues to decrease and approaches the region C where contact approaches complete elastic conformity. In this circumstance, bore scuffing is more likely to occur and it may represent the life of an engine.

10.1.2 Sealing

It has been realised that materials with low modulus have better sealing ability than materials with higher modulus. For any materials, the smoother the surface, the greater the ability for sealing. This can be explained by the elastic conformity principle, with the conformity of the surfaces dependent on the load, material' moduli and topographic parameters. The load required for complete sealing of a random rough surface is governed by equation 6.15. Therefore, for a given material and external load, the ability for sealing can be determined by the ratio σ/β^* .

10.1.3 Directional Friction Behaviour

Many engineering surfaces, such as those prepared by grinding, shaping, turning, etc, possess a pronounced lay such that the wavelength structure in the lay direction is considerably larger than the wavelength structure across the lay. Figure 10.1 shows the friction curves of a steel ball on stainless steel with an intermediate value of σ/β^* along and across the lay direction. The friction coefficient in the lay direction is twice that obtained

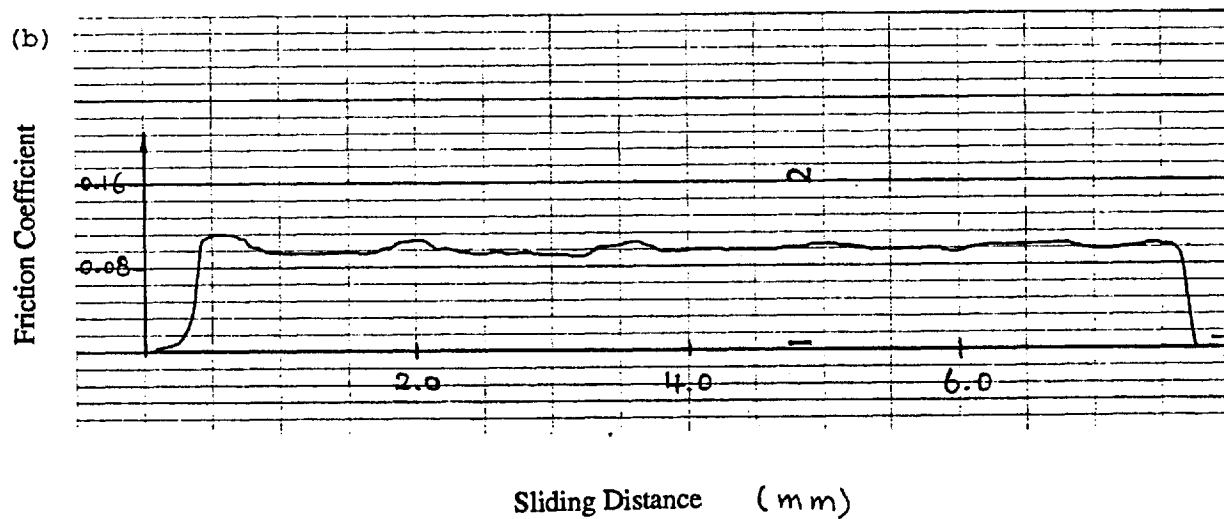
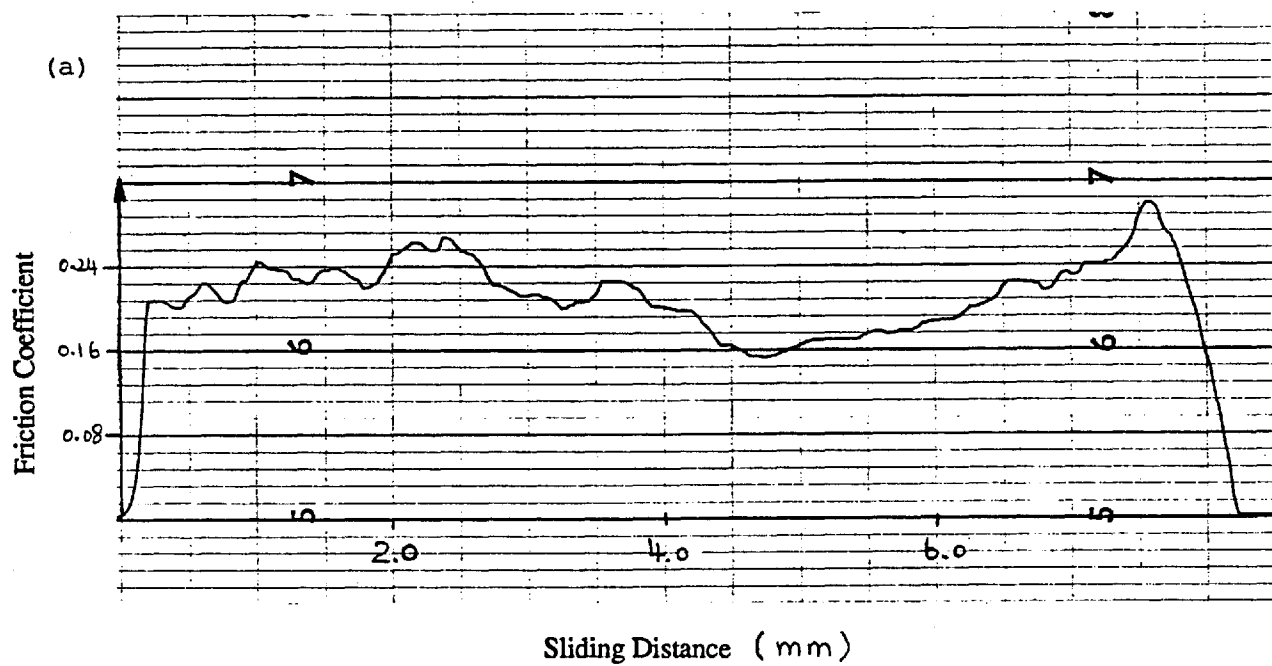


Figure 10.1 Friction curves of a steel ball on stainless steel. (a) along the lay direction (b) across the lay direction.

across the lay direction.

This revealing result adds considerable credence to the adhesive friction mechanisms described earlier where junction growth is prevented by the valley structure of the roughness acting to break adhesive junctions. In the friction of Figure 10.1 the real contact area is the same whichever direction of sliding is involved. However sliding across the lay offers a much greater potential for breaking local adhesive junctions than sliding along the lay. Thus the friction across the lay would be expected to be less than friction along the lay if this adhesive mechanism was important, which is the case of this particular test.

The magnitude and character of this frictional difference is significant and requires further research, particularly as this type of test seems to offer a means of isolating the effects of adhesive mechanisms.

10.2 Work Suggestions

10.2.1 Running-in

Surface contact occurs on the topmost strata of the surface. After prolonged sliding the surface asperities at the top will be removed while the asperities below the mean line will usually remain unchanged. Topographic studies of surfaces during running-in are usually performed in terms of the change in topography as a function of time. The performance of a surface can often be related directly to the surface topography. In the process of running-in, the surface topography will change which in turn affects the surface performance. It is thus instructive to monitor the change in surface performance and also the change in surface topography. Any further study can make full use of the numerical contact analysis techniques to obtain information of the contact conditions, such as local

contact stresses and areas. The contact analysis results can be used to correlate with the surface performance such as friction and wear based on the three regimes of contact described here.

10.2.2 Friction Tests at a Different Load

In the friction tests reported here, it has been confirmed that the surface failure of a very smooth surface is related to the contact conformity. The tests were carried out at a constant load. In Chapter 6, it is shown that the degree of conformity of rough surfaces is dependent on load. Therefore, the lower boundary (such as that shown in Chapter 8), that divides safe and unsafe surfaces on a σ - β^* map, would move up as the load increases. It is instructive to carry out friction tests to verify the theory predicted by this conformity principle.

10.2.3 Fluid Sealing

The numerical contact analysis takes advantage of the interaction of deformations from all the contact points. The deflection can be calculated for the points not in contact from the contact pressure. Therefore, the gaps between surfaces can be obtained. These are expected to be dependent on the load, σ and β^* . Fluid-leakage results can thus be related to load and the suitable surface parameters. Thus, it should be possible to predict the fluid-leakage through gaps between surfaces for a given load by measuring the texture of the surfaces.

Appendix 1

The total surface displacement u_i due to the pressure acting at j is given by equation 4.24.

$$u_i = \frac{1-\nu^2}{\pi E} \sum_j \sum_k p_j \alpha_k F_{xy} \quad (A1)$$

Differentiate equation A1 with respect to w_j ,

$$\frac{du_i}{dw_j} = \frac{1-\nu^2}{\pi E} \sum_j p_j \sum_k \alpha_k F'_{xy} + \frac{1-\nu^2}{\pi E} \sum_j p'_j \sum_k \alpha_k F_{xy} \quad (A2)$$

At the boundary of contact, equation 4.11 can be used as a good approximation, i.e.

$$F_{xy} = \frac{2a \cdot 2b}{\sqrt{x^2 + y^2}} \quad (A3)$$

b and y are function of w_j and are given by,

$$b = \mu w_j / 2 \quad (A4)$$

and

$$y = \mu k w_j \quad (A5)$$

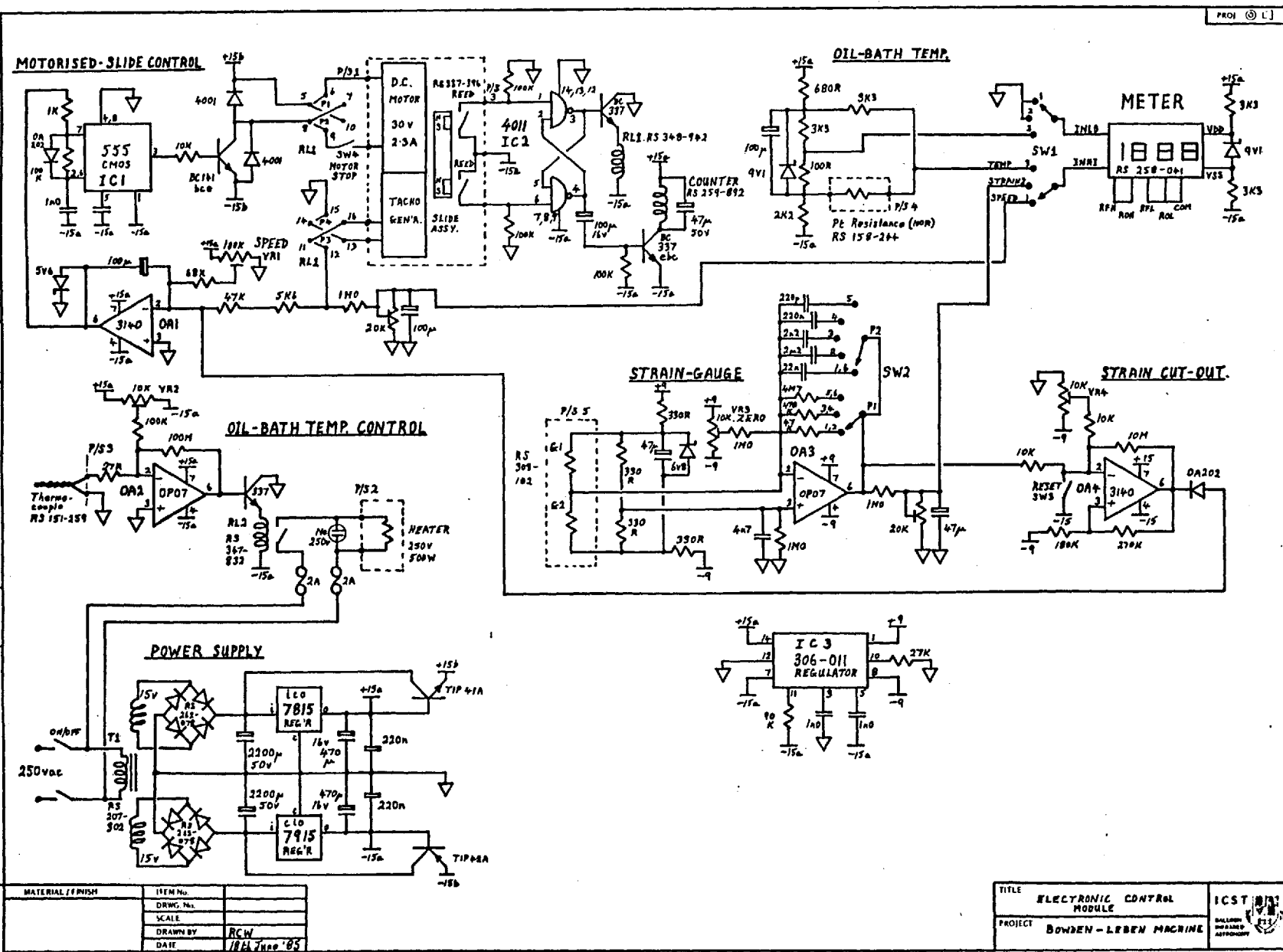
where $\mu = 2/(2N_y + 1)$

Differentiate equation A3 with respect to w_j , and using equations (A4) and (A5),

$$\begin{aligned} F'_{xy} &= \frac{2a\mu(x^2 + y^2 - \mu k w_j y)}{(x^2 + y^2)^{\frac{3}{2}}} \\ &= \frac{2a\mu x^2}{(x^2 + y^2)^{\frac{3}{2}}} \end{aligned} \quad (A6)$$

Substitute equations A3 and A6 into equation A2,

$$\begin{aligned} \frac{du_i}{dw_j} = & \frac{2 a (1 - v^2)}{\pi E} \sum_j p_j \sum_k \alpha_k \frac{\mu x^2}{(x^2 + y^2)^{\frac{3}{2}}} \\ & + \frac{2 a (1 - v^2)}{\pi E} \sum_j \frac{dp_j}{dw_j} \sum_k \alpha_k \frac{\mu w_j}{(x^2 + y^2)^{\frac{1}{2}}} \end{aligned} \quad (A7)$$



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