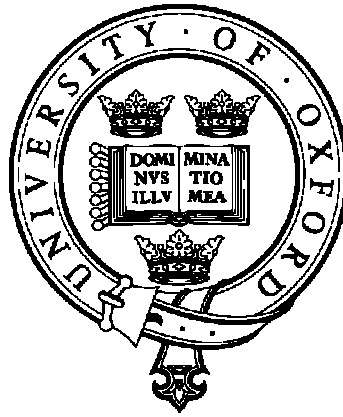




STUDIES OF FRICTIONAL INTERFACE BEHAVIOUR: EXPERIMENTS AND MODELLING

Thesis Submitted to the University of Oxford for the Degree

of

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Studies of Frictional Interface Behaviour: Experiments and Modelling

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ABSTRACT

Predictive models of structures containing frictional joints presently suffer from poor descriptions of interface behaviour at the joints. This thesis aims to address this shortfall by furthering the physical understanding of parameters affecting interface behaviour such as friction and contact stiffness. Aspects of friction and contact stiffness relevant to the characterisation of fretting joints are investigated by a combined modelling and experimental approach.

Friction and wear behaviour in gross-slip fretting are investigated by in-line and rotational fretting tests. New 3D topography parameters are found to be useful in the analysis of surfaces during fretting. Wear-scar shape is found to be dependent on material. A phenomenon whereby friction increases during the gross-slip phase of individual cycles is found to be due to wear-scar interaction primarily through the interference of local features distributed over the contact area. These features are similar in size to the applied fretting stroke. A simple model to explain the behaviour is put forward which shows that wear-scar shape determines the form of the friction variation.

A finite-element (FE) model of the interaction of an elastic-plastic asperity junction is used to predict sliding friction coefficients. The modelling differs from previous work by: permitting greater asperity overlaps, enforcing an interface shear strength, and allowing material failure. The results are also used to predict friction coefficients for a stochastic rough surface. The magnitudes of the predicted friction coefficients are generally representative of experimental measurements. Results suggest that friction arises from both plasticity and tangential interface adhesion.

Contact stiffness is studied for both fretting and non-fretting. A technique to isolate the true interface stiffness from results derived from load-deflection data is developed by comparing experimental and FE results. In the fretting wear case, comparison of tangential contact stiffness results in the literature with FE results reveals an interface whose compliance dominates the response to the extent that stiffness is proportional to contact area. In fretting tests such as this, wear debris is thought to be a factor contributing to high interface compliance. Non-fretting experiments performed here show that, at higher pressures, interface domination is reduced as the contact approaches the smooth case. Experiments are performed where contact stiffness is measured simultaneously by both ultrasound and digital image correlation. The effect of normal and tangential loading upon the contact stiffness (normal and tangential) is investigated. Experimental evidence showing that ultrasound measures an ‘unloading’ stiffness while DIC measures a ‘loading’ stiffness is obtained for the case of tangential loading where the ‘DIC stiffness’ decreases with increasing tangential load whereas the ‘ultrasound stiffness’ remains approximately constant. On average, ultrasound gives magnitudes 3.5 and 2.5 times stiffer than the DIC results for the normal and tangential stiffness cases, respectively. The difference in magnitudes can largely be physically explained, and is relatively small considering the significant differences between the techniques. Therefore, both methods can claim to give valid measurements of contact stiffness – though each has its own limitations which are outlined herein.

“to strive, to seek, to find, and not to yield”

Alfred Lord Tennyson

In dedication to my parents Daniel and Monica

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PREFACE

This dissertation is the result of work carried out by the author in the Solid Mechanics Group of the Department of Engineering Science at the University of Oxford in the period between October 2008 and June 2012. The work was supervised by Prof. David Nowell and funded by a studentship bestowed by the Engineering and Physical Sciences Research Council (EPSRC) under grant reference EP/E058337/1 – “A Predictive Approach to Modelling Frictional Joint Performance” (PAMFJP). No part of the research herein has previously been submitted for a degree at this or any other institution. The work is original, and due attention has been drawn to the work of others. The work in Chapters 4 and 5 resulted in the award of first price in the 2011 ‘Young Stress Analyst Competition’ by the British Society for Strain Measurement (BSSM).

Some of the work described herein has contributed to the following publications:

1. Mulvihill D.M., Brunskill H, Kartal M.E., Dwyer-Joyce R.S., and Nowell D. ‘A comparison of contact stiffness measurements obtained by the digital image correlation and ultrasound techniques’, to be published in an international journal.
2. Mulvihill D.M., Kartal M.E., Nowell D., and Hills D.A. (2011) ‘An elastic-plastic asperity interaction model for sliding friction’, *Tribology International*, **44**(12), pp. 1679–1694
3. Mulvihill D.M., Kartal M.E., Nowell D., and Hills D.A. (2011) ‘Investigation of non-Coulomb friction behaviour in reciprocating sliding’, *Wear*, **271**(5,6), pp. 802–816.
4. Kartal M.E., Mulvihill D.M., Nowell D., and Hills D.A. (2011) ‘Measurements of pressure and area dependent tangential contact stiffness between rough surfaces using digital image correlation’, *Tribology International*, **44**(10), pp. 1188–1198
5. Kartal M.E., Mulvihill D.M., Nowell D, and Hills D.A. (2011) ‘Determination of the frictional properties of titanium and nickel alloys using the digital image correlation technique’, *Experimental Mechanics*, **51**(3), pp. 359–371.
6. Mulvihill D.M., Kartal M.E., Nowell D., and Hills D.A. (2011) ‘Investigation of the friction variation with sliding which is commonly observed in individual fretting test cycles’, *Applied Mechanics and Materials*, **70**, pp. 213–218. [Presented at the 8th International Conference on Advances in Experimental Mechanics, Edinburgh, UK, Sept. 7th–9th, 2011]
7. Kartal M.E., Mulvihill D.M., Nowell D., and Hills D.A. (2011) ‘Measurement of tangential contact stiffness in frictional contacts: the effect of normal pressure’, *Applied Mechanics and Materials*, **70**, pp. 321–326. [Presented at the 8th International Conference on Advances in Experimental Mechanics, Edinburgh, UK, Sept. 7th–9th, 2011]

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10. Dini D., Medina S., Proprentner D., Schwingshackl C.W., Olver, A.V., Petrov E., Ewins D.J., Mulvihill D., Kartal M., Hills D.A., and Nowell D. (2009) ‘The Influence of the Microscopic Contact Behaviour on Energy Dissipation and Hysteretic Response of Frictional Joints for Aeronautical Applications’, in *Proceedings of the Fourth World Tribology Congress*, Paper No. I–314, Kyoto, Japan, Sept 6th–11th.

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Go raibh mile maith agaibh go leir,

D.M. Mulvihill

Oxford, June 2012

NOMENCLATURE

Roman symbols

A, B	complex amplitude coefficients of displacement waves
A_r	area of real contact
a	contact half-width
b	contact depth
$C_S(x_0, y_0)$	coordinates of the centre of a grid before deformation
$C_S'(x_0', y_0')$	coordinates of the centre of a grid after deformation
c	wave speed
c_1, c_2	wave speeds in bodies 1 and 2
D_1, D_2, D_3	parameters of the Johnson-Cook failure criterion
d	normal distance between surface reference planes
E	Young's modulus
E_1, E_2	Young's modulus in bodies 1 and 2
$e_{D,i}^P$	equivalent plastic strain at initiation of damage
F_n	local normal force at a peak-trough interface
f	coefficient of friction
f_u	ultrasound frequency
h	normalised distance between surface reference planes
$I_u(x_j, y_k)$	grey level intensity function (initial image)
$I_d(x_j', y_k')$	grey level intensity function (deformed image)
K_{DIC}	contact stiffness measured by digital image correlation
K_{FE}	contact stiffness derived from a finite element model
$K_{Interface}$	contact stiffness of an interface isolated from other contributions
$K_{Monolith}$	contact stiffness relating to a single piece specimen (i.e. no interfaces)
K_n	normal (longitudinal) contact stiffness
$K_{Pad/Specimen}$	contact stiffness relating to an experiment with a pad/specimen interface
K_t	tangential (shear) contact stiffness
l	vertical separation distance between asperity bases
M, N	number of grid divisions in two defined directions

m	mass
N_a	number of asperities on a surface
N_c	number of cycles
n	power/index of a polynomial representing a surface groove
P	total normal force
P'	total normal force at a single asperity
$\bar{P}$	mean total normal force for an asperity junction interaction
P_c	critical normal load required to cause first yield
$P_s(x_j, y_k)$	coordinates of an arbitrary grid point
$P'_s(x'_j, y'_k)$	coordinates of an arbitrary grid point after deformation
p	mean normal pressure at a single asperity junction
p_h	penetration hardness
p_m	mean normal pressure
Q	total tangential force
Q'	total tangential force at a single asperity
$\bar{Q}$	mean total tangential force for an asperity junction interaction
q_m	mean tangential traction
R	reflection coefficient
R_a	average surface roughness of a line profile
R_q	root mean square surface roughness of a line profile
r	radial coordinate (radius)
r^*	normalised asperity radius
r_i	inner radius
r_o	outer radius
S_a	average areal surface roughness
S_k, S_{pk}, S_{vk}	bearing area curve parameters
S_{ku}	kurtosis of a surface
S_p	maximum peak height of a surface
$S_{p,i}$	i^{th} largest peak height of a surface
S_q	areal root mean square roughness (standard deviation of surface heights)
$S_{q,pad}$	pad areal root mean square roughness (standard deviation of surface heights)

$S_{q,spec}$	specimen areal root mean square roughness (standard dev. of surface heights)
S_{sk}	skewness of a surface
S_v	maximum valley depth of a surface
$S_{v,i}$	i^{th} lowest valley depth of a surface
S_z	maximum thickness of a surface (equivalent to flatness)
S_{10z}	ten point thickness of a surface
Sal	auto-correlation length (linear)
Str	texture aspect ratio
Stdi	texture direction index
s	normalised asperity peak height
T	total torque (arising from both contact pairs)
t	time
U_E	work done by external forces
U_P	energy dissipated due to plasticity
U_S	energy dissipated in overcoming interface shear strength
u	displacement
u_1, u_2	ultrasonic displacement waves in bodies 1 and 2
u_1	incident ultrasound displacement wave
u_R	reflected ultrasound displacement wave
u_T	transmitted ultrasound displacement wave
u_x, u_y, u_z	displacements in the Cartesian coordinate directions
ur_x, ur_y, ur_z	rotations about the Cartesian coordinate axes
x, y, z	Cartesian coordinates, (z is normal to a surface)
Z	acoustic impedance
Z_1, Z_2	acoustic impedance in bodies 1 and 2
$z(x_j, y_k)$	surface height at a Cartesian grid point
$z(r_j, \theta_k)$	surface height at a radial grid point
$z_{pad}(r_j, \theta_k)$	pad surface height at a radial grid point
$z_{spec}(r_j, \theta_k)$	specimen surface height at a radial grid point
z_p	asperity peak height

Greek symbols

α, β, γ	von Mises yield criterion constants
$\Delta\theta$	increment of relative angular position
δ	relative linear tangential position with respect to the aligned configuration
δ_i	initial relative tangential position with respect to aligned configuration
δ_f	final relative tangential position with respect to the aligned configuration
δ_{actual}	actual tangential displacement amplitude (zero-to-peak)
δ_{app}	applied tangential displacement amplitude (zero-to-peak)
δ_{slip}	slip displacement amplitude (zero-to-peak)
δ_θ	relative angular displacement
$\delta_{\theta,\text{app}}$	applied angular displacement amplitude (zero-to-peak)
ε	overlap (interference) between asperities
ζ	segmented ring area divided by full ring area
η	stress triaxiality
θ	angular coordinate
κ	contact stiffness per unit nominal area
κ_n	normal (longitudinal) contact stiffness per unit nominal area
κ_t	tangential (shear) contact stiffness per unit nominal area
ρ	density
σ_p	standard deviation of asperity peak heights
σ_1, σ_2	stress normal to the interface in bodies 1 and 2
σ_b	normal stress at a boundary
σ_Y	normal yield stress
τ	interface adhesional shear strength at a single asperity junction
τ_Y	yield stress in shear
ν	Poisson's ratio
ϕ	probability distribution of asperity peak heights
ω	angular frequency

Abbreviations

ACF	auto-correlation function
AFM	atomic force microscopy
CCF	cross-correlation function
DIC	digital image correlation
EDX	energy-dispersive X-ray spectroscopy
FE	finite element
FFT	fast Fourier transform
JC	Johnson-Cook
LNS	laser nano sensor
LSF	least-squares function
LVDT	linear variable differential transformer
NF	no failure
PEEQ	ABAQUS abbreviation for equivalent plastic strain
RMS	root mean square
SEM	scanning electron microscope
UPR	ultrasound pulser-receiver

Chapter 1

Introduction

1.1 Motivation and background

Frictional joints occur in a vast range of engineering circumstances where components are joined together and where loads are transferred between parts. Important examples include: bolted and riveted joints, keyway shaft couplings, dovetail connections, and spline couplings.

When joints are subjected to dynamic loading, as in most structures and machines, they influence the performance and life of the overall structure by affecting two very important aspects of design: structural integrity and vibration response. From a structural integrity point of view, dynamically loaded joints are usually adversely affected by fretting. Fretting occurs when two contacting solid bodies experience small oscillatory movement at the joint interface: when these movements are small enough to allow some parts of the contact to remain stuck while others slip (called partial slip), fretting fatigue (which involves the formation and propagation of detrimental cracks) occurs; whereas, larger amplitude movements involving periodic full sliding of the contact (called gross slip) are associated with fretting wear (which involves the degradation of a joint by material removal). From a vibration response viewpoint, joints affect both the stiffness

and damping of the structure. They reduce stiffness as a result of the compliance introduced by the presence of a ‘real rough interface’, and they provide damping by dissipation of frictional energy. Thus, if these aspects of joint design are not fully accounted for in machine design, catastrophic failure or poor performance of the overall structure is the likely result. Hence, the parameters which characterise frictional joints need to be fully understood (and ideally should be capable of being predicted) if the structural integrity and vibration influencing factors are to be properly accounted for in engineering design.

At present, the vibration and stress analysis aspects of the individual components that comprise structures can be predicted extremely accurately by methods such as finite element analysis. However, current predictive methods of modelling joints are underdeveloped – with the result that modelling predictions of jointed structures are often inaccurate due to a failure to correctly model the physics of joint behaviour. Hence, the inclusion of joint design in overall vibration and structural models of assemblies inevitably involves the experimental measurement of parameters which characterise the joint properties. Thus, current methods of modelling overall assemblies are not truly predictive. A group which has focused particular attention on this problem is that of the aero-engine industry. In the aero-engine, there are a number of critical frictional joints where structural and vibration response properties are heavily affected such as: blade-tip shrouds, blade-disc dovetail connections (Figure 1.1), blade under-platform dampers, and spline couplings. Figure 1.1b shows a slice of a dovetail blade-root representative of how fan blades (Figure 1.1a) are joined to their holding disc. Figure 1.1c shows a schematic of the dovetail blade-root fretting problem; while, Figure 1.1d shows how the situation at a dovetail joint can be modelled by a flat-and-rounded punch fretting against a flat (the rounded edges give an equivalent form of contact pressure distribution)

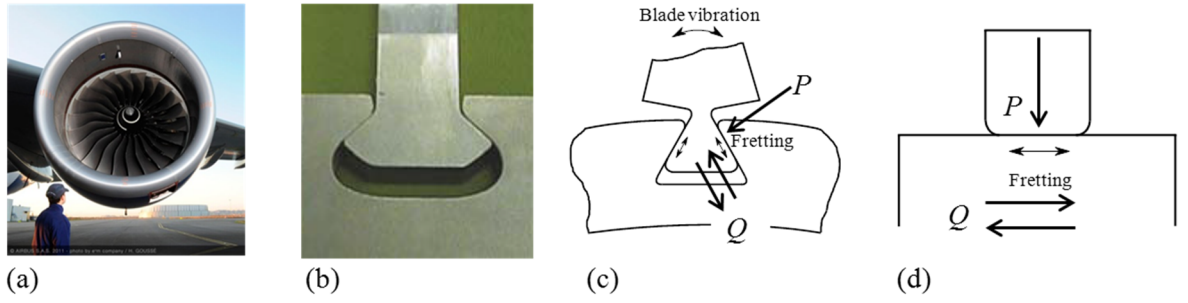


Figure 1.1: Illustration of fretting in the fan-blade dovetail joint of gas turbine engines: (a) Rolls-Royce Trent XWB aero-engine showing the inlet fan (Rolls-Royce [1]), (b) representative slice of a fan blade/disc dovetail joint (Mugadu [2]), (c) schematic diagram describing fretting at a blade root, and (d) an approximate representation by a flat-and-rounded punch fretting against a flat.

When experiments are carried out to characterise joints, tangential force hysteresis loops are usually measured in the laboratory. The tangential force hysteresis loop is a plot (for a single fretting cycle) of tangential force against relative tangential displacement. Figure 1.2 shows the general types of hysteresis loop obtained from fretting joints in each of the fretting regimes: the partial-slip regime (Figure 1.2a) and the gross-sliding regime (Figure 1.2b). Notice, that there is also a partial-slip phase in a gross-sliding loop – this is sometimes referred to as the pre-sliding or micro-slip phase.

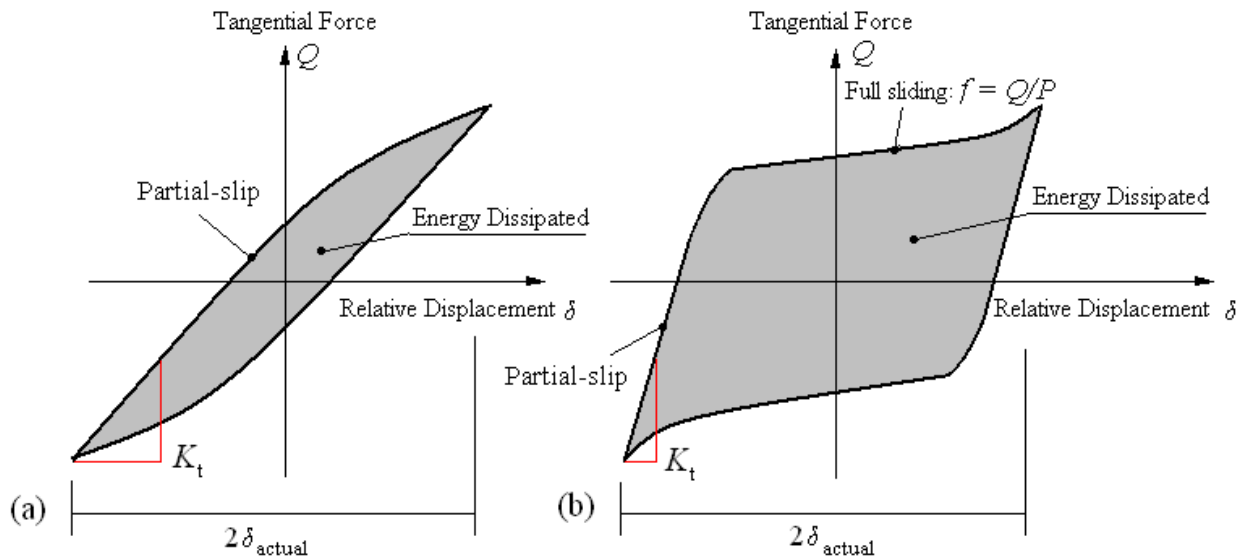


Figure 1.2: Typical fretting hysteresis loops for (a) the partial-slip regime (associated with fretting fatigue), and (b) the gross-sliding regime (associated with fretting wear).

This thesis is mainly concerned with the full-sliding case (Figure 1.2b). Three important parameters can be identified from the loop in Figure 1.2b: the friction coefficient f , the tangential contact stiffness K_t , and the actual displacement amplitude $2\delta_{\text{actual}}$ (peak-to-peak). The friction coefficient corresponds to the tangential force during sliding divided by the normal force on the contact (in accordance with the Amontons-Coulomb ‘law’ [3, 4]). The tangential contact stiffness is best given by the initial slope of the hysteresis loop just after the reversal of motion when the contact is reacting to changes in tangential load while remaining mostly fully stuck. This is, of course, only equivalent to the *true* contact stiffness if no bulk or machine compliance is included in the relative displacement measurements. The actual peak-to-peak displacement amplitude is easily obtained from the loops by determining the difference between the maximum and minimum relative displacement. Due to various compliances in the test machine and specimen holding devices, the actual displacement amplitude is usually somewhat less than the applied displacement amplitude.

The question arises as to why these parameters are important. The contact stiffness obviously directly influences resonant frequencies of jointed assemblies as it introduces an additional compliance ‘spring’ in addition to bulk material stiffness. The contact stiffness of a joint has the effect of reducing the overall stiffness of an assembled structure: since it adds an extra flexible element. The area enclosed in the loop corresponds to the energy expended by the joint in frictional work for a given cycle. It is the loss of this energy from the structure which gives rise to frictional vibration damping, and it is the investment of this energy in frictional work, which promotes fatigue crack initiation and fretting wear as a result of higher shear stresses. Thus, there is a trade-off between designing for vibration damping and designing for fretting fatigue and wear reduction (vibration engineers often want higher damping, while structural integrity

engineers demand lower friction). From a geometrical analysis of the hysteresis loop, it can easily be seen that this energy (the hysteresis loop area) is largely determined by the three parameters mentioned above: the friction, the tangential contact stiffness and the true peak-to-peak oscillation amplitude. It should be clear from the foregoing analysis that these three parameters almost fully define the hysteresis loops and therefore characterise the joint's properties in relation to fretting and vibration issues. We should also note here that the vibration problem and the fretting problem are coupled, since, although the fretting loops influence the vibration amplitude, it is this amplitude which determines the peak-to-peak fretting displacement (or stroke) imposed on the joint.

At this point, it is worthwhile taking an introductory look at the basic definitions of friction coefficient and contact stiffness. According to the Amontons-Coulomb 'laws' of friction [3, 4], a certain critical tangential force is needed to initiate sliding of a contact, and this critical tangential force is proportional to the normal force acting on the contact. If the normal force is doubled, then the tangential force will need to be doubled to initiate sliding. The coefficient of proportionality in this relationship is known as the coefficient of friction (i.e. $f = Q/P$, where Q , here, is the critical tangential force needed). It is also part of the Amontons-Coulomb description of friction that this critical tangential (or friction) force is independent of both nominal contact area and of sliding displacement. Proportionality with normal load and independence from nominal area are usually well observed in practice for most engineering surfaces. However, the friction coefficient during sliding often behaves differently (i.e. it can be dependent on velocity) and is viewed as a separate friction coefficient – the kinetic coefficient of friction (also defined as Q/P where Q , in this case, is the tangential force during sliding). The friction coefficient associated with the initiation of sliding is known as the static coefficient of friction.

Contact stiffness can be defined either in the normal or tangential direction. A schematic diagram illustrating the definition of rough surface contact stiffness is given in Figure 1.3.

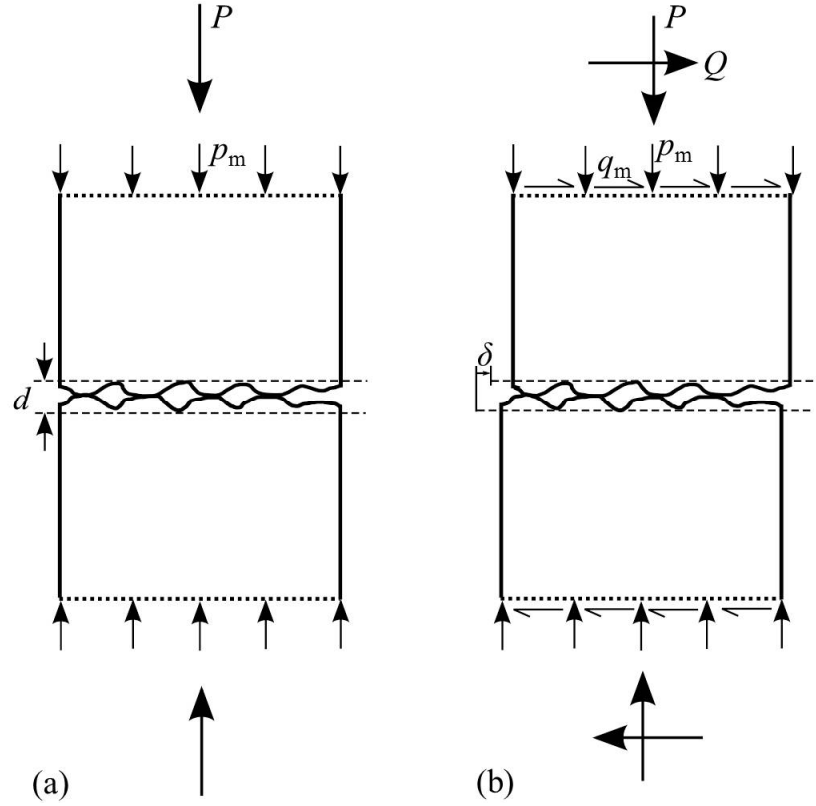


Figure 1.3: Illustration of normal contact stiffness (a) and tangential contact stiffness (b).

Normal contact stiffness K_n may be defined as the rate of change of normal load P with respect to the change in normal separation d of the bodies due solely to the rough interface:

$$K_n = \left| \frac{dP}{dd} \right|. \quad (1.1)$$

Tangential contact stiffness K_t can be defined as the rate of change of tangential load Q with respect to change in tangential separation δ of the bodies due purely to the rough interface:

$$K_t = \left| \frac{dQ}{d\delta} \right|. \quad (1.2)$$

It is sometimes more convenient to express contact stiffness as a stiffness per unit nominal contact area. In this case, the normal stiffness per unit area κ_n would be given by the rate of change of mean normal pressure p_m with respect to change in normal separation:

$$\kappa_n = \left| \frac{dp_m}{dd} \right|. \quad (1.3)$$

Similarly, a tangential contact stiffness per unit nominal area κ_t may be defined as the rate of change of mean tangential traction q_m with respect to change in tangential separation:

$$\kappa_t = \left| \frac{dq_m}{d\delta} \right|. \quad (1.4)$$

These are, of course, instantaneous tangent stiffness definitions: secant stiffness values are also commonly reported.

1.2 Aims, objectives, and structure of the thesis

In a sense, this thesis is unified by the hysteresis loop since the work investigates various aspects of friction and contact stiffness relevant to the fretting problem. The overarching aim is to be able to develop accurate predictive models of these parameters (i.e. from knowledge of the material properties, surface topography, and macroscopic geometry of the joint etc.). This has been recognised as being a ‘grand challenge’ problem in mechanics by recent workshops on joint modelling (Segalman *et al.* [5]). However, a necessary step on this path is to more fully understand the behaviour of these parameters, their physics, and how they are affected by other joint parameters. The aim of the present study is to tackle both of these challenges by a series of experimental and modelling investigations. Attention is mainly focused on developing physical understanding, but in the case of contact stiffness, for example, considerable effort has been invested in

evaluating measurement techniques (since before a parameter can be understood, agreement has to have been reached about the most suitable and reliable ways to measure it). The materials and scope of the study have been chosen to be representative of dovetail blade-roots in gas turbine engines, since this is a severely loaded contact which is expected to be one of the first areas to benefit from the improved understanding and predictive capabilities (hence the titanium alloy Ti-6Al-4V and the nickel alloy Udimet 720 are used throughout). The specific ways in which this thesis addresses these challenges is now presented with reference to the appropriate chapters which follow:

Chapter 2 presents a review of the literature concentrating closely on topics covered by the present work. Literature on gross-slip fretting, friction modelling, and contact stiffness is discussed.

Chapter 3 undertakes some preliminary work on friction and wear behaviour in gross-slip fretting. It makes comprehensive use of new 3D surface topography parameters and techniques for use in surface analysis in fretting-wear tests. The effect of materials selection and roughness on the fretting response is also addressed.

Chapter 4 investigates the physical origins of a phenomenon (which is observed in Chapter 3 and in the literature) whereby the friction force varies during individual gross-slip fretting cycles. This variation usually takes the form of a significant increase in the friction force as sliding proceeds from one extremity to the other. In-line and torsional fretting tests are performed to investigate the phenomenon, and both qualitative and quantitative analysis of post-test wear scars are undertaken to further probe its cause. A simple model to explain the behaviour is also proposed.

Chapter 5 addresses the problem of friction modelling. An elastic-plastic asperity interaction model is developed which makes considerable advances over previous

modelling attempts. Greater overlap between asperities, inclusion of an adhesional shear strength at the asperity junction interfaces, and the incorporation of asperity damage are among the improvements. Coefficients of friction are predicted using the model. Results from the asperity junction model are also incorporated into a stochastic surface contact model allowing friction predictions to be made when a given statistical distribution of asperities is present. Finally, the modelling predictions are compared to experimental results.

Chapter 6 is a detailed study of the contact stiffness of real rough interfaces. Observations are made about contact stiffness in both fretting and non-fretting environments. In the fretting case, a finite element model is developed to compare with tangential contact stiffness results in the literature determined by digital image correlation (DIC). The model is used to determine the relative contributions from the fretting interface, and from the bulk material between DIC measurement points, to the contact stiffness with the aim of isolating the true interface stiffness by a ‘springs in series’ type approach. A compliant interface layer is then included in the FE model in order to simulate the experimental results. This model allows the effect of nominal contact area on the contact stiffness to be investigated for the fretting-wear case. In addition, a comprehensive comparison is undertaken between contact stiffness measurements (normal and tangential) derived from DIC and ultrasound methods. Tests are undertaken where contact stiffness is concurrently measured in the same experiment by both ultrasound and DIC. In these tests, the effect of both normal and tangential load on the normal and tangential contact stiffness is investigated. Important differences between the DIC and ultrasound techniques are shown experimentally for the first time.

Chapter 7 draws overall conclusions from the work and indicates the future work required for further advancement.

Chapter 2

Literature review

2.1 Introduction

The literature review sets out the background research which has been undertaken over the years by various researchers in the specific areas of frictional behaviour addressed by this thesis. It is divided into three sections. The first, ‘Gross-slip fretting’, gives an overview of fretting wear, looks at the experimentally observed behaviour of friction in gross-slip fretting, and discusses surface analysis in fretting. The second, ‘Friction modelling’, addresses attempts to model friction and develop a physical understanding of the mechanism of friction. The third, ‘Contact stiffness of rough interfaces’ looks in particular at load-deflection and ultrasound based methods for measuring contact stiffness, and at the increased physical understanding of contact stiffness which these methods have facilitated. A summary and critical analysis section focusing in particular on shortcomings in the literature which are addressed by the present study is then given.

2.2 Gross-slip fretting

2.2.1 Introduction

This section discusses the literature relating to gross-slip fretting (the regime in which fretting wear occurs). An overview of some of the main developments in fretting wear is first given followed by a discussion of the literature specifically related to frictional behaviour and surface analysis in gross-slip fretting.

2.2.2 Overview of fretting wear

The first occurrence of what is now referred to as ‘fretting’ was reported in 1911 by Eden *et al.* [6] who noted the presence of brown iron oxide debris in the vicinity of the contact interface between machine grips and steel specimens where some small scale oscillatory motion had been occurring. It was not until 1927 that the first investigation dedicated to the study of fretting damage was carried out on steel specimens by Tomlinson [7]. Tomlinson was the first to use the term ‘fretting’ and coined the term ‘fretting corrosion’ for the phenomenon where iron oxide wear debris was induced by small cyclic interface slip in his tests. He recognised that small-amplitude slip was an essential factor in producing this particular type of damage. Tomlinson’s idea of ‘fretting corrosion’ viewed the process as a chemical one with oxidation being the main driver. However, Godfrey [8], Godfrey and Bailey [9], and Bailey and Godfrey [10] noted that similar wear occurred in the fretting of non-metals and noble metals. This development emphasised the importance of mechanical mechanisms and relegated oxidation as a cause to a secondary factor.

In a review of the subject in 1970, Hurricks [11] collated and interpreted the developments in the existing literature and put forward a unified view of the mechanism

of fretting wear. He divided the process into three stages: initial adhesion and metal transfer, the production of debris in a normally oxidised state, and the steady-state wear condition. In the initial stage, the protective oxide layer is broken apart by rubbing of the surfaces resulting in metal-to-metal contact which facilitates the formation of adhesion junctions (the idea of surface adhesion is discussed in more detail in Section 2.3.2). Wear debris is then created by plastic deformation of the junctions and fatigue cracking of newly formed work-hardened and oxidised surface layers. When enough debris is created, a protective intermediate layer is formed which promotes a steady-state wear evolution. Hurricks [11] also considered abrasion by the debris layer as a possible wear promoting factor, but viewed it as a more minor mechanism as the wear rate would be expected to increase if this was a dominating mechanism and this is not observed in practice (the wear rate in fact reduces due to the protection from the adhesion mechanism afforded by the debris layer).

In the 1980s further developments were made. There had been some confusion about why some fretting situations cause failure by detrimental propagation of fatigue cracks and why others instead cause gross wear of the surfaces. In 1988, Vingsbo and Söderberg [12] showed that the type of fretting damage which occurs depends on the fretting regime. In partial slip, fretting fatigue generally occurs, while in gross sliding, fretting wear dominates. Around the same time, researchers in Lyon (Godet [13] and Berthier *et al.* [14, 15]), developed the so called ‘third body approach’ to fretting wear. They focused attention on the layer of wear debris entrapped by the contact interface which they referred to as the ‘third body’. Essentially, they outlined why this third body was more important than previously thought. Previously, attention had been focused on the ‘source’ of the wear debris by adhesion, plastic deformation, fatigue cracking of surface layers and so on, but Berthier *et al.* [14] argued that the ‘sink’ of the debris (i.e.

how debris is removed from the contact) was probably more important. The process of fretting will have a particular capability for creating debris in the absence of any inhibiting factors, but it will be limited by any build-up of debris in the interface. Thus, the wear situation at the interface is ultimately governed by how quickly the debris is ejected from the interface. In conformal contacts, where it is difficult for debris to escape, the wear rate will be reduced by the third body build-up; whereas, for non-conformal contacts like a sphere-on-flat, debris is more easily ejected and the wear rate is less limited by the third body and more influenced by the rate of debris formation. Berthier *et al.* [14] viewed the fretting mechanism as being a competition between the initiation of damaging surface cracks and the formation of protective third bodies.

Some progress has also been made on modelling and quantifying fretting wear. The most widely used wear ‘law’ is that introduced by Archard [16] in 1953. He observed that the wear volume is approximately proportional to the product of normal load and total accumulated slip distance. The constant of proportionality in this relationship is known as the ‘Archard wear coefficient’ and has units of volume per unit local slip per unit contact pressure. Like the coefficient of friction, this wear coefficient has to be determined by experiment. Fouvry *et al.* [17, 18] developed another important way of understanding the wear mechanism using an energy approach. They noted that Archard’s model does not work when the friction coefficient is not constant and argued that it is more useful to consider the interfacial shear work as the significant wear parameter. Fouvry *et al.* [17, 18] observed that the wear volume is approximately directly proportional to the total frictional energy dissipated in a test and defined the constant of proportionality as the ‘energy wear coefficient’. The energy approach is a powerful tool and has been used to compare wear resistance of surface coatings (Fouvry *et al.* [17]). More recently, advances have been made in modelling fretting wear using the finite

element method. McColl *et al.* [19] incorporated a modified Archard wear ‘law’ into their 2D cylinder-on-flat model using a Fortran subroutine with ABAQUS finite element (FE) software. The predicted wear profiles were similar to those measured by experiment in terms of general shape, depth and width. Ability to predict wear profile evolution during fretting is quite a useful development since it allows other important properties which evolve with the contact geometry evolution to be studied. For example, in a study by Ding *et al.* [20], the approach has proven particularly useful in predicting the likely location, and number of cycles to nucleation, of fatigue cracks in mixed-slip and partial-slip regimes (where contact geometry also evolves). This was achieved by calculating a multi-axial fatigue damage parameter as the contact geometry evolved during the simulations.

The present thesis addresses issues specifically concerning the behaviour of friction and contact stiffness in gross-slip fretting, and also the application of advances in surface metrology to the fretting-wear problem. Having given a general overview of the main developments in fretting wear above, attention is focused in the next section on the literature related to the specific areas of frictional behaviour and use of surface metrology in gross-slip fretting (the literature on contact stiffness in fretting is discussed in Section 2.4). Throughout this thesis, the two materials used were the titanium alloy Ti-6Al-4V and the nickel alloy Udimet 720. It seems that little or no literature is available on fretting wear of Udimet 720, but there is a significant body of work available on Ti-6Al-4V. Therefore, where possible, the literature on fretting wear of Ti-6Al-4V will be particularly highlighted.

2.2.3 Frictional behaviour in gross-slip fretting

The literature here can be divided into two parts relating to: how friction evolves with number of cycles during a fretting-wear test, and how friction behaves in individual fretting-wear cycles.

Taking the friction evolution during fretting tests first, Mohd Tobi *et al.* [21], Everitt *et al.* [22] and Ding *et al.* [20] have all reported the evolution of friction coefficient with number of fretting wear cycles for a cylinder-on-flat arrangement for Ti-6Al-4V on Ti-6Al-4V. They found that the friction coefficient rapidly increased from values less than 0.1 up to 0.8 in the first few cycles, then underwent a gradual increase until a steady value of about 0.9 was reached after approximately 10,000–15,000 cycles. The initial rapid increase or ‘running in’ is attributable to the removal of surface layers such as an adsorbed gas layer and an oxide layer. The gradual increase up to the final steady value appears to be associated with a transition to a steady-state regime.

We turn now to the behaviour of the friction force in individual gross-slip fretting cycles. Simple models of frictional behaviour involving constant friction with sliding, such as that proposed by Amontons [3] and Coulomb [4], are frequently used to describe the measured tangential force hysteresis loops. However, an interesting feature is present in the hysteresis loops reported by many researchers, namely that the friction force varies during sliding between the two extremities of motion (most often, the friction force increases more and more rapidly as sliding proceeds resembling a *hook feature*). Figure 2.1 compares an ideal Amontons-Coulomb hysteresis loop (Figure 2.1a) with loops measured experimentally by Fouvry *et al.* [23] (Figure 2.1b). This behaviour (seen in Figure 2.1b), is usually not present in the initial fretting cycles, but develops as the test proceeds (Magaziner *et al.* [24]; Lee and Mall [25]; Elleuch and Fouvry [26]; Zhou and

Vincent [27]). It has been observed for a variety of material pairs and contact geometries: titanium alloy-on-titanium alloy with cylinder-on-flat contact geometry (Magaziner *et al.* [24]; Lee and Mall [25]; Everitt *et al.* [22]; Fouvry *et al.* [23]; Paulin *et al.* [28]), aluminium alloy-on-steel with the sphere-on-flat configuration (Elleuch *et al.* [26]), aluminium alloy-on-aluminium alloy for the sphere-on-flat arrangement (Zhou and Vincent [27]), and steel-on-steel with cylindrical punch-on-flat contact geometry (Rybiak *et al.* [29]). Very recently, Eriten *et al.* [21] observed the effect when carrying out fretting tests on both steel and aluminium lap-joints.

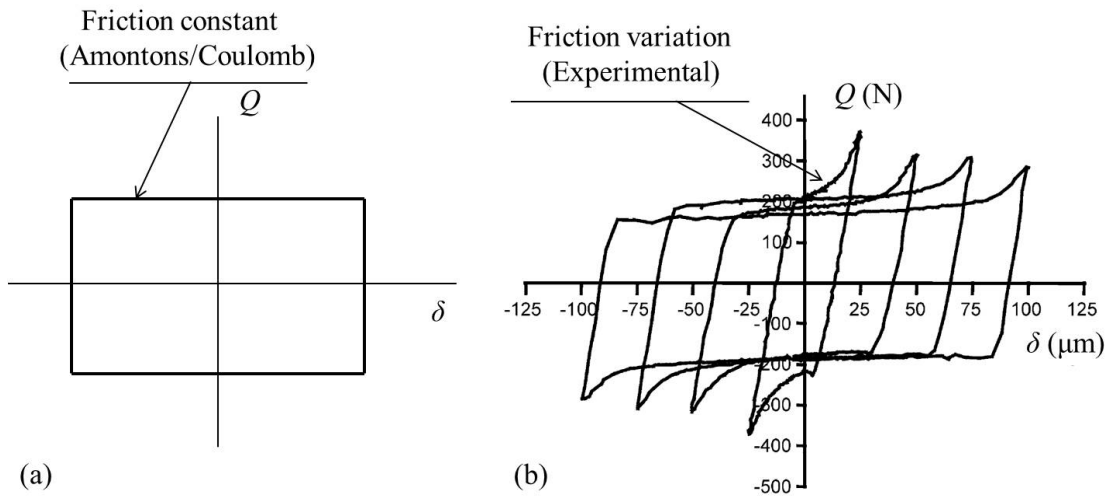


Figure 2.1: Comparison of: (a) an ideal Amontons-Coulomb frictional hysteresis loop with (b) hysteresis loops measured by Fouvry *et al.* [23] in cylinder-on-flat fretting of Ti-6Al-4V.

Since knowledge of the friction force during fretting is needed for predicting both the severity of wear and the damping characteristics, it is important to understand the mechanism behind the increasing friction phenomenon. Given that the applied displacements in most of the above studies are sinusoidal functions of time, the sliding velocity decreases as the point of motion reversal is approached. It is tempting, therefore, to ask whether the so called ‘velocity weakening’ effect (the increase in friction as velocity decreases) may be responsible for the phenomenon. Al-Bender *et al.* [30, 31]

and De Moerlooze *et al.* [32] have developed sophisticated friction models based on the physics of point masses on springs interacting with a rigid rough surface. The models simulate a wide range of frictional behaviour such as velocity weakening, frictional hysteresis, and stick-slip phenomenon. These studies show that a model incorporating velocity weakening can reproduce the commonly observed friction variation effect for periodic sinusoidal displacement oscillations. Other similar models developed by the dynamics/control community which show non-Coulomb friction during gross slip are outlined in Olsson *et al.* [33], Wojewoda *et al.* [34], and De Moerlooze and Al-Bender [35]. Lampaert *et al.* [36] observed the variation in friction force during slip in fretting experiments, and their data were later used in Al-Bender *et al.* [30] as experimental validation for the velocity effect captured by their friction models; however, no conclusive evidence has been provided to show that the phenomenon occurring in fretting tests is actually a ‘velocity effect’. In fact, this thesis will show that the phenomenon is almost certainly caused by wear-scar interaction.

The ‘velocity theory’, however, would also explain why the friction force in some loops can decrease slightly just after the beginning of the gross-slip phase, but then increase as the other extremity of motion is approached. This is because the velocity (for a sinusoidal displacement oscillation) increases from the point where sliding begins, to the point where the pad is in the mid-range, and subsequently decreases from here to the other extreme position. However, this explanation requires either a strong velocity weakening effect or large variations in velocity during a given fretting cycle. Further evidence against a velocity effect being the probable explanation is that, the friction variation effect is usually not present in the initial cycles of testing (Zhou and Vincent [27]; Elleuch and Fouvry [26]; Magaziner *et al.* [24]; Lee and Mall [25]; Lee *et al.* [37]). Rabinowicz [38] showed that friction only decreases slightly with large increases in

velocity for most metals (for titanium-on-titanium he noted a decrease from 0.6 to 0.5 in the friction coefficient as the velocity increased from 10^{-7} to 10^3 mm/s).

Others researchers (Berthier *et al.* [15]; Fouvry *et al.* [23]; Mohd Tobi *et al.* [21]; Lavella and Botto [39]) have suggested that the effect arises, instead, from wear-scar interaction effects. Such a mechanism explains why the friction increase is absent from the initial cycles. Fouvry *et al.* [23] and Lavella and Botto [39], suggested that the friction force increases because of interference between the macroscopic ends of the overall wear scars as the maximum displacement is approached, but provided no corroborating experimental investigation. They also noted that the effect is usually not observed on hard materials or coatings, and seems to be more prevalent in ductile materials. In fact, Lavella and Botto [39] noted distinct ‘hook’ features in their hysteresis loops for single-crystal nickel alloy (CMSX4) without a coating, but with a T-800 cobalt-chromium-molybdenum coating no ‘hook’ feature was observed after corresponding numbers of cycles of ball-on-flat fretting at 800 °C.

Paulin *et al.* [28] noted the effect in both a fretting experiment and a finite element model (with worn surface evolution) of the cylinder-on-flat arrangement, and hypothesised that the phenomenon might be caused by either the macro-interactions at the contact edges or at the ‘asperity scale’ within the fretting scar; however, no tests were carried out to investigate this. Berthier *et al.* [15] also noted the effect in experiments and theorised about debris compacting to form dense struts which adhere to one body, gouge a counter-face in the opposing body, and get jammed at the end of the stroke causing a significant increase in tangential force. Again, this was merely a hypothesis and no experimental proof was provided. Mohd Tobi *et al.* [21], who also used finite element methods of analysing fretting wear, predicted high plastic strains at the macroscopic wear-scar ends

and proposed that these regions may interact to produce an increase in tangential force as the extremity of motion is approached.

2.2.4 Surface analysis in fretting wear and reciprocating sliding studies

The consideration of worn surfaces in fretting and reciprocating sliding can take the form of mechanical, chemical, and topographical analysis. The analysis can be either qualitative where micrographs are visually examined for clues about the underlying wear processes, or quantitative where specific quantities are calculated such as, for example: hardness, surface roughness or oxygen content.

Taking surface topography first, the reporting of wear-scar photographs at various scales has always been common. For example, Magaziner *et al.* [40] analysed optical micrographs of full wear scars, and used scanning electron microscopy (SEM) images to focus in closer on smaller noteworthy regions of the surface. By simply examining the surface texture in the images, Magaziner *et al.* [40] were able to say, for example, whether wear was occurring by adhesion or by delamination etc. Obtaining surface coordinates has also been widespread: 2D profiles have been obtained (Stachowiak and Stachowiak [41]; Stemp *et al.* [42]; Wimmer *et al.* [43]; Lloyd and Noel [44]; Hayward *et al.* [45]) using such techniques as traditional tactile stylus profilometry (Waterhouse and Trowsdale, [46]; Lloyd and Noel [44]; Hayward *et al.* [45]), or laser profilometry (Stemp *et al.* [42] and Wimmer *et al.* [43]). These were historically more common, but today 3D profiles (Yoon *et al.* [47]; Menezes *et al.* [48]; Achanta *et al.* [49]; Guangxiong *et al.* [50]; Li *et al.* [51]; Howell *et al.* [52]; Kubiak *et al.* [53]; Yuan *et al.* [54]) can easily be obtained by such techniques as confocal laser scanning based microscopy (Yuan *et al.* [54]), interferometry based microscopy (Achanta *et al.* [49]; Howell *et al.* [52]; Kubaik *et al.* [53]), atomic force microscopy (Yoon *et al.* [47]; Achanta *et al.* [49]; Li *et al.* [51]),

and stylus tip profilometry with area scanning capabilities (Guangxiong *et al.* [50]). Those that obtained 2D profiles were inevitably only able to calculate 2D roughness parameters for a line profile such as the average surface roughness R_a and standard deviation of surface heights R_q . Researchers with full 3D surface profile data can calculate 3D roughness parameters, but most of the studies cited have only used a small number of the new 3D areal roughness parameters (such as average areal roughness S_a or areal standard deviation of asperity heights S_q) set out in the new 2012 ISO standard on 3D surface topography [55].

Some of the above studies investigated the changes in surface roughness occurring during a reciprocating sliding/fretting-wear test (Stachowiak and Stachowiak [41]; Stemp *et al.* [42]; Wimmer *et al.* [43]; Guangxiong *et al.* [50]; Hayward *et al.* [45]; Howell *et al.* [52]), but the results were quite variable and dependant on the specific contact conditions and materials involved. Some found a reduction in roughness as the surfaces became more polished by the rubbing action (Stachowiak and Stachowiak [41]; Hayward *et al.* [45]; Howell *et al.* [52]), and others an increase as the surfaces became roughened by wear (Stachowiak and Stachowiak [41]; Wimmer *et al.* [43]; Guangxiong *et al.* [50]; Yuan *et al.* [54]). The effect of initial surface topography on friction and wear in fretting/reciprocating sliding has also received some attention. Yoon *et al.* [47] found that the material combination had a far greater effect on the evolution of fretting wear than initial surface roughness for a sphere-on-flat contact using stainless steel, with and without, a gold coating. They also observed somewhat higher friction for the tests with rougher surfaces.

Menezes *et al.* [48] very recently performed reciprocating sliding tests on spherical tipped Al-4Mg alloy pins sliding on flat steel surfaces with three surface types: unidirectional ground, 8-ground (prepared using abrasion in a figure of '8' pattern), and random

surfaces. The unidirectional surfaces in the configuration perpendicular to sliding gave the highest friction, followed by the 8-ground, the unidirectional surfaces in the parallel configuration, and the random surfaces which gave the lowest friction. The difference in friction coefficient at the beginning of the tests between the surface type with the highest and lowest friction was almost 0.4, but after 20 cycles the difference was only about 0.05. The authors attributed this to a fairly rapid ‘rubbing out’ of the initial roughness by the wear process. Within each surface type, five different average roughness values covering the range 0.1–0.6 μm were used and these showed little difference in friction coefficient.

Kubaik *et al.* [53], also in a recent study, investigated the effect of initial surface roughness (covering the range of areal average roughness values S_a of 0.09 to 4.15 μm achieved by milling and polishing) in tests where the applied fretting amplitude was gradually stepped up through the partial-slip regime and into the full-sliding regime. They used steel-on-steel with the sphere-on-flat arrangement. In contrast to Yoon *et al.* [47], they found that the smoother surfaces had higher friction. The reason for this may lie in the interplay between high roughness values which tend to increase the plastic deformation contribution of friction, and low roughness values which tend to produce a higher real area of contact, thereby increasing the adhesion contribution to the friction. This idea is explained more fully in Section 2.3.2. Like Menezes *et al.* [48], Kubiak *et al.* [53] also found that after a sufficient number of cycles, all tests which had initially had different surface topography, converged on approximately the same value of the friction coefficient, again indicating that the fretting wear has erased the impact of the initial roughness. These authors also found that initial roughness had an effect on the critical applied amplitude which causes transition from partial slip to gross slip: with smoother surfaces delaying the transition.

Mohd Tobi *et al.* [21], Everitt *et al.* [22], and Magaziner *et al.* [40] have all examined post-test wear scars from cylinder-on-flat gross slip fretting tests on Ti-6Al-4V. Each of these works noted the existence of a W-shaped wear scar. Magaziner *et al.* [40] suggested that the two valley-regions in the 'W' shape might have been formed due to a 'digging-in' action owing to slight pad rotation at the extremities of the fretting motion. Mohd Tobi *et al.* [21], on the other hand pointed to the fact that the accumulated plastic strain distributions determined from their continuum FE wear modelling approach (using kinematic hardening plasticity) were also of a 'W' shape when plotted across the contact width. However, their FE models (using a modified Archard wear 'law') predicted 'U' shaped wear scars rather than the experimentally observed 'W' shape.

Everitt *et al.* [22] specifically examined the near-surface chemical and mechanical properties of the Ti-6Al-4V wear scars. They sectioned the flat specimens in the direction parallel to sliding and found that the wear debris existed as a compacted layer structure on the top of the Ti-6Al-4V substrate. They noted that this layer was comprised of two constituent sub-layers: one about 40 μm thick next to the substrate, and the other about 1 μm thick at the free surface. They performed an energy-dispersive X-ray spectroscopy (EDX) analysis of these two layers, as well as the underlying substrate, and found that the fractions of aluminium and titanium present drop off significantly with increasing depth from the surface. By contrast, the proportion of oxygen was found to increase as the free surface was approached with a peak occurring in the 1 μm top-surface layer. This suggests that the debris at the surface was highly oxidised.

Everitt *et al.* [22] also investigated the mechanical properties of the debris layer using nano-indentation tests and found that the hardness (and therefore, the yield stress) of the compacted oxidised debris was about double that of the substrate, while the Young's modulus was similar to that of the underlying material. In the central part of the wear

scar, just below the debris layer, Everitt *et al.* [22] observed deformation of the grains indicating plastic deformation and also noted the existence of what is known as the tribologically transformed structure (TTS). The idea of the TTS has been outlined in detail in works such as Sauger *et al.* [56] who also observed its occurrence in fretting wear of Ti-6Al-4V. The TTS is formed by a transformation of the microstructure near the top of the substrate to a nano-crystalline structure corresponding to the chemical composition of the bulk material, but made entirely of the more stable structure according to the equilibrium diagram. Sauger *et al.* [56] showed that this re-crystallisation begins to form very early on in fretting tests (after less than 10 cycles), and its initiation is dependent on the investment of a critical dissipated energy due to severe plastic deformation. It is the material in the TTS region that often forms the wear debris, which subsequently becomes part of the compacted debris layer discussed above.

2.3 Friction modelling

2.3.1 Introduction

In this section, some of the main theories and models which have been put forward to explain the phenomenon of friction are outlined. The discussion begins with the first observations of the empirical friction ‘laws’ and charts some of the major developments in reaching our present day understanding of friction. The more historic, fundamental and well established theories of friction are first considered. The discussion then focuses on one modelling approach in particular: asperity modelling.

2.3.2 Theories of friction

The study of friction begins with the discovery by Leonardo da Vinci (1452-1519) of what are now known as the first two ‘laws’ of friction:

1. The friction force during sliding Q is proportional to the normal force P via the coefficient of friction f :

$$Q = fP \tag{2.1}$$

2. The friction force is independent of the nominal area of contact.

These laws were later rediscovered and studied in more detail by Amontons [3] and Coulomb [4] in their 1699 and 1785 publications, respectively. Hence, the laws are commonly referred to as the Amontons-Coulomb laws of friction. Coulomb [4] added the observation that the friction force is constant during sliding, and this is often referred to as the ‘third law’ – though (as we have seen in Section 2.2), it is much less widely satisfied than the first two laws. These first two laws, which are simply based on physical observation, apply (at least approximately) to a remarkably vast range of

engineering contacts. An explanation of their physical origins, and hence the ability to predict friction coefficients, has been less comprehensive, and much work remains to be done.

Early researchers such as Coulomb [4] put forward the so called ‘roughness hypothesis’ to explain the above laws. This theory involved the interlocking of asperities where the friction force was assumed to be derived from the forces required to move asperities up and over other asperities. It is clear that Coulomb also considered the possibility of molecular ‘adhesion’ or bonding between surfaces as a possible explanation for the friction force but rejected it on the grounds that, if this were the case, friction should be proportional to nominal area rather than being independent of it as the second law demands. However, a major flaw in the ‘roughness hypothesis’ was noticed by researchers such as Leslie [57] who noted (in 1804) the failure of the theory to explain the energy dissipation of friction: the problem being that the potential energy invested in sliding up an asperity would be recovered in sliding back down. The same problem would apply to any attempt to attribute friction to elastic deformation of asperities.

Further impetus for a need to adopt a different view came early in the twentieth century (1919) when Hardy and Hardy [58] showed that the introduction of thin contamination or lubricant layers could have a large effect on the measured friction: if friction was due to the roughness hypothesis then surely the introduction of such thin layers would have little effect. The ‘area problem’ which had dissuaded earlier researchers from pursuing the adhesion theory was finally resolved by the work carried out by Holm [59], Bowden and Tabor [60], and Ernst and Merchant [61] between 1938 and 1940. They realised, by taking electrical resistance measurements, that the actual or ‘real’ area of contact between surfaces under static loading was much less than the nominal contact area (see Figure 2.2) and seemed, in fact, to be proportional to normal loading.

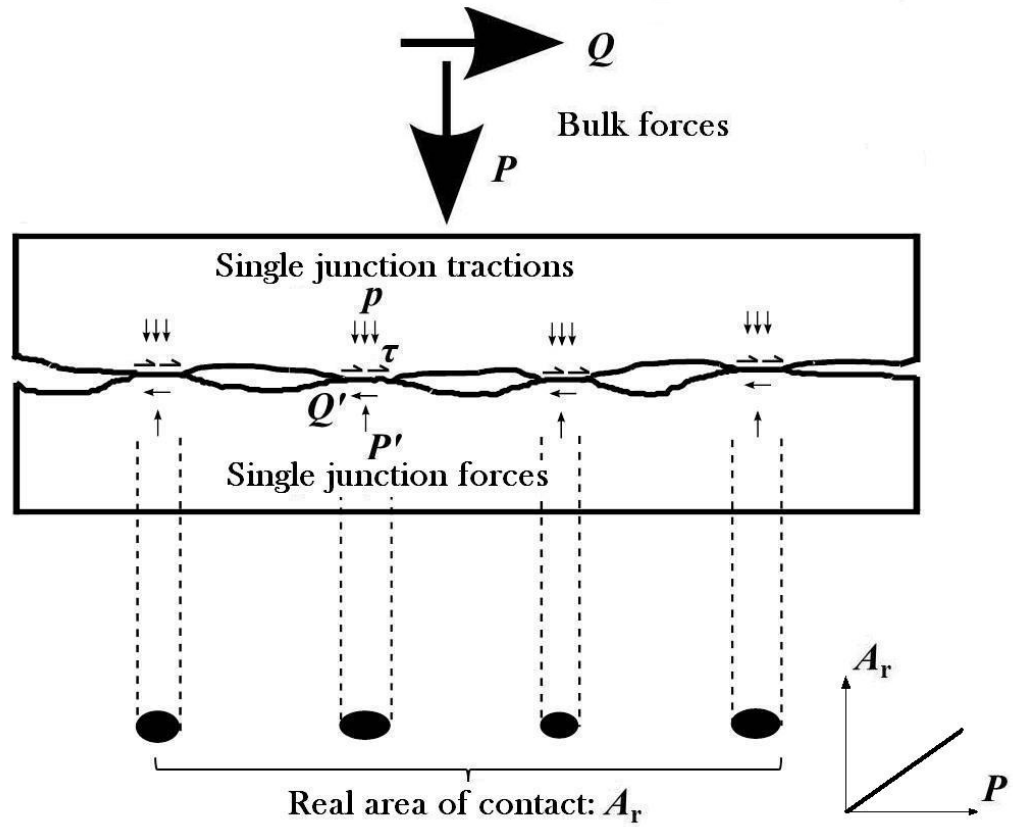


Figure 2.2: Schematic diagram illustrating the difference between real area of contact and nominal area of contact. The bulk contact forces, and the forces and tractions at a single asperity junction are illustrated. Also depicted is the direct proportionality between real contact area and normal load.

Bowden and Tabor [62] developed a theory based on plasticity to explain this observation. If p_m is the mean normal stress acting on the total real area of contact A_r , then, for a given normal load: $P = p_m A_r$ (see Figure 2.2). Since the real contacts were found to be plastic due to the high local pressures at individual asperities, p_m was identified with the material hardness p_h , giving P proportional to A_r as follows:

$$P = p_h A_r. \quad (2.2)$$

Bowden and Tabor [62] also developed an adhesion theory of friction based on the idea that, under the high pressures involved, cold welding between asperities, dictated by short range atomic forces, would occur. In order to move one surface over another, a mean

shear stress q_m acting over the real contact area would be required to yield these junctions. Therefore, the limiting friction force $Q = q_m A_r$ (see Figure 2.2). They then hypothesised that the mean shear stress would be equal to the bulk yield stress in shear τ_Y of the softer of the two materials involved in the interaction. Therefore, the total friction force Q would be:

$$Q = \tau_Y A_r. \quad (2.3)$$

Eliminating A_r from Equations 2.2 and 2.3 would then give:

$$Q = \frac{\tau_Y}{p_h} P. \quad (2.4)$$

Since both q_m and p_m are now given by material constants τ_Y and p_h respectively, Equation 2.4 immediately satisfies both laws of friction: proportionality with normal load and independence from nominal area. Perhaps the most striking outcome of this analysis is that it is the proportionality of normal load with real contact area (owing to plastic deformation) that explains the proportionality of normal load and friction force.

Prior to a study published in 1957 by Archard [63], it had been thought that proportionality of friction force with normal load could only be explained by recourse to Bowden and Tabor's plastic deformation in the way just described. However, Archard [63] showed that elastically deforming rough contacts could also satisfy proportionality of real contact area with normal load. Previous to Archard's paper, elastic deformation was presumed to be inconsistent with direct proportionality of real contact area and load, since the theory for a single Hertz contact (such as a sphere-on-flat) predicts that area of contact will be proportional to the normal load raised to the power $2/3$.

In his theoretical analysis, Archard took an elastic sphere contacting a flat and added a pattern of small spheres of much smaller radius to the spherical surface. He then added

further patterns of even smaller spheres to those new spheres. He found that, with each additional layer of ‘asperities on asperities’, the power in the relation between real contact area and normal load departed from $2/3$ and approached ever closer to unity (i.e direct proportionality). Archard had discovered an important property of what would today be called fractal surfaces (where surface patterns are scale invariant), although he did not use the word ‘fractal’. He realised that the key to the proportionality of friction force and normal force for elastic contacts was the formation of new contact areas with increasing normal load, thus requiring a multi-contact interface. Archard’s elastic experiments on crossed rough Perspex and metal cylinders verified his theoretical conclusions: he found that the friction coefficient was constant when the loads were low enough so that the contact was a multi-contact interface, but followed the non-constant curve consistent with Hertzian behaviour for higher loads (since the contact had become a single Hertz contact area).

Greenwood and Williamson [64] built on Archard’s work in their seminal 1966 paper. They developed a multi-asperity elastic model of a rough interface based on a surface of elastic non-interacting spherical asperities (which could have a prescribed distribution of asperity heights) contacting a rigid flat. Their work showed that for an exponential distribution of asperity heights, real contact area was exactly proportional to normal load, and for a Gaussian distribution (typical of many engineering surfaces), the relationship was close to direct proportionality. Their model was not fractal like Archard’s single fractal asperity, but they also concluded that the formation of new contact areas with increasing normal load was required for direct proportionality. In fact, for an exponential distribution of asperity heights, as normal load is increased, new (smaller) contacts are generated and existing ones are enlarged so that the average contact size remains exactly constant. Archard and Greenwood and Williamson’s work show that for elastically

contacting surfaces, it is the statistics of the multi-contact interface alone which determines whether real contact area is proportional to normal load. Therefore, this linearly proportional real contact area versus load behaviour can occur for elastic, as well as, plastic multi-asperity surfaces. When combined with the adhesion theory of friction, Amontons-Coloumb friction is then satisfied.

It was noted, however, that firstly: Equation 2.4 gave friction coefficients which were lower than those achieved in practice (Equation 2.4 predicts $f \approx 0.16$ for steel which is low compared to typical measured values in air of around 1) and second: that p_h and τ_Y cannot be treated as independent yield criteria. Therefore, McFarlane and Tabor [65] accounted for this by using the von Mises yield criteria, which, applied to arbitrary asperity geometry might be of the form:

$$\alpha p^2 + \beta \tau^2 = \gamma \sigma_Y^2, \quad (2.5)$$

where p and τ are the normal and shear stress, respectively, at a given asperity junction, σ_Y is the bulk yield stress and α , β and γ are coefficients to be determined. The distinction made here between contact forces and tractions at the asperity junction level, and those on the bulk level is illustrated schematically in Figure 2.2. McFarlane and Tabor [65] then used their von Mises criterion to develop a junction growth model to explain why real friction coefficients are often greater than those given by Equation 2.4. They reasoned as follows: initially when the normal load is applied in the absence of tangential loading, the asperities will deform plastically, and the imposition of an increment of tangential loading would then cause the junction to flow plastically in the tangential direction, thereby increasing the area of contact. An increased area of contact would then decrease the normal pressure p so that an increase in τ would be required to maintain yielding of the junction (i.e. slip) according to Equation 2.5. This process

obviously increases the friction coefficient above that predicted by Equation 2.4. With soft metals, this can lead to excessive friction coefficients where a bond is formed under a small normal pressure. This has been observed by Bowden and Tabor [62] for indium sliding on steel. In cases like this, accompanying normal adhesion associated with the local bonding has been directly measured and found to be responsible for the high friction coefficients observed.

Many were slow to accept the adhesion theory of friction and various questions were raised, such as, why there is no pull-off force required to separate everyday engineering frictional joints, when the normal load is removed. This is explained by noting that released elastic stresses break the adhesion bonds for harder materials [38]. Apart from the direct observation of pull-off forces with softer materials, there have been a number of other experimental verifications of the adhesion theory. Bailey and Courtney-Pratt [66] showed that atomically smooth surfaces of mica, produced by cleavage, show very high friction. Bowden and Tabor [62] found that even the thinnest films of surface contamination (such as adsorbed gases, oxide layers and lubricant layers) vastly reduce the friction from its clean metal-to-metal value. These findings are clearly inconsistent with the notion that friction is solely due to asperity deformation and roughness – they strongly suggest surface bonding. More recent studies (Carpick *et al.* [67] and Szlufarska *et al.* [68]) using atomic force microscopy have also detected the existence of an asperity level interfacial shear strength associated with adhesion of the asperity junction.

Adhesion is now widely accepted as a significant player in the generation of friction, but as Suh [69] points out: it does not provide a complete explanation. According to Suh, the friction force is mainly attributable to three processes: adhesion, asperity deformation and ploughing by third bodies. These factors contribute to friction in different proportions depending on variables such as material properties, surface geometry and

environment etc. An example of this complex interaction can be seen by the way in which surface roughness affects friction. Rabinowicz [38] showed that the variation of friction coefficient with average roughness is roughly U-shaped with low roughness values giving rise to high adhesive friction because of the greater area of contact, intermediate roughness values giving a plateau region of moderate friction and high roughness values giving rise to high friction as a result of increased asperity deformation.

2.3.3 Asperity modelling

The preceding section has given a detailed explanation for the proportionality between friction force and normal load. The shearing of adhesion bonds together with bulk deformation of asperities was put forward to explain the origin of friction. However, it is not entirely clear how this plays out on the asperity scale. It is clear that friction is a dissipative process, but how much frictional dissipation is due to plastic deformation of asperities and how much is due to the breaking of adhesion bonds is unclear. In any case, more detailed models are required to allow prediction of reasonable friction coefficients. It seems natural to start this process with the modelling of asperity interactions. We look now at the attempts that have been made to predict overall friction coefficients from data derived from single asperity junction models.

It was shown above that Bowden and Tabor [62] had realised that the average normal and tangential stresses acting on the real area of contact, at the beginning of, or during sliding, were not constant flow stresses but were governed by a continuum plasticity yield criteria. Therefore, how one might determine these stresses was largely unknown. However, Green [70] made the observation that: during steady sliding, the surfaces must move parallel to each other. The implication of this observation for a single asperity junction is that the asperities cannot separate or move closer together. According to

Green, this is not an assumption but is implicit in the concept of a steady state, and it is this condition which determines the normal and tangential stresses at a junction p and τ , respectively. Green supposed, from a study of plasticine asperities, that each junction would pass through a cycle of formation, deformation and fracture. He proposed a method of predicting an overall friction coefficient from tangential and normal force data derived from the life cycle of a single asperity junction. Green assumed that since the overall contact would consist of many such asperity junctions at different stages of development, the average normal and tangential forces $\int P' d\delta/\Delta\delta$ and $\int Q' d\delta/\Delta\delta$, respectively, derived over the complete evolution of a single junction, would therefore represent the averages $\bar{P}$ and $\bar{Q}$ for the entire contact at any given instant (where P' and Q' are the total normal and tangential forces for a single asperity junction at a given point in its evolution and δ is the relative distance moved). Then at any instant during steady sliding, the overall friction coefficient f would be given by $\bar{Q}/\bar{P}$.

Green [70, 71] proceeded to analyse asperity junctions as 2-D wedge shapes (Figure 2.3) using slip-line field theory for both plane stress and plane strain using the assumption of a rigid perfectly plastic material. He imagined the wedge angles changing as deformation proceeded, but he only found theoretical solutions for the wedge angles which would be typical of the initial stages of deformation. For this reason, and because he could not say when fracture of the junction might occur, he was unable to derive any quantitative friction coefficients. Using his partial solution and his plasticine experiments, he did, however, develop a qualitative picture of how P' and Q' might vary during a typical interaction.

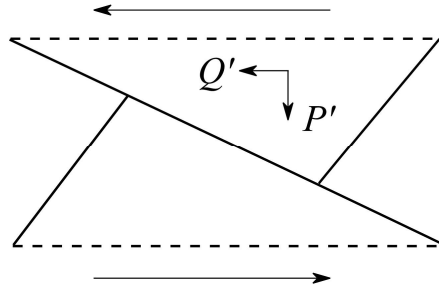


Figure 2.3: 2D wedge type junction, after Green [70].

Greenwood and Tabor [72] performed a series of experiments to investigate Green's analysis by shearing large scale metal wedge shaped models. They investigated junctions with strong adhesion (single-piece arrangement), weak adhesion (two-piece junction) and lubricated junctions (two-piece lubricated) and found results which agreed with most of Green's qualitative analysis. The resulting friction coefficients were typical of what might be expected: values of between 3 and 10 for the single-piece strong adhesion arrangements, values of the order of unity for two-piece metal junctions with oxide layers and values of 0.05–0.22 for soap lubricated two-piece junctions. For lubricated junctions, where adhesion is negligible, they concluded that plastic deformation of the sliding metals may contribute an appreciable part of the observed friction. Another conclusion of these early works on asperity modelling was that brittle material junctions, which fractured earlier in the junction life cycle, would give lower friction than ductile junctions, which by surviving for longer, allowed the normal force to reduce relative to the tangential force.

Since the work of Green [70], many researchers (Challen and Oxley [73]; Edwards and Halling [74]; Suh and Sin [75]; Torrance *et al.* [76]) have pursued the 2D wedge-shaped asperity junction model using tools such as slip-line field theory and finite-element (FE) analysis to determine the force evolutions. However, these studies have some significant limitations: they consider asperities to be wedge-shaped with rigid-perfectly plastic

behaviour, whereas real asperities are smoothly curved and have elastic-plastic properties. To address this shortcoming, researchers began to use finite element methods to analyse elastic-plastic cylindrical and spherical asperities, and to derive predicted friction coefficients based on the method outlined by Green [70].

Tangena and Wijnhoven [77] were the first to apply the finite-element method to model the sliding interaction of two cylindrical elastic-plastic asperities sliding under constant separation. In their study, they moved a rigid cylinder through a soft elastic-plastic strain hardening asperity with the properties of aluminium. They ran the model for three different asperity overlaps for both the frictionless case, and the case where an Amontons-Coulomb friction coefficient of 0.1 was imposed at the interface. They found that, in both the frictionless and frictional cases, the predicted friction coefficient increased with asperity overlap.

Faulkner and Arnell [78] were the first to develop a 3D elastic-plastic asperity interaction model using the finite-element method (Figure 2.4). Their model involved two elastic-plastic strain hardening spherical asperities whereby an upper softer asperity (with the properties of aluminium) was forced over a harder stationary lower asperity (with the properties of high strength steel) for three different overlaps. The model was run, both under frictionless conditions, and with a small interfacial friction coefficient ($f = 0.1$). For the maximum asperity overlap investigated (7.5% of the radius), using the model with $f = 0.1$, the maximum predicted friction coefficient calculated by Green's method was 0.187. The results were then incorporated into a statistical model to allow different asperity junctions to have different overlaps: this produced a plot predicting friction coefficient as a function of the ratio of standard deviation of asperity peak heights to asperity radius.

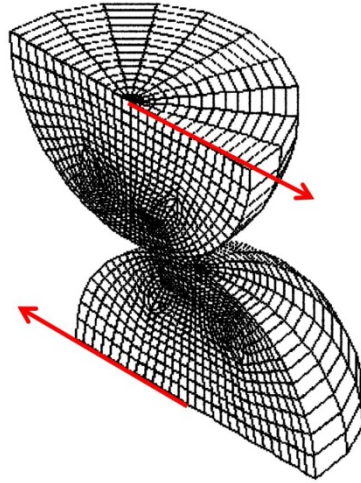


Figure 2.4: Finite element asperity contact model used by Faulkner and Arnell [78].

Jackson *et al.* [79] developed both a finite element model and a semi-analytical model of elastic-perfectly plastic frictionless spherical asperity interaction, and used these to formulate empirical equations describing the average tangential and normal forces resulting from sliding interaction. Their asperities had a Young's modulus of 200 GPa and a yield strength of 1000 MPa (i.e. high strength steel). The evolution of the normal and tangential junction forces throughout the lifecycle of an interaction where the overlap was 1% of the radius is given in Figure 2.5. They reasoned that if the asperities were elastic, then the friction coefficient as calculated by Green's method would be zero. This is because the work done in deforming the asperities is recovered as the deformed asperities move out of contact. Their results showed that 'elastic rebound' also occurred with elastic-plastic asperities, but to a lesser extent, so that a non-zero average tangential force (and thus a non-zero friction coefficient) results. This can be seen in Figure 2.5b where the work recovered in asperity rebound (area of the negative peak) is less than the work done in deforming the asperity (area of the positive peak). Jackson *et al.* [79] also performed a parametric study showing that the effective friction coefficient between asperities increases with elastic modulus and asperity overlap, and decreases with yield

strength. Figure 2.6 shows the plot of predicted friction coefficient versus normalised asperity overlap obtained by Jackson *et al.* [79]. For the largest overlap investigated (2.5% of the radius), it can be seen that the predicted friction coefficient was only 0.043.

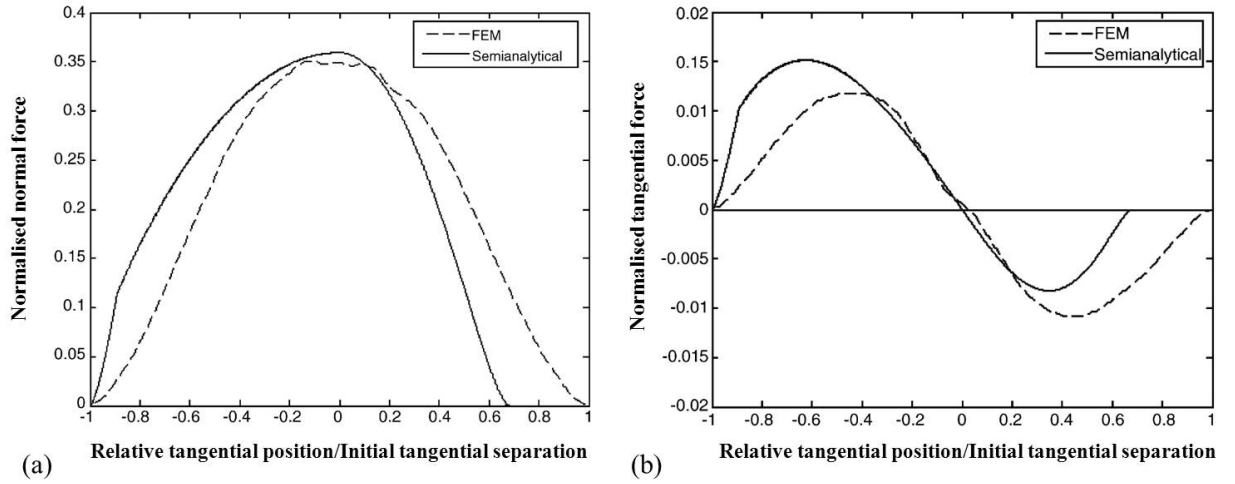


Figure 2.5: Typical evolution of (a) normal, and (b) tangential asperity junction forces (normalised by the corresponding maximum Hertz force) obtained by Jackson *et al.* [79] for an overlap of 1% of the radius. The asperities each had a Young's modulus of 200 GPa and a yield strength of 1000 MPa.

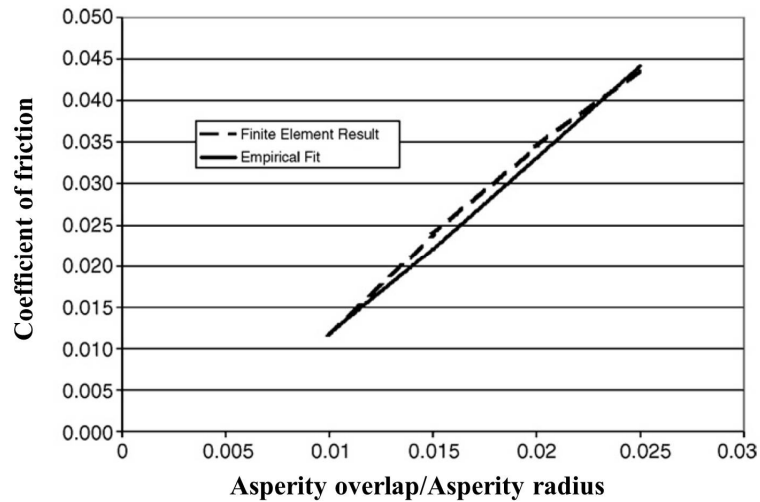


Figure 2.6: The effect of asperity overlap on the friction coefficients predicted by the modelling work of Jackson *et al.* [79] for asperities with a Young's modulus and yield strength similar to those of high strength steel (i.e. 200 GPa and 1000 MPa, respectively).

A study by Vijaywargiya and Green [80] examined trends in deformation, reaction forces, stresses, and energy losses as a function of interference and sliding distance for elastic-perfectly plastic steel cylindrical asperity junctions. A study by Boucly *et al.* [81] addressed many of the same issues for spherical asperities with the same material properties. However, these authors [80, 81] did not calculate effective friction coefficients for the complete asperity interactions. A very similar study, which does report the overall friction coefficients, was conducted by Moody [82], who expanded on the modelling carried out by Boucly *et al.* [81] using spherical asperities. Moody derived results for both a frictionless interface and one with a friction coefficient of 0.3. For steel-on-steel at the highest overlap investigated (1.4% of the radius), the predicted friction coefficient was 0.012 for the frictionless case.

There are two main problems with the FE modelling work described above, namely: the predicted friction coefficients are very small and are more representative of lubricated contact rather than those of metals (which tend to give friction coefficients in the range 0.2–1.5 when tested in air [83]). This is partly because, as discussed above, researchers have only analysed asperity junctions with relatively small overlaps (due to the limitations of the implicit finite element method). Secondly, many of the research works described have attempted to model tangential adhesion by enforcing a small Amomtons–Coulomb friction coefficient at the interface, with the aim of obtaining more realistic friction coefficients. Since this is the very ‘law’ for which we are hoping to develop a physical understanding, it seems unreasonable to incorporate it in a friction model.

2.4 Contact stiffness of rough interfaces

2.4.1 Introduction

In this section the literature relating to the experimental investigation of contact stiffness of rough interfaces is discussed. Methods for measurement of the parameter, as well as the physics which has so far been uncovered experimentally, are reviewed. The definition of contact stiffness has already been outlined in detail in Chapter 1, but essentially normal contact stiffness is the ratio of an incremental increase in normal force to the corresponding change in normal surface separation caused purely by the rough interface, while tangential contact stiffness is defined similarly but applies to an incremental tangential force and a corresponding tangential separation (of course tangential contact stiffness applies only to the pre-sliding regime). This review is concerned particularly with two methods of measurement of contact stiffness: those based on load-deflection data and those based on ultrasound measurements. Both methods have been used to investigate the properties of contact stiffness in rough surface contact. Developments in the measurement techniques are first reviewed for each method, and then a discussion of the physical insights which they have facilitated is given.

2.4.2 Load-deflection method

In this method the load applied to the contact is measured while the corresponding relative displacement is also measured. The slope of the load-deflection graph then gives the required stiffness at any particular load. Measurement of the load is relatively straightforward (either the applied loads are known, or they are measured by a force measuring device such as a load cell), but measurement of the displacement caused exclusively by the ‘real rough interface’ is a more challenging matter. This is because the

measurement will inevitably include some displacement due to bulk material deformation. The experimental investigations that have measured load-deflection data can be divided into two types: those that measured relative displacements remotely (i.e. not close to the contact and often on some other part of the experimental apparatus) [84-92], and those that measured displacements from the contacting specimens themselves at points more local to the contact [39, 93-97].

Discussing first the remote measurement works, Johnson [84] investigated tangential load-deflection behaviour using a torsional rig involving three balls arranged in a circular disk (the balls were in contact with a flat surface) and used both a microscope and a Rochelle salt crystal gramophone pick-up to track the displacement of points on the ball-holding disk at a radius further out than where the contacts were positioned. Courtney-Pratt and Eisner [85], also looking at pre-sliding behaviour of the ball-on-flat arrangement, used a multiple beam interferometer to track the displacements of a flat silvered glass plate which was attached to one of the specimens. O'Connor and Johnson [86], measured displacements by means of a pneumatic displacement head which was attached to one of the specimens and rested against the other at a position remote from the contact region. Fridrici *et al.* [89] used an extensometer positioned remote from the two specimens, and Ovcharenko *et al.* [90] used a proximity probe to measure displacement of one of the specimens by making contact with the lever responsible for applying tangential load to that specimen. All of these papers were using the ball-on-flat arrangement, but other arrangements have also been investigated in this way. Berthoud and Baumberger [87] measured displacements via inductive displacement gauges making contact with the upper specimen holding fixture. Fujimoto *et al.* [88] used differential transformers in contact with both specimens (two coaxial rings in flat-on-flat contact). Eriten *et al.* [91, 92] measured the tangential stiffness of a single bolt lap joint using a

laser nano sensor (LNS) reflecting from a mirror attached to one side of the upper half of the joint. Of the above authors, only Berthoud and Baumberger [87], Fujimoto *et al.* [88], Fridrici *et al.* [89] and Eriten *et al.* [91, 92] actually calculated contact stiffness as the other research works mentioned were concerned with studying various other aspects of contact using the load-deflection curves.

A significant disadvantage of the remote method is that a significant amount of bulk displacement can creep into the measurement either via deformation within the specimens, or via deformations arising from various rig fixtures (though some of the above authors will have taken great care to minimise such effects). To allow the measurement to focus on displacements arising solely from the rough surface interaction, it seems clear that local measurements from points positioned as close as possible to the interface (and on either side so that relative displacement can be measured) are required. Two methods which have achieved this aim are the laser Doppler vibrometry and digital image correlation (DIC) techniques.

Filippi *et al.* [93], Lavella and Botto [39], and Schwingshackl *et al.* [96] have each used a laser Doppler vibrometer to measure the relative displacement between two points positioned close to the interface on the two bodies in contact. In this technique, a laser beam is projected towards two points on either side of the contact interface. Specimen movement causes Doppler shift of the laser beam frequency which is directly related to the velocity of the specimens: with knowledge of the time taken, displacements can be easily calculated. The tests by Filippi *et al.* [93], and Lavella and Botto [39] were performed using the ball-on-flat arrangement while Schwingshackl *et al.* [96] used two crossed rectangular flats in contact. Contact stiffness was successfully reported in each of these three publications and the measurements were carried out during gross-slip fretting (in the partial-slip phase of the fretting cycles).

The digital image correlation technique was first used very recently to measure contact stiffness by Kartal *et al.* [95], and this was followed by further work by Kartal *et al.* [94] and de Crevoisier *et al.* [97]. The DIC technique involves obtaining high resolution digital images containing the interface (and the area local to each side of the interface) while load is applied. By comparing subsequent images to an image taken at the beginning of loading, displacements can be calculated on each side of the interface and then subtracted to give local relative displacement. A more detailed outline of the technique is given in Chapter 6 (Section 6.2). Each of these three papers used the DIC technique for calculation of tangential contact stiffness during fretting tests (in gross sliding for flat-and-rounded pads contacting a flat in Kartal *et al.* [94, 95], and in partial slip for a single bolt double lap joint in de Crevoisier *et al.* [97]) Figure 2.7 shows a schematic diagram comparing the local and remote methods of displacement measurement discussed above.

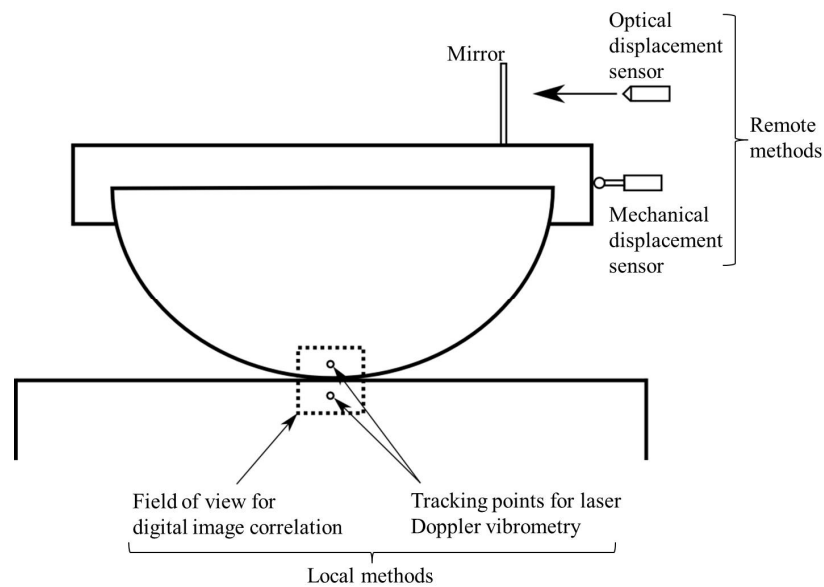


Figure 2.7: Comparison of local and remote methods for measuring load-deflection data.

Even with these local methods for determining load-deflection data there will still be some amount of bulk deformation (however negligible) occurring between the measurement points as it is difficult to position the points any closer than about 50 microns from the interface (since a certain minimum area on either side is needed for image correlation). A technique that purports to measure the contact stiffness of the rough interface itself without any bulk contribution (i.e. that stiffness which is the series difference between the rough contact and the smooth contact case) is the ultrasound technique, and this is discussed next.

2.4.3 Ultrasound method

Ultrasound is a vibration whose frequency is above the audible range. In a study attempting to use ultrasound to measure the real area of contact at an interface, Kendall and Tabor [98] realised that the amount of ultrasound reflected from a rough interface was directly related to the stiffness of the interface. Since a given contact stiffness can be achieved with a single large area of contact, or with a smaller area divided into a number of contact regions which are spaced apart, they concluded that the technique could not hope to directly determine the real area of contact. However, the ability to measure contact stiffness was a promising development and Tattersall [99] went on to further define and generalise how this might be done. Essentially these two early works proposed that the interface be considered as a region having the stiffness of a simple Hookean spring (see Figure 6.8 in Chapter 6). When an ultrasound wave passes to a region of different material stiffness (or density) through a perfect interface, some of the wave is transmitted through, while the remainder is reflected. The proportion of the incident wave which is reflected (called the reflection coefficient) is directly related to the stiffness (and density) of each material. Similarly, when an ultrasound wave is projected

through a rough interface, the contact stiffness, as well as the stiffness and mass properties of the two materials, will determine the reflection coefficient. Tattersall [99] used the ‘spring model’ to develop a simple formula (Equation 6.19) expressing the reflection coefficient in terms of the acoustic impedance of the two materials and the interface stiffness. The derivation of this formula is given in Chapter 6 (Section 6.4.2). With this approach, experiments are required to determine the reflection coefficient, and with knowledge of the acoustic impedance of the contacting materials, the contact stiffness per unit area can easily be calculated.

The spring model is, of course, a quasi-static model of the interface which ignores any dynamic effects and in the years following the introduction of the spring model by Kendall and Tabor [98] and Tattersall [99], there was some concern about whether a quasi-static model was appropriate. For example, Królikowski and Szczepek [100] pointed out that there had been some experimental evidence (Królikowski *et al.* [101] and Królikowski *et al.* [102]) to suggest that the stiffness values calculated using the spring model may not be entirely independent of the frequency of the ultrasonic excitation. It was for this reason that some researchers developed some additions to the spring model to account for dynamic effects. Baik and Thompson [103] incorporated a mass term into the model making it a ‘spring-mass’ system but showed that this had a negligible effect on the resulting stiffness values. Margetan *et al.* [104] also examined the effect of a mass contribution: they concluded that the mass term would be insignificant for the type of interfaces occurring with common engineering surfaces. This is because, at a typical engineering interface, the gaps are generally very thin and contain low density air which means that the interfacial mass term can have little effect on the reflection coefficient. Królikowski *et al.* [102] introduced two damping mechanisms into the ‘spring model’ formulation: a viscous friction damping term responsible for heat dissipation and a

‘reradiation damping’ term accounting for acoustic energy loss. However, Pialucha [105] demonstrated that damping also has an insignificant effect on reflection. An important study by Drinkwater *et al.* [106] in 1996 settled the question of the appropriateness of the ‘spring model’. They pointed out that the studies carried out in the previous years to investigate possible frequency dependence were using single frequency transducers – thus requiring multiple tests to examine different frequencies. Since ultrasonic reflection is sensitive to the precise contact alignment, performing repeated tests could easily lead to variations in the reported stiffness. Drinkwater *et al.* [106], however, were able to use broadband transducers and were thus able to address the question of the frequency dependence in a single test by calculating reflection coefficients and stiffnesses across the frequency range of the transducer. They found that the reflection coefficient was dependent on frequency in the same way as predicted by the ‘spring model’, but showed that contact stiffness was independent of the frequency – thereby verifying the quasi-static ‘spring model’.

There are, however, some limitations on the frequency range which will give acceptable results. A succinct discussion of this is given in Drinkwater *et al.* [106]. If the wavelength of the ultrasound is similar to the size of the interface gaps, then complex scattering will occur where resonances are established between gaps (Rose [107]). If the wavelength is considerably smaller than the gap sizes, the results will not be affected by the gaps; however, the frequencies required to achieve sufficiently small wavelengths will be of the order of GHz for most metals. At frequencies this high, the attenuation of the signal would be too great to measure meaningful results. If wavelengths significantly greater than the gap sizes are used, then the reflection and transmission of ultrasound will be unaffected by the size, shape and distribution of gaps, and attenuation of the signal will be low. Therefore measurements are generally taken in the low frequency/long

wavelength regime. Most work has been done on metals and the frequency range of the transducers generally fits inside the range 1–20 MHz [106, 108-113].

Ultrasonic transducers are always a piece of piezoelectric material whose dimensions oscillate as potential difference across the transducer is cycled. Evidently, the coupling between the transducer and the specimen is important. A number of techniques have been used: Drinkwater *et al.* [106] and Dwyer-Joyce *et al.* [109] used a water bath to transmit the ultrasound from the transducer to the specimen; Dwyer-Joyce and Gonzalez-Valadez [112] used water based gel and molasses for transmission of longitudinal and shear waves, respectively; Królikowski *et al.* [101], Królikowski *et al.* [102], and Królikowski and Szczeppek [100] used non-aqueous grease; and Kendall and Tabor [98], Gonzalez-Valadez and Dwyer-Joyce [114], and Gonzalez-Valadez *et al.* [113] used a suitable temperature stable adhesive.

Most researchers (Królikowski *et al.* [101]; Królikowski *et al.* [102]; Drinkwater *et al.* [106]; Dwyer-Joyce *et al.* [109]; Baltazar *et al.* [110]; Dwyer-Joyce and Gonzalez-Valadez [112]; Kim *et al.* [111]; Gonzalez-Valadez and Dwyer-Joyce [114]; Gonzalez-Valadez *et al.* [113]) have used the pulse-echo technique where a package of ultrasound waves is transmitted in emission mode, and sometime later, the reflection is received by the same transducer in pick-up mode. Others (Kendall and Tabor [98], and Królikowski and Szczeppek [100]) have used two separate transducers: one to emit the signal and another (on the other specimen) to receive the signal which has been transmitted through the interface. Either transmission, or reflection coefficient can be used to determine contact stiffness, and the choice of technique simply depends on user preference. Baltazar *et al.* [110] actually emitted ultrasound at oblique angles with respect to the contact surface using two transducers (one to emit and one to receive). They showed results that indicate that the contact stiffness derived from the reflections is independent

of the angle of incidence of the waves (at least for up to 40 degrees off the interface normal which is the maximum angle for which results were given).

Procedures for the acquisition and analysis of the ultrasonic data are well established at this stage. The procedure used by Drinkwater *et al.* [106], Dwyer-Joyce *et al.* [109], Dwyer-Joyce and Gonzalez-Valadez [112], Gonzalez-Valadez and Dwyer-Joyce [114], and Gonzalez-Valadez *et al.* [113] will be mentioned here as their method is followed later in this thesis. An ultrasonic pulsar-receiver (UPR) is used to generate the voltage pulses to actuate the transducers. The reflected signal, as a function of time, is then amplified, stored on a digital oscilloscope, and passed to a PC for signal processing. A fast Fourier transform (FFT) is then performed on the amplitude-time data to convert it to the frequency domain (amplitude versus frequency). A signal equivalent to the incident signal is achieved by measuring the reflected waveform when the specimens are out of contact. As this produces a material-air interface with a very large difference in acoustic impedance, almost the entire incident wave (approximately 100%) is reflected since air acts as a mirror to ultrasound in this case. The amplitude-frequency spectrum for this 'equivalent incident signal' is obtained in the same way as for the reflected signals by performing an FFT. The reflected spectrum is then divided by the 'equivalent incident' spectrum to give a reflection coefficient spectrum. This is then converted at each frequency point to stiffness using the spring model equation (Equation 6.19) and averaged to give the contact stiffness for a particular contact load.

The ultrasound method has more often been used for the calculation of normal contact stiffness (all of the experimental ultrasound papers discussed above used longitudinal transducers), but the approach is equally applicable to tangential stiffness as long as an appropriate ultrasonic shear transducer is used and the wave speed in shear is used in calculating acoustic impedances. The response of a rough contact to shear waves was

investigated successfully by Królikowski and Szczepek [108], Baltazar *et al.* [110], Dwyer-Joyce and Gonzalez-Valadez [112], and Gonzalez-Valadez *et al.* [113]. One of the major advantages of the ultrasound technique (which was referred to in the original paper by Kendall and Tabor [98]) is that it can be applied to such a wide range of materials (i.e. metals and non-metals, transparent materials and opaque materials).

2.4.4 Load-deflection versus ultrasound

There appears to be no available literature in which contact stiffness measurements derived from load-deflection data are compared to ultrasonically obtained stiffness values in the same test. It is possible only to perform a very rudimentary comparison of the results from the two techniques by comparing work by various researchers where the materials and contact configurations vary. For example, to compare digital image correlation based tangential contact stiffness results (load-deflection) to ultrasound based tangential contact stiffness results, the DIC results of Kartal *et al.* [94, 95] can be compared to the ultrasound results of Gonzalez-Valadez *et al.* [113] and Baltazar *et al.* [110]. Kartal *et al.* [94, 95] were using Ti-6Al-4V with flat-and-rounded pads in contact with a flat, Gonzalez-Valadez *et al.* [113] were using steel-on-steel with a cylindrical punch-on-flat configuration, and Baltazar *et al.* [110] were using aluminium-on-aluminium also with the cylindrical punch-on-flat arrangement. It is possible to compare results by these researchers at a nominal contact pressure of 70 MPa. At this pressure, Kartal *et al.* [94, 95] measured normalised stiffness values in the range 8–12 kN/mm³, Gonzalez-Valadez *et al.* [113] measured a range of 180–650 kN/mm³, and Baltazar *et al.* [110] noted a range of 200–700 kN/mm³. The ultrasound results are significantly greater; however, apart from the difference in material and contact configuration, there are some other differences: the results quoted from Kartal *et al.* [94, 95] are averages from in

excess of 5000 gross-slip fretting cycles at constant normal pressure, while Gonzalez-Valadez *et al.* [113] and Baltazar *et al.* [110] took their results in a non-fretting situation with zero bulk tangential load applied. Ultrasonic shear waves apply an Ångstrom scale tangential displacement to the interface which means that the results in Gonzalez-Valadez *et al.* [113] and Baltazar *et al.* [110] should be equivalent to the tangent stiffness of the load-deflection curve at the very onset of bulk tangential load application. Whereas, the stiffness values reported in Kartal *et al.* [94, 95] were secant stiffness measurements taken from a number of points on the load-deflection curve in a finite region after the onset of the partial-slip regime (which would likely give a lower stiffness). In addition, wear debris may have increased the contact compliance in Kartal *et al.* [94, 95]. These reasons may go some way towards explaining why the ultrasound result was much stiffer than the DIC result for the literature discussed in this section, but a comprehensive study comparing the two techniques is required.

A key difference between the contact stiffness results derived from tangent values of load-deflection results and those measured using ultrasound was hypothesised by Kim *et al.* [111]. These authors realised that an ultrasound wave actually imparts a small scale load-unload cycle upon the bulk static load. Therefore, if the load-deflection curve is elastic both methods should measure the same stiffness. However, if there is plasticity or some source of irreversibility, the ultrasound can be expected to measure the local unloading stiffness while the load-deflection technique will be measuring the tangent stiffness of the loading curve. Although Kim *et al.* [111] demonstrated the difference by comparing the contact of two elastic spheres (representing the ‘ultrasound unloading stiffness’) and two elastic-plastic spheres (representing the load-deflection tangent stiffness), no experimental evidence demonstrating the difference has so far been uncovered. This is redressed in Chapter 6 (Section 6.4.7).

2.4.5 Physics of rough interface contact stiffness

Having looked in detail at the developments that have occurred in two major methods of measuring rough surface contact stiffness (load-deflection, and ultrasound), we turn now to discussing the physical insights that have been achieved with these methods. Courtney-Pratt and Eisner [85] measured tangential load-deflection data in pre-sliding using the ball-on-flat arrangement (using various metals: platinum, steel, tin, and gold), but did not calculate values of tangential contact stiffness. A result for platinum is shown in Figure 2.8. They noted that the first tangential ‘load-up’ of the contact was plastic and nonlinear, and that any subsequent unloading and reloading occurred in almost elastic reversible hysteresis loops unless the previous maximum tangential load was exceeded whereupon the plastic behaviour would reinitiate. The contacts in Courtney-Pratt and Eisner [85] were already plastic under the normal load before any tangential load was applied, and hence they found that there was no finite range of values of the tangential force for which the tangential displacements were reversible during ‘first load-up’. They argued that many researchers who had observed reversibility had done so because the contact had already been subject to some tangential loading.

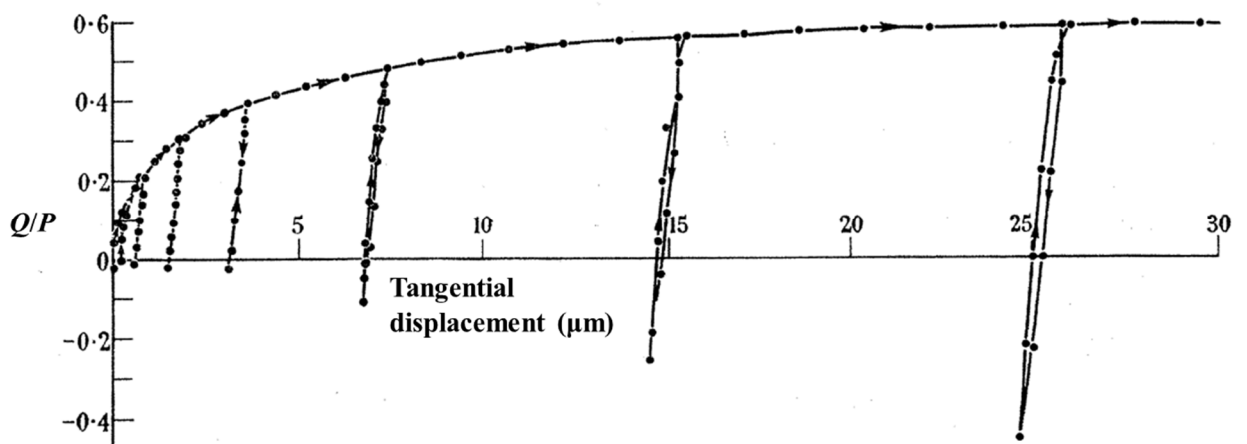


Figure 2.8: Tangential load-deflection plot for ‘first load-up’ of a polished platinum ball-on-flat contact from Courtney-Pratt and Eisner [85].

Berthoud and Baumberger [87], however, observed a linear region at the beginning of tangential load application where the tangential displacements *were* in fact reversible. Their result for contact between two flat blocks of poly methyl methacrylate (PMMA) is shown in Figure 2.9. The difference between this work and the work of Courtney-Pratt and Eisner [85] may be that Berthoud and Baumberger [87] were using low normal pressures and high displacement resolution in order to capture this reversible region (the reversible region in Figure 2.9 only extends to about $0.2\ \mu\text{m}$ of tangential displacement). It is likely that the magnitude of the normal pressure will have a major impact on how much of the tangential ‘first load-up’ of a rough contact is reversible.

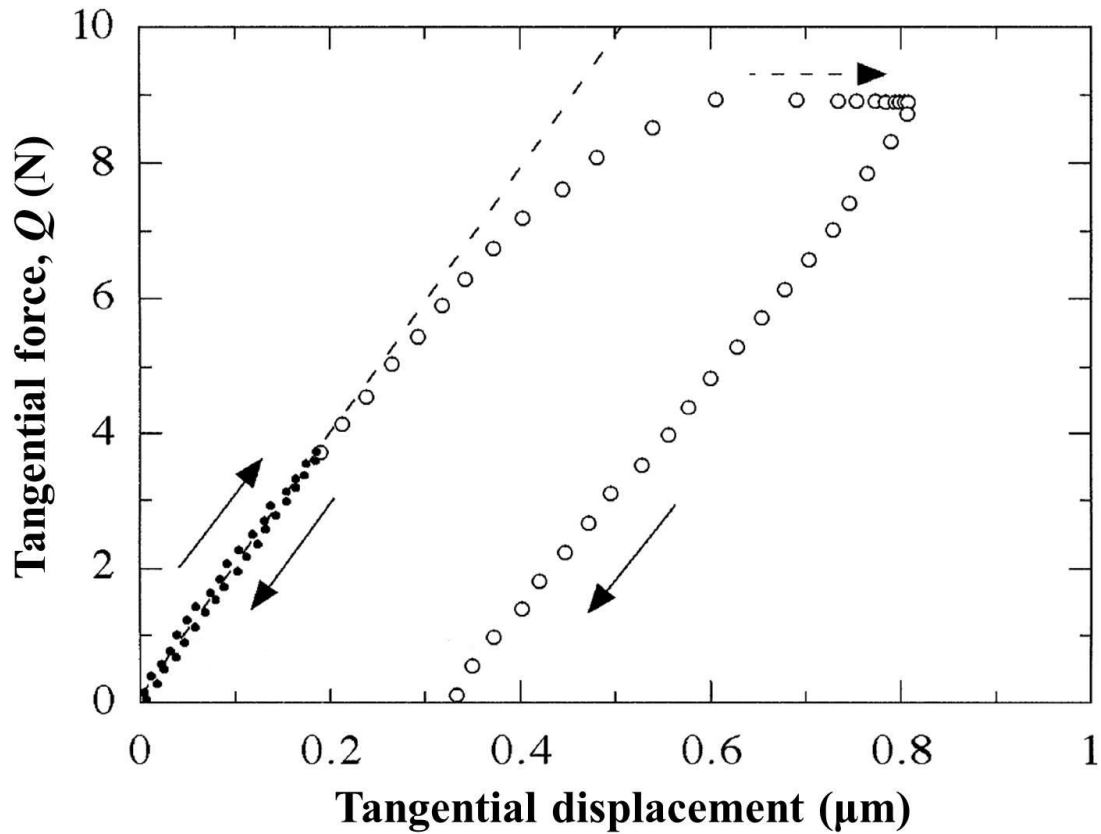


Figure 2.9: Load-deflection data for tangential loading and unloading of two flat PMMA blocks from Berthoud and Baumberger [87]. For a small enough tangential load, the loading-unloading curve is reversible (black dots).

Johnson [84] and O'Connor and Johnson [86] also measured tangential load-deflection data in pre-sliding (using a ball-on-flat steel-on-steel arrangement), but they were concerned with comparing their experimental data with the smooth contact prediction of Mindlin [115] (an elastic solution). They found that their tangential load-deflection curves gave good agreement with Mindlin's solution [115] for tangential loading of a sphere contacting a flat. These were the first investigations to suggest that Mindlin's prediction of the tangential load-deflection curve could be applied to contacts with real roughness. The suggestion inherent in these results is that the 'interface contact stiffness' arising from the roughness was so high that it had little impact on the response. However, some authors have found the opposite behaviour. Berthoud and Baumberger [87] using the flat-on-fat arrangement (with aluminium alloy AG5 contacts and PMMA contacts) found that the tangential load-deflection response was dominated by the 'interface stiffness' and that the bulk deflection contribution was negligible in comparison. Shi and Polycarpou [116] found (using ball-on-flat stainless steel contacts), for the case of normal contact stiffness, that the experimental values were considerably less stiff than the smooth contact Hertz prediction. Clearly this issue of how the stiffness due to the surface roughness compares with the stiffness arising from bulk deflections needs further consideration. A more detailed analysis of this question is given in Chapter 6 (Section 6.4.7).

The relationship between contact stiffness (normal and tangential) and normal loading of the contact has been the subject of much investigation (Kendall and Tabor [98]; Królikowski and Szczepek [100]; Drinkwater *et al.* [106], Berthoud and Baumberger [87]; Fujimoto *et al.* [88]; Dwyer-Joyce *et al.* [109]; Baltazar *et al.* [110]; Dwyer-Joyce and Gonzalez-Valadez [112]; Kim *et al.* [111]; Filippi *et al.* [93]; Shi and Polycarpou [116]; Gonzalez-Valadez *et al.* [113]; Kartal *et al.* [95]). Of these studies, Fujimoto *et*

al. [88], Filippi *et al.* [93], and Kartal *et al.* [95] used load-deflection methods, Shi and Polycarpou [116] used a contact resonance method while the others all used ultrasound. All of these researchers found that contact stiffness increases with increased normal contact pressure. This occurs due to the accompanying increase in the real area of contact. Most of the observed results (contact stiffness versus normal pressure) follow a concave up square root type increase with loading. Of particular interest is the observations of Drinkwater *et al.* [106], Dwyer-Joyce *et al.* [109], Dwyer-Joyce and Gonzalez-Valadez [112], and Gonzalez-Valadez *et al.* [113]. These authors performed a number of cycles of increasing and decreasing the normal loading while measuring the contact stiffness using ultrasound. It should be noted that they did not fully remove the load during unloading so as to ensure that the contact registration at the asperity level was not disturbed. A representative result from one of these papers (Gonzalez-Valadez *et al.* [113]) where a steel-on-steel cylindrical punch-on-flat arrangement was used is shown in Figure 2.10.

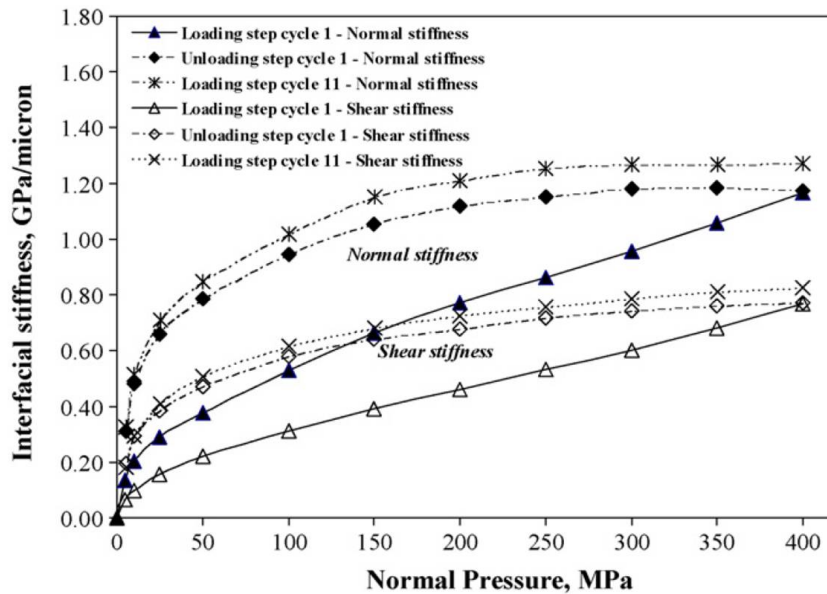


Figure 2.10: Variation of both normal and tangential (shear) contact stiffness with normal pressure measured by ultrasound for a steel-on-steel, cylindrical punch-on-flat arrangement. The first loading-unloading step, and the 11th loading step is shown. From Gonzalez-Valadez *et al.* [113].

The plot shows results for both normal and tangential (shear) stiffness for the ‘first load-up’, the ‘first unload’, and the ‘eleventh load-up’. It was found that the ‘first load-up’ of normal load on the contact produced a different curve than subsequent unloading and reloading. The first load-unload cycle was found to show a large amount of hysteresis with subsequent cycles showing relatively little hysteresis. This was attributed to the idea that most of the plastic deformation occurs during the first cycle. The first unloading curve follows a different curve to the first loading curve and shows higher stiffness values for a given normal pressure. This may be explained by presuming that the asperities have been flattened during first loading – hence giving a higher contact area. In subsequent loading and unloading cycles, the curve follows the form of the first unloading curve. However, the curve shifts upward slightly as further cycles are performed. For example, in Figure 2.10, the loading curve for cycle 11 follows the same path as the unloading curve from cycle 1, but has a somewhat greater magnitude. This indicates that some plasticity is still present and that full plastic shakedown has not occurred even by cycle 11. The authors suggest that this behaviour may be related to stress relaxation and creep (which was studied for rough interfaces by Gonzalez-Valadez and Dwyer-Joyce [114]). It should be noted that both normal and tangential contact stiffness behaved in a similar way apart from the magnitude difference whereby shear stiffness was about 0.6 times the normal stiffness for the result shown in Figure 2.10. The form of the stiffness-pressure curve in cycle 11 of Figure 2.10 is typical of the type of response commonly observed. Kartal *et al.* [95] and Filippi *et al.* [93] observed a similar type of curve with the load-deflection method (using DIC and laser Doppler vibrometry, respectively).

However, Berthoud and Baumberger [87] (using load-deflection) observed a linear variation of contact stiffness (tangential in this case) with normal load. Based on this

observation, they defined a ‘characteristic length’ as the reciprocal of the constant of proportionality between tangential contact stiffness and normal load. They found that this characteristic length was roughly material independent (they tested metal and polymer contacts), and was of the order of the scale of the roughness of the surfaces used. Thus, they saw this parameter as a ‘fingerprint’ of the ‘mesoscopic’ interface structure. The linearity in Berthoud and Baumberger’s result and the square-root type variation in other results are easily reconciled by noting that the normal pressures in Berthoud and Baumberger’s work were very small and hence the contact behaved in a linear elastic manner. The results of Gonzalez-Valadez *et al.* [113] show that as the contact pressure is increased, linearity is lost as the real contact area increases less and less with equivalent increments of normal load. Indeed, the Nowell model for tangential contact stiffness (outlined in [95]) predicts linearity with normal load, but this model is based on non-interacting elastic asperities: an assumption which is likely to break down at higher normal loads (the Nowell model applies Mindlin’s result [115] to the statistical rough interface analysis of Greenwood and Williamson [64] using an exponential distribution of asperity heights).

The effect of nominal contact area on contact stiffness has received little examination in the literature. This has important practical implications. For example, researchers (Petrov and Ewins [117], and Petrov [118]) who model frictional joints for the purposes of vibration analysis often assume that contact stiffness is proportional to the nominal contact area. The conditions under which this is true require investigation.

The effect of surface roughness on contact stiffness has been examined using ultrasound by both Kim *et al.* [111] and Gonzalez-Valadez *et al.* [113]. Both found that contact stiffness increases as the contacting surfaces are made smoother. This is reasonable since a smoother surface will afford a greater area of contact with less flexibility in the local

asperity contacts. Figure 2.11 shows the normal contact stiffness results obtained by Kim *et al.* [111] for three different surface roughness configurations: (a) both surfaces having an RMS roughness of 0.25 μm (smooth-smooth), (b) one surface having a roughness of 0.25 μm and the other surface having 2.4 μm RMS roughness, and (c) each surface having an RMS roughness of 2.4 μm (rough-rough). They used a punch-on-flat aluminium alloy (6061-T6) contact. It is clear that the smoother the contact interface, the higher was the contact stiffness. These authors also developed a micro-mechanical elasto-plastic model of a multi-contact interface and found good agreement with the experiments (Figure 2.11).

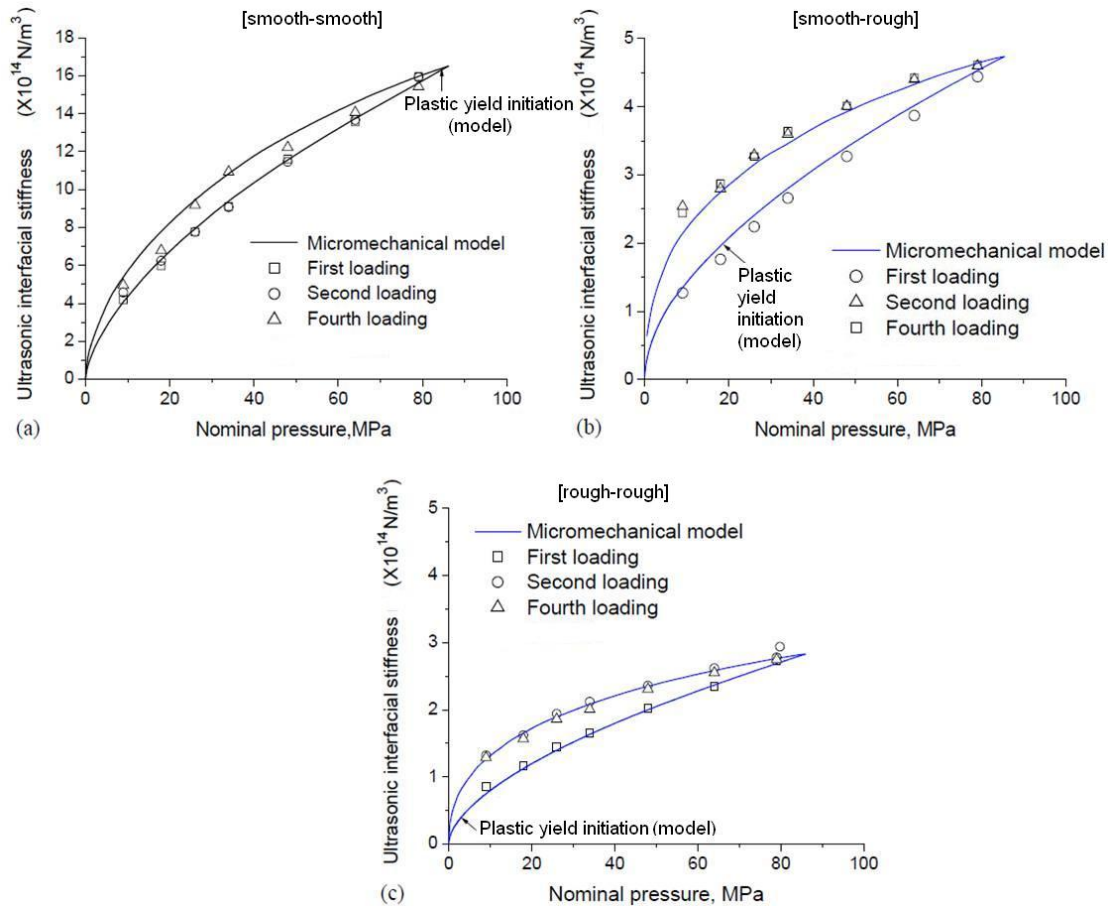


Figure 2.11: Normal contact stiffness (measured by ultrasound) versus normal pressure for a punch-on-flat aluminium alloy (6061-T6) contact: (a) each surface had RMS roughness of 0.25 μm (smooth-smooth), (b) one surface had RMS roughness of 0.25 μm and the other had 2.4 μm (smooth-rough), and (c) both surfaces had RMS roughness of 2.4 μm (rough-rough). Also shown is the prediction of a micromechanical elasto-plastic multi-contact model. From Kim *et al.* [111].

The point of plastic yield initiation predicted by their multi-contact model is indicated.

A number of researchers have also studied contact stiffness during either partial-slip or gross-slip fretting (Filippi *et al.* [93]; Kartal *et al.* [94, 95]; Eriten *et al.* [91, 92]; Lavella and Botto [39]; Schwingshackl *et al.* [96]; de Crevoisier [97]). All of these studies where investigations of tangential contact stiffness performed using the load deflection method. Kartal *et al.* [94, 95] measured tangential contact stiffness using DIC during gross-slip fretting wear tests on flat-and-rounded pads pressed against flat specimens using Ti-6Al-4V. Kartal *et al.* [95] found that at the lower normal pressures tested (35 and 70 MPa), the tangential contact stiffness was roughly constant with the number of fretting cycles, while for the higher contact pressures (105 and 140 MPa), the stiffness increased during the tests (see Figure 2.12). They suggested that more severe wear causing greater conformity between the surfaces may be the cause of this behaviour.

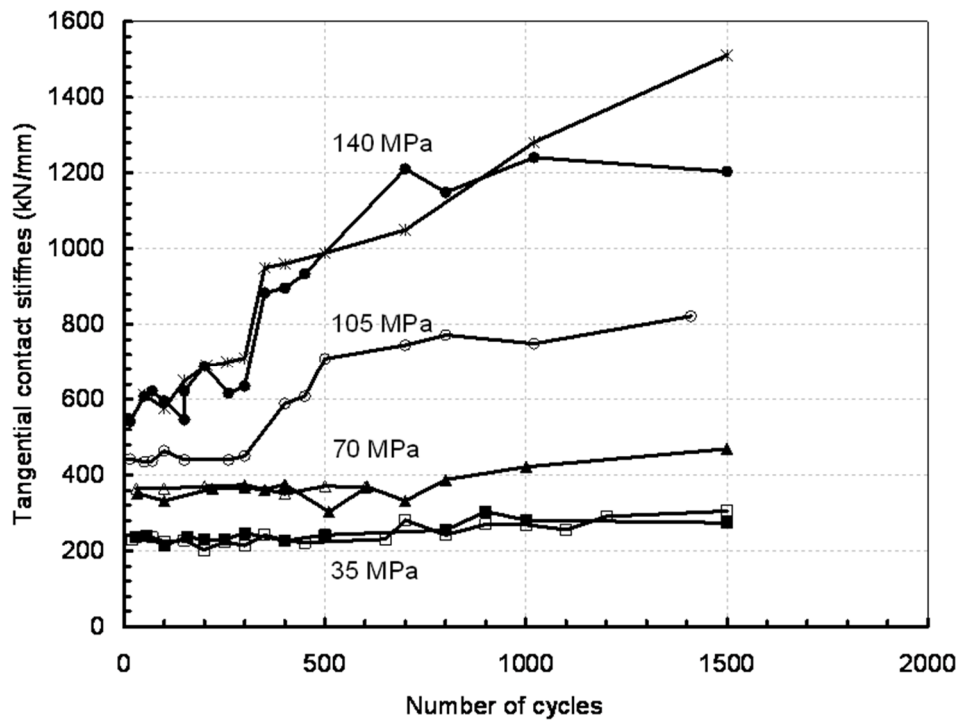


Figure 2.12: Tangential contact stiffness (measured by DIC) versus number of test cycles for gross-slip fretting of Ti-6Al-4V using flat-and-rounded pads pressed against a flat sided specimen. Results are shown for four different values of the nominal normal pressure. From Kartal *et al.* [95].

Contact stiffness results associated with various distances of the DIC measurement points from the interface are reported in Kartal *et al.* [94, 95] (see Figure 2.13). The stiffness values were found to increase as the measurement points approached the interface (and also with contact pressure in the manner discussed above). For contact stiffness results derived from relative DIC displacements between two given points on either side of the interface, it is difficult to decide how much of the relative displacement was due to bulk material deformation between measurement points and how much was due to ‘rough interface’ compliance. Although plastic shakedown might be assumed to occur after repeated load cycles, there is also a complicating factor for the fretting case in the form of the effect of wear debris upon the interface compliance. In any case, it is necessary to develop a means of deciding how much of the relative displacement originates from the interface and how much is attributable to the bulk material. If this can be done then the true ‘interface stiffness’ can be isolated from the experimental results. This topic is investigated in Chapter 6 (Section 6.3).

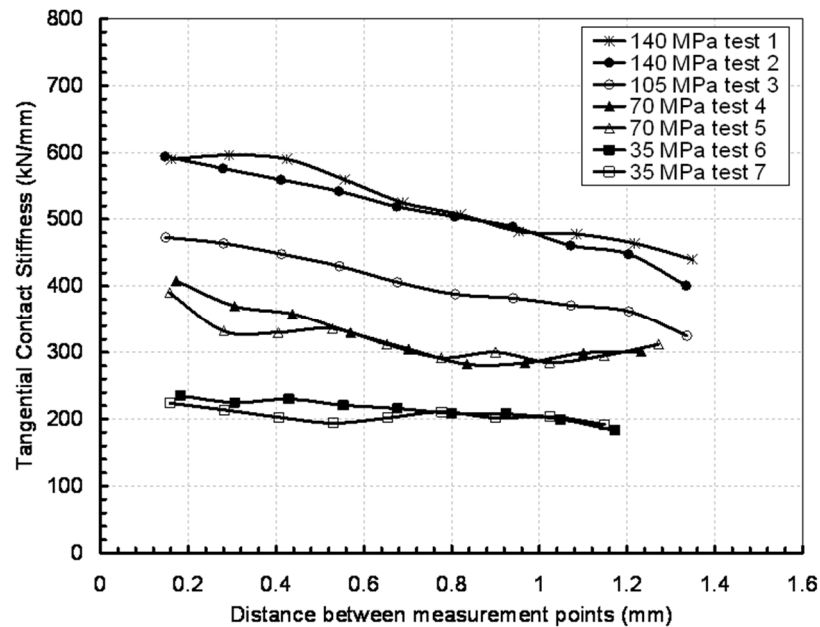


Figure 2.13: Tangential contact stiffness (measured by DIC) versus distance between measurement points (i.e. twice the distance to the interface) for a number of gross-slip fretting tests on Ti-6Al-4V at various normal pressures (taken at cycle 100). From Kartal *et al.* [95].

2.5 Summary and critical analysis of the literature

2.5.1 Introduction

The literature review has focused on particular developments in three distinct but closely related areas: gross-slip fretting, friction modelling, and the contact stiffness of rough interfaces. A summary and critical analysis of the work which has been discussed is given next for each of the three topics with particular emphasis on gaps in the literature which this thesis aims to address.

2.5.2 Gross-slip fretting

The wear mechanism was seen to involve the formation of adhesion junctions and the subsequent detachment of wear particles by such processes as plastic deformation and fracture. It was shown that the fretting regime is the dominant factor in determining the type of fretting damage which occurs: partial slip promotes fretting fatigue, while gross slip promotes fretting wear. The importance of entrapped wear debris in the contact interface was outlined by reference to the ‘third body approach’ to wear which sees the flow of third bodies out of the contact interface as a major factor in regulating wear particle generation. Ways of thinking about the modelling of wear by expressing the wear volume in terms of the product of accumulated sliding distance and normal load – as in the case of the Archard model, and in terms of the total dissipated frictional energy – as in the Fouvry model – were outlined. Recent methods of modelling wear scar evolution using finite elements were also discussed.

A particular feature which was noted in very many fretting wear tests in the literature was the significant variation in the friction force which was seen to occur during the slip phase of individual cycles. Various possible mechanisms were discussed to explain this

behaviour: some authors thought that interaction at wear-scar ends may be responsible for the behaviour, while others wondered whether a similar wear-scar interaction might be occurring locally throughout the contact. A velocity effect was also put forward as an explanation. It was clear from the literature that, although many researchers had commented on the phenomenon, no experimental investigation had been done to try to comprehensively isolate its cause. Therefore, there is a need for a detailed experimental study to address this shortfall.

Various studies in the literature which investigated surface properties during fretting were discussed. Many researchers have used traditional 2D surface topography parameters and some have more recently used 3D surface parameters, but the full range of 3D surface topography parameters have not been utilised. For example, the areal parameters which characterise the spatial (or lateral) nature of surfaces (such as auto-correlation and cross-correlation) appear to have received little use. There is clearly more work which can be done to fully realise the application of the full range of 3D surface topography parameters to fretting wear issues.

Researchers performing cylinder-on-flat fretting tests on Ti-6Al-4V noted the existence of a W-shaped wear scar which was contrary to the U-shaped wear scar predicted by finite element modelling using an Archard approach. Two different explanations were put forward based on slight pad rotation causing a 'dig-in' to produce the dips in the 'W' shape, and on the accumulated plastic strain distribution across the contact having a similar 'W' shape. A useful step to take may be to see whether this type of wear scar occurs with a different contact arrangement such as the flat punch with rounded edges fretting against a flat, and with a different material tested under the same conditions.

2.5.3 Friction modelling

It was shown that real area of contact is nearly always directly proportional to normal load for rough engineering interfaces in contact. How this result follows directly from the notion of plastically deforming asperities was outlined. It was also shown that proportionality could also be achieved with an elastically deforming multi-contact interface depending on the surface statistics. Indeed many common engineering surfaces have distributions which give an almost linearly proportional relationship (such as Gaussian). Combining the proportionality of real contact area with the idea that a constant shear strength is required to slide regions of real contact (i.e. the adhesion theory) then leads elegantly to the Amontons-Coulomb laws.

Various attempts to model the frictional process by the sliding interaction of spherical or cylindrical asperities using finite element analysis were then reviewed. These studies illustrated that elastically deforming asperities predicted a zero friction coefficient, but asperities undergoing some plastic deformation predicted a net non-zero friction coefficient. However, it was found that these studies had always predicted friction coefficients for metals which were considerably less than those observed by experiment in ambient conditions. In fact the friction coefficients predicted by these modelling attempts were more typical of lubricated contact. There are, however, a number of deficiencies with these modelling attempts. None of these studies have incorporated an interface shear strength as envisaged by Bowden and Tabor's adhesion theory. The asperity overlaps for which models have been run have also been small, and this is probably due to convergence difficulties with the implicit finite element solvers used. Incorporating the adhesion theory in models of sliding asperity interaction, and allowing for greater asperity overlaps would seem to be a way of improving these modelling

approaches. Neither has asperity damage been accounted for in the literature, and with the inclusion of tangential adhesion and the modelling of greater overlaps, it would seem that damage and failure would also need to be addressed.

2.5.4 Contact stiffness of rough interfaces

On contact stiffness, two methods of experimental measurement have been discussed: the load-deflection method, and the ultrasound method. It was suggested that techniques which measure relative displacement close to the interface are preferable as they are more likely to eliminate bulk deformation and give a result which is closer to the actual rough interface stiffness. The load-deflection method has been used successfully by some researchers to investigate contact behaviour in the pre-sliding regime – though not all of these studies have calculated contact stiffness. First tangential ‘load-up’ is usually plastic and irreversible, but under low normal pressures and using high resolution measurements, a linear reversible region has been noted. Further load-unload cycles were found to be largely reversible. Two load-deflection techniques which have successfully measured displacements locally are the laser Doppler vibrometry technique and the digital image correlation (DIC) technique. DIC has been employed very recently to measure tangential contact stiffness during fretting but does not appear to have been used in the measurement of normal contact stiffness or in a non-fretting environment.

Developments in the measurement of contact stiffness using ultrasound have also been discussed. The effect of cycling the normal pressure and varying the surface roughness has been studied with the method. Normal and shear stiffness have been found to increase with normal pressure and decrease as surface roughness is increased. Interestingly, no ultrasound studies of contact stiffness in the pre-sliding regime appear to be available. One of the major studies which is absent from the literature is a

comprehensive comparison of the load-deflection method with the ultrasound method. This would require using both methods to simultaneously take measurements during the same test. The techniques are very different in that ultrasound takes its measurement form Ångstrom scale displacements; whereas, load-deflection methods rely on the slope of the load-deflection plot over a range of up to a few microns. Hence, gauging the level of agreement between the two methods is of great interest.

A difference between the two methods which was hypothesised by Kim *et al.* [111] is that ultrasound actually measures ‘local elastic unloading stiffness’ while load-deflection methods measure the ‘tangent stiffness’ of the static loading curve. No experiments explicitly showing this difference appear to be available as tests where both ultrasound and load-deflection methods are used simultaneously would again be needed.

A few researches have recently measured tangential contact stiffness during gross-slip fretting using the DIC technique. Contact stiffness was found to increase: as the number of cycles was increased for high pressures (but not for lower pressures), as distance between measurement points was reduced, and as normal pressure was increased. Kartal *et al.* [94, 95] have obtained stiffness values for a range of distances from the interface. There is a need to interrogate these results to determine how much of these stiffness values are governed by rough interface movement, and how much is derived from displacements due to bulk material deflection between the measurement points. To achieve this, a technique which can isolate the ‘rough surface contact stiffness’ from the experimental results would be required. Finally (and this is related to the interface compliance to bulk compliance ratio), the effect of nominal contact area on contact stiffness has received little attention in the tribology literature, and since this is important for the vibration modelling community, some investigation is required.

Chapter 3

Preliminary characterisation work on friction and wear behaviour in gross-slip fretting

3.1 Introduction

In this chapter a preliminary experimental study is carried out to investigate frictional behaviour in gross-slip fretting and to help develop a foundation for further research. A basis is laid for using 3D surface topography scans to enhance understanding of frictional behaviour by imaging both pre and post-test surfaces, and appropriate methodology and settings are established. Surfaces are imaged in 3D at high resolution using advanced focus variation microscopy. Area based roughness parameters are used rather than the traditional line-profile based parameters in order to include more surface information. Fretting tests on two separate materials (Ti-6Al-4V and Udimet 720) with surfaces exhibiting three grades of roughness are carried out to assess their effect on frictional

behaviour. Frictional hysteresis loops (tangential force versus relative displacement) are measured and examined. In particular, how material choice and roughness affect the measured loops is investigated. Whether there is an increase in friction force during the sliding stroke as observed by some authors (see Section 2.2.3) in one or either material is of particular interest. Post-test wear scars are investigated both qualitatively and quantitatively to better understand the frictional mechanisms obtaining. An attempt to isolate tangential contact stiffness using remote measurements of relative tangential displacement measured using a linear variable differential transformer (LVDT) is made by comparing results from the frictional tests with equivalent results from a monolithic test piece containing no frictional contact. A discussion of the limitations of this approach and the requirements for accurate measurement of tangential contact stiffness is then given. Before describing the tests, the techniques and parameters required for 3D surface characterisation are outlined.

3.2 Characterisation of surface topography

3.2.1 Focus variation based 3D surface profilometry

For characterisation of post-test and pre-test surfaces, the Alicona InfiniteFocus profilometer (Figure 3.1) was used throughout this thesis.

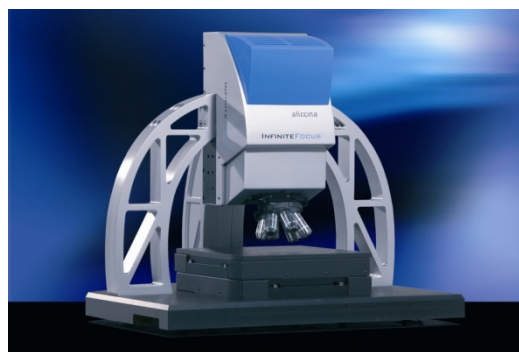


Figure 3.1: Alicona InfiniteFocus 3D surface profilometer [119].

This is a focus variation microscope which facilitates high resolution 3D surface scans. These allowed calculation of area based surface topography parameters which contain more surface information than parameters based on a simple traditional line profile. The technique combines the high resolution possible with small depth of focus optical microscopy with vertical scanning to acquire topographical and colour information from a surface. The principle of operation of the technique is shown schematically in Figure 3.2.

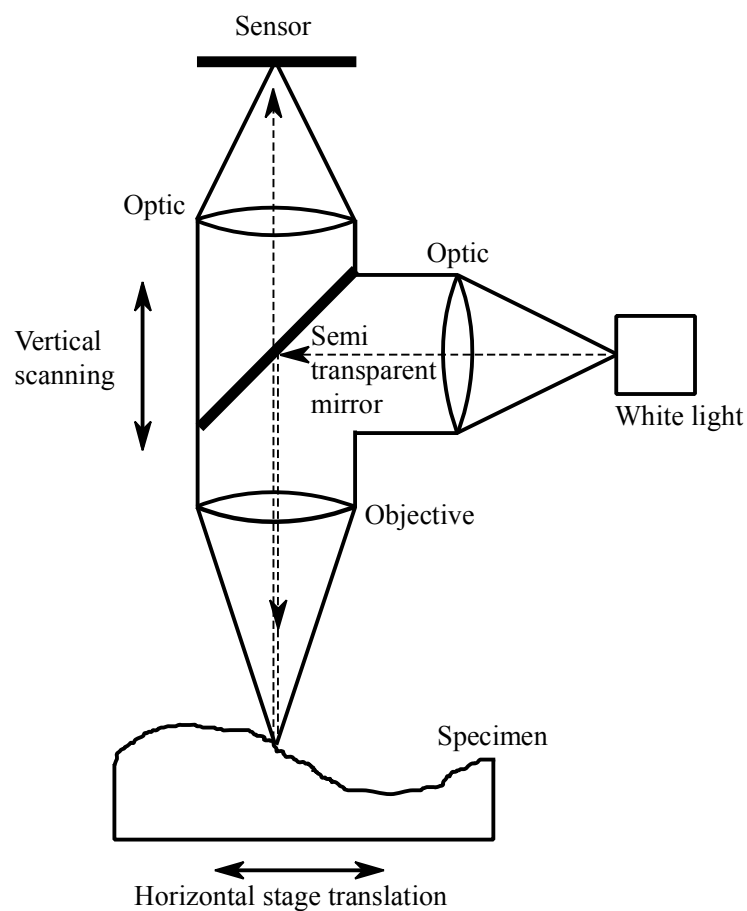


Figure 3.2: Principle of operation of focus variation microscopy.

Coaxial white light is emitted by a light source and passed through a beam splitting mirror to an objective which focuses the beam on the specimen surface. Reflected light from the surface is then combined by the optics, passed through the beam splitter, and

projected onto a digital light-sensitive sensor. The depth of field of the optics is small, and so, in order to obtain the full surface profile, the machine head containing the optics is moved in the vertical direction while images are continuously captured. Special algorithms are used to determine where the surface points lie (i.e. the optimum focus) at each level of vertical scanning. This is done by maximising the variation of image sharpness since sharpness varies most when an image is almost in focus.

The horizontal and vertical resolution of the resulting dataset is determined by the objective selected by the user. The InfiniteFocus provides a choice of selectable objectives ($\times 2.5$, $\times 5$, $\times 10$, $\times 20$, $\times 50$, and $\times 100$) contained in a six-piece nose-piece. Again owing to the optics, the field of view of even a coarse objective will be relatively small. In order to allow larger areas to be scanned, the work-holding stage is motorised in the horizontal x and y directions: as a result, a series of fields of view can be stitched together to form an image field which can allow an area of tens of millimetres to be scanned. For surfaces which are difficult to image, features such as a ring-light or a polariser can be enabled. The ring-light provides more diffuse light to the surface. The polariser filters direct reflections and passes only diffuse reflection to the sensor causing the resulting image brightness to depend on surface texture only rather than surface tilt. This can thus be useful when imaging uneven surfaces with various flats and flanks which result in a large variation of brightness: since the polariser evens out the brightness.

Vertical and horizontal resolution, vary from 2300 nm and 3520 nm, respectively, with the $\times 2.5$ objective to 10 nm and 90 nm, respectively, for the $\times 100$ objective. Using a very high resolution objective reduces depth of focus and field of view (field of view is $5716\text{ }\mu\text{m} \times 4351\text{ }\mu\text{m}$ for the $\times 2.5$ objective, but only $143\text{ }\mu\text{m} \times 109\text{ }\mu\text{m}$ for the $\times 100$ objective): this can result in prohibitive scan times depending on the size of the area

required to be scanned. Therefore, the selection of an appropriate objective will depend on the lowest allowable resolution, and the minimum area required to be imaged.

3.2.2 Areal surface topography parameters

In order to more fully investigate frictional behaviour, parameters for surface characterisation are required. Parameters derived from line profiles measured using stylus instruments have long been used to characterise rough surfaces. More information, however, can be obtained from 3D area scans. The recent standard ISO 25178-2:2012 [55] outlines, for the first time, the relevant measurement techniques and parameters for area based topography characterisation. Area based parameters are generally denoted by the letter S rather than R which is traditionally used for 2D measurements. Areal parameters are used throughout this thesis. Most, but not all, of the parameters used are covered in the ISO standard: though each is independently explained below. Roughness parameters are examined before and after certain tests carried out in this thesis to attempt to gain more information about the frictional behaviour at the contact. The Alicona IFM G4 software (version 3.5) is used to calculate all the parameters outlined in this section. Parameters describing the amplitude and height distribution properties of surfaces are outlined first. All parameters are determined from an area of $M \times N$ discrete surface points $z(x_j, y_k)$ and defined in a coordinate system where $z = 0$ is the surface mean plane.

3.2.2.1 Amplitude and height distribution parameters

1. S_a – Surface roughness

This is the arithmetic average of the absolute value of surface heights:

$$S_a = \frac{1}{MN} \sum_{k=1}^N \sum_{j=1}^M |z(x_j, y_k)| \quad (3.1)$$

2. S_q – Root mean square (RMS) roughness

This is the root mean square of surface heights:

$$S_q = \sqrt{\frac{1}{MN} \sum_{k=1}^N \sum_{j=1}^M z^2(x_j, y_k)} \quad (3.2)$$

3. S_p – Absolute value of the maximum peak height of selected area

4. S_v – Absolute value of the maximum valley depth of selected area

5. S_z – Maximum surface thickness of selected area (i.e. flatness)

$$S_z = S_p + S_v \quad (3.3)$$

6. S_{10z} – Ten-point thickness of a surface

This is the average of the five greatest surface thicknesses. It is calculated based on the five greatest peak and valley heights:

$$S_{10z} = \frac{\sum_{i=1}^{i=5} S_{p,i} + \sum_{i=1}^{i=5} S_{v,i}}{5}, \quad (3.4)$$

where $S_{p,i}$ and $S_{v,i}$ ($i = 1, 2, \dots, 5$) are greatest peak heights and valleys depths, respectively. This parameter is more robust than S_z .

7. S_{sk} – Skewness

Skewness is the third order moment of the surface height distribution:

$$S_{sk} = \frac{1}{S_q^3} \frac{1}{MN} \sum_{k=1}^N \sum_{j=1}^M z^3(x_j, y_k) \quad (3.5)$$

It is a measure of the asymmetry of the surface heights about the mean plane. A symmetrical Gaussian distribution has zero skewness. If the points furthest from the mean plane line predominantly above the mean, a positive skewness will result. If they lie below, skewness will be negative. Essentially, it tells us whether the surface has spikes protruding out of, or into, the surface.

8. S_{ku} – Kurtosis

Kurtosis is the fourth moment of the surface height distribution:

$$S_{ku} = \frac{1}{S_q^4} \frac{1}{MN} \sum_{k=1}^N \sum_{j=1}^M z^4(x_j, y_k) \quad (3.6)$$

It characterises the spread of the surface height distribution. A Gaussian surface will have a kurtosis value of 3, a centrally distributed surface will have a value greater than 3, and a wide distribution of surface heights will give a value smaller than 3.

9. S_k , S_{pk} , S_{vk} – Bearing area curve parameters

These parameters are calculated graphically from the bearing area curve for a 3D scan area (the bearing area curve was first introduced by Abbot and Firestone [120]). Mathematically, the bearing area curve is the cumulative probability density function of the surface heights. The method of calculation of the parameters is shown in Figure 3.3a: they are determined in the same way as in ISO 13565-2:1996 [121] for a 2D line profile.

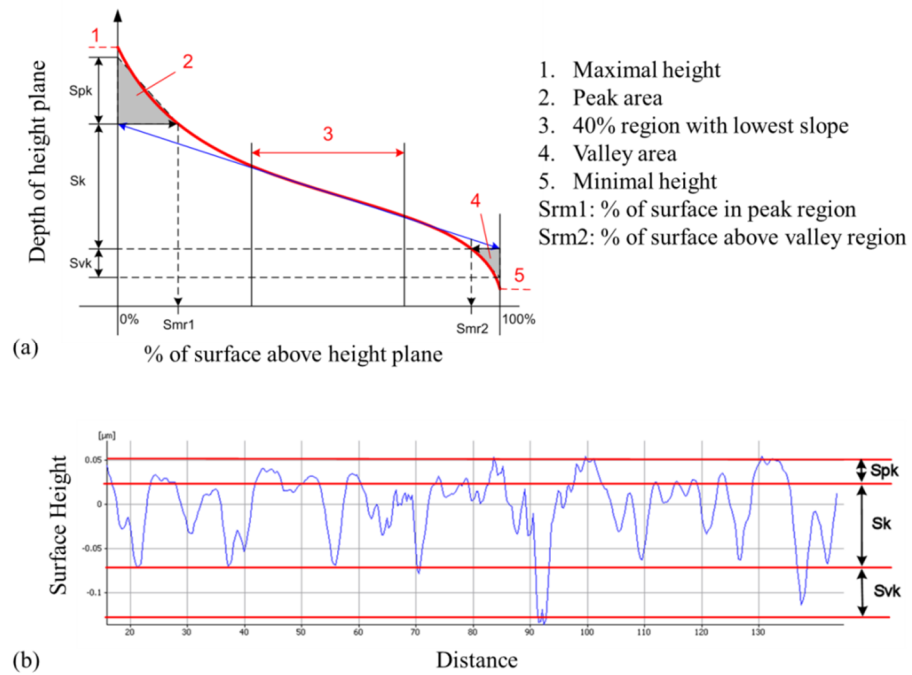


Figure 3.3: Definition of bearing area curve parameters: (a) Graphical construction for obtaining bearing area curve parameters, and (b) Illustration of the meaning of the parameters for a sample surface profile. After [119].

A least squares line is fitted to the 40% region of the curve with the lowest slope – marked (3). Two horizontal lines are drawn where this line crosses the 0 and 100% axes. The two points where these two lines intersect the bearing area curve are marked, and in the case of the upper horizontal line, a triangle (marked (2)) equal in area to the portion of the curve above the horizontal line (the peak area) is constructed (a triangle is constructed in a similar way at the lower horizontal line for the valley area – marked (4)). S_k is core roughness depth, S_{pk} is reduced peak height, and S_{vk} is reduced valley height. S_k is an alternative roughness measure, while S_{pk} and S_{vk} provide a more robust measure of peak and valley height than S_p and S_v , respectively as shown in Figure 3.3b.

3.2.2.2 Spatial parameters

The above parameters give no indication about the spatial distribution of surface features, their directional properties, or their size. The following three parameters are used to address this:

10. Sal – Auto-correlation length

The definition of the autocorrelation length depends on the normalised autocorrelation function (ACF). The ACF is the correlation between a surface and itself as displaced by Δx , and Δy :

$$ACF(\Delta x, \Delta y) = \frac{1}{S_q^2} \frac{1}{MN} \sum_{k=1}^N \sum_{j=1}^M z(x_j, y_k) z(x_j + \Delta x, y_k + \Delta y) \quad (3.7)$$

When the surfaces overlap exactly, the correlation is unity. However, as Δx and Δy are increased, the correlation reduces until the surfaces are fully uncorrelated. The auto-correlation length, Sal , is defined as the minimum horizontal distance (i.e. $\min\sqrt{\Delta x^2 + \Delta y^2}$) for which the ACF has decayed to an agreed value (0.2 is the usual standard [122]). It is a measure of how far one has to move laterally on a surface from some original position in order to find a surface which differs statistically from the original. Hence, it gives a measure of the dominant feature size in the case of surfaces exhibiting distinct features. A large Sal value indicates a surface dominated by low frequencies and high wavelengths (i.e. larger features); a low value suggests high frequencies and short wavelengths (i.e. small features).

11. Str – Texture aspect ratio

Ratio of the minimum horizontal distance for which the ACF has decayed to 0.2 to the maximum for which it has decayed to 0.2.

$$Str = \frac{\min\sqrt{\Delta x^2 + \Delta y^2}}{\max\sqrt{\Delta x^2 + \Delta y^2}} \quad (3.8)$$

Str can take on values between 0 and 1. Larger values ($Str > 0.5$) indicate uniform texture; whereas, smaller values ($Str < 0.3$) suggest strong directional structure.

12. Stdi – Texture direction index

This is the average value of the angular power spectrum divided by the value for the dominating direction. Smaller values indicate stronger directional structure.

3.2.3 Verification of measurements and determination of settings

Before using the Alicona InfiniteFocus for measurement of surface topography, a comparison of Alicona results with those taken from a different surface profilometer with a different operational principle was undertaken. The Wyco profilometer manufactured by Veeco (a 3D profilometer based on the principle of interferometry) housed at Imperial College London¹ was used for the comparison. A flat ground surface was used. A magnification of $\times 10$ was used with both profilometers. The vertical and horizontal resolutions were 3 nm and 898.3 nm, respectively for the Veeco Profilometer, and 100 nm and 879.8 nm for the Alicona. Figure 3.4 compares primary roughness parameters taken from exactly the same $1 \times 1 \text{ mm}^2$ scan area as imaged by the two different machines and calculated using the accompanying software from each manufacturer.

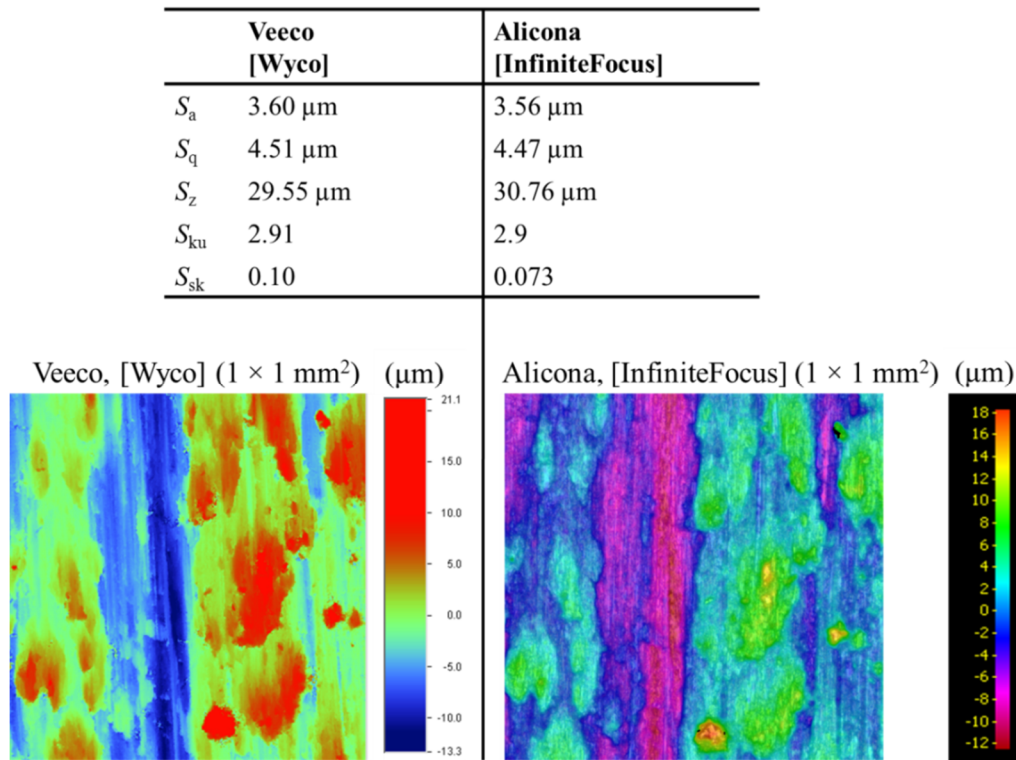


Figure 3.4: Comparison of surface height profiles and primary areal roughness parameters measured on the same $1 \times 1 \text{ mm}^2$ portion of sample coarse ground surface using the Veeco (interferometry based) and Alicona (focus variation based) 3D profilometers.

¹ Miss Daniela Proprentner is acknowledged for collaborating in facilitating the Veeco scan.

Good agreement can be seen between the two instruments: therefore, allowing confidence in results derived using the Alicona InfiniteFocus. Before using the InfiniteFocus on test samples, however, it was first necessary to decide on the measurement settings to be used. Settings discussed here apply to pre-test measurement of ground surfaces. Scan area, and horizontal and vertical resolution are among the most critical settings. Figure 3.5 shows the result of a test carried out on a ground surface to determine which scan area size and horizontal resolution to use. Average roughness S_a was calculated for a range of square scan areas, first with a horizontal resolution of 440 nm ($\times 20$ objective), and then with a horizontal resolution of 880 nm ($\times 10$ objective).

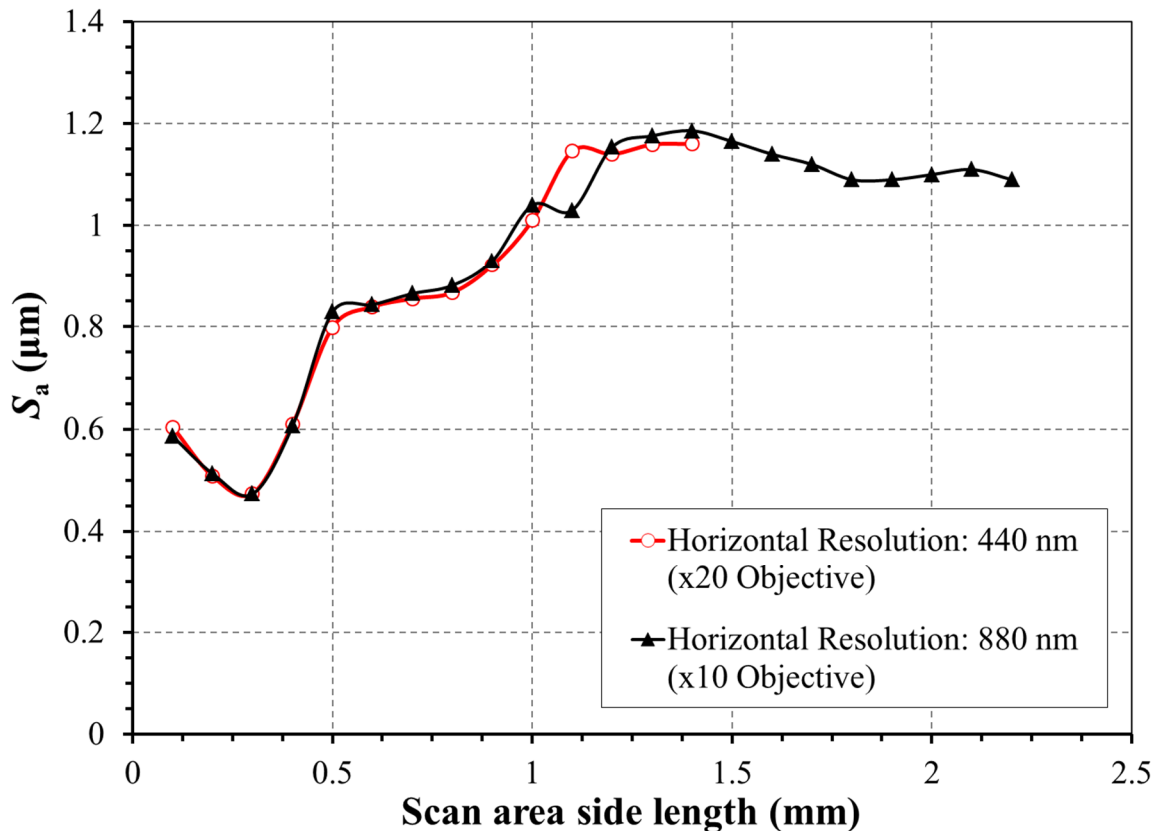


Figure 3.5: Effect of scan area and horizontal resolution (objective) on the primary average areal roughness S_a of a sample ground surface.

The results suggest that a scan area of at least about $1.2 \times 1.2 \text{ mm}^2$ is required to give representative results. It is also clear that there is little difference in results for the two horizontal resolutions (i.e. the two objectives). In light of this test, a scan area of $2 \times 2 \text{ mm}^2$ and a horizontal resolution of 880 nm ($\times 10$ objective) was agreed upon for measurement of pre-test roughness. The highest vertical resolution possible with the $\times 10$ lens (100 nm) was used. Following the advice of Blunt [123], primary (i.e. unfiltered) roughness values are reported throughout this thesis as filtering for waviness can cause loss of important information when considering the ‘before’ and ‘after’ effects of surface wear. This is permissible since all of the surfaces scanned are nominally flat.

3.3 Preliminary reciprocating sliding tests

3.3.1 Outline of preliminary experiments

The aim of the work in this section is to make comprehensive experimental measurements of macroscopic load-displacement (hysteresis) loops during gross-slip fretting for two material pairs with three well-characterised surface topographies (each with different surface roughness). The materials and scope of the investigation have been chosen to be representative of the blade-roots of gas turbines since this is a contact type which is expected to be one of the first areas to benefit from improved understanding. Surfaces were first comprehensively characterised in terms of their surface topography, and hysteresis loops were then measured during gross-slip fretting tests. The loops were used to investigate both friction and contact stiffness. Following the tests, surfaces were again scanned and compared to the pre-test surfaces in order to learn more about the frictional processes obtaining at the contact. Work was concentrated on the ‘steady-state’ regime (approximately 100 to 10,000 cycles). By concentrating efforts in this regime, it

was hoped that the complications of initial surface contamination in early cycles and the more severe gross wear in later cycles (i.e. wear particle generation and third body lubrication) could be avoided – thereby making the measurements more repeatable and suitable for comparison to tests carried out at other labs.

3.3.2 Materials and specimens

Two materials were chosen for the experiments: the titanium alloy Ti-6Al-4V and the nickel alloy Udimet 720. These were chosen since they are the two materials often used to form fan, compressor, and turbine blades in gas turbine engines where contact between the blade-root and the dovetail slot in the disc will be a primary area of application for increased knowledge of friction and contact stiffness. Ti-6Al-4V is used for fan and compressor blades because of its good strength-to-weight ratio, and Udimet 720 is used in turbine blades owing to its appealing strength properties at high temperature. The mechanical and chemical properties of the two materials are given in Table 3.1.

Table 3.1: Mechanical and chemical properties of Ti-6Al-4V and Udimet 720.

Material	Young's modulus (GPa)	Poisson's ratio	Yield stress (0.2% offset) (MPa)	Chemical Composition (%)
Ti-6Al-4V	115	0.31	999	Al(5.5–6.75), V(3.5–4.5), N(0–0.05), C(0–0.08), O(0–0.2), Fe(0–0.4), H(0–0.02) Ti(Balance)
Udimet 720	219	0.33	881	Cr(15.5–16.5), Co(14.0–15.5), Mo(2.8–3.3), W(1.0–1.5), Ti(4.8–5.3), Al(2.3–2.8), C(0.01– 0.02), Zr(0.03–0.05), B(0.01–0.02), Ni(Balance)

The material was supplied by project collaborators Rolls-Royce plc. The test geometry involved flat pads with rounded corners perpendicular to the sliding direction pressed against either side of a prismatic specimen of square cross section. The geometry of the contact arrangement is shown in Figure 3.6. This setup is intended to be representative of a slice of a blade-root dovetail joint where the rounded corners give a similar pressure distribution as that which would obtain at a real blade-root (see Figure 1.1). Pads are held against the specimen at a constant normal load P . The specimen is then given a fixed applied sinusoidal displacement oscillation to induce reciprocating sliding. This gives rise to a friction force Q at each of the two pad-specimen contacts, which (assuming a symmetrical response) produces a total tangential force $2Q$ at the top of the specimen where it is measured by a load cell.

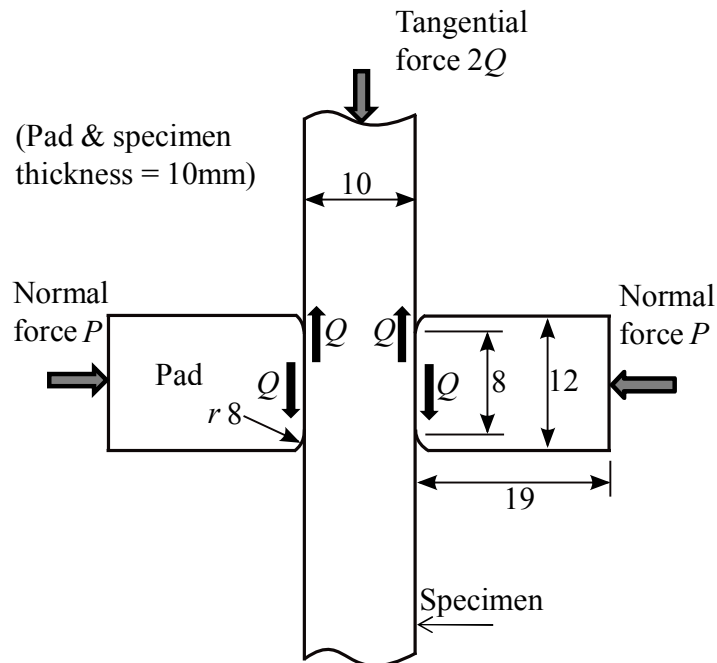


Figure 3.6: Schematic diagram of pad-specimen geometry.

A photograph of two pads placed against a specimen is given in Figure 3.7.

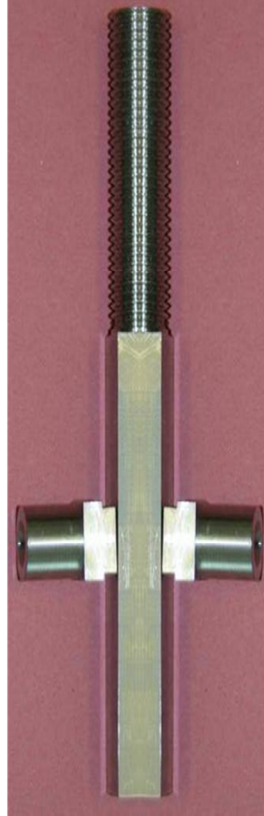


Figure 3.7: Photograph of a specimen with two pads.

Unidirectional grinding was used to produce the surface finish on the pads and specimens. A single surface condition was specified for the nickel alloy Udimet 720, while pads with two surface types were machined for the titanium alloy: Ti-6Al-4V ‘rough’ and Ti-6Al-4V ‘smooth’. Each pad pair was tested against a specimen of nominally the same material and finish. The pre-test surfaces were then scanned using the Alicona InfiniteFocus 3D profilometer (Section 3.2.1). Five $2 \times 2 \text{ mm}^2$ scans were undertaken on each pad as shown schematically in Figure 3.8.

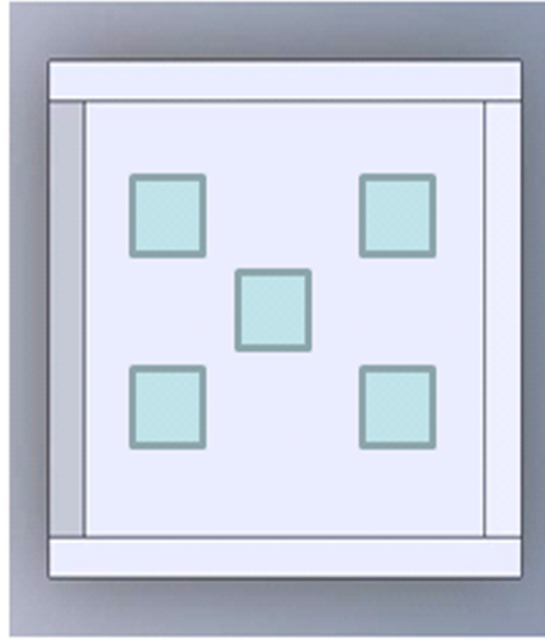


Figure 3.8: Methodology for scanning pre-test surface topography. Five $2 \times 2 \text{ mm}^2$ scan areas (as shown) were used on each pad.

A mean plane was calculated for each scan and all 12 roughness parameters outlined in Section 3.2.2 above were calculated to give a comprehensive characterisation of surface topography before testing. The settings established for measurement of pre-test roughness in Section 3.2.3 were used. Table 3.2 shows the amplitude based roughness parameters for the three surface finishes. The table gives the mean and standard deviations for all scans carried out on all pads of a particular type. For S_a , S_q , and S_z , measurements are also compared against results derived from the interferometry based Veeco instrument housed at Imperial College London. Given that the Veeco measurements were obtained in a different way (mean of two thin perpendicular $0.57 \times 6.9 \text{ mm}^2$ strips centred on each pad surface) there is reasonable agreement between the two.

Table 3.2: Pre-test amplitude based roughness parameters. Alicona results give overall averages of five $2 \times 2 \text{ mm}^2$ scans (see Figure 3.8) taken on each of six different pad surfaces. Bracketed values are standard deviations. Alicona and Veeco profilometers are compared for S_a , S_q , and S_z .

Veeco results are an overall average based on two thin perpendicular strips $0.57 \times 6.9 \text{ mm}^2$ centred on each pad surface. The settings used for measurement of pre-test roughness using the Alicona are given in section 3.2.3

	Roughness Parameter										
	S_a	S_q	S_p	S_v	S_z	S_{10z}	S_{sk}	S_{ku}	S_k	S_{pk}	S_{vk}
	(μm)	(μm)	(μm)	(μm)	(μm)	(μm)			(μm)	(μm)	(μm)
Ti-6Al-4V	1.2	1.5	7.8	7.1	14.9	13.1	-0.01	3.0	3.8	1.3	1.4
‘smooth’	(0.2)	(0.2)	(1.6)	(1.3)	(2.1)	(1.7)	(0.4)	(0.3)	(0.5)	(0.3)	(0.6)
(Alicona)											
Ti-6Al-4V	1.0	1.3			14.4						
‘smooth’											
(Veeco)											
Udimet 720	1.5	1.9	8.9	7.9	16.8	15.5	0.04	3.1	4.7	1.7	1.8
	(0.4)	(0.4)	(1.4)	(0.9)	(1.8)	(1.7)	(0.2)	(0.5)	(1.0)	(0.5)	(0.3)
(Alicona)											
Udimet 720	1.4	1.7			26.2						
(Veeco)											
Ti-6Al-4V	2.1	2.7	12.7	10.6	22.8	20.4	0.5	3.5	6.2	3.6	2.2
‘rough’	(0.3)	(0.4)	(2.9)	(3.1)	(3.9)	(2.5)	(0.3)	(0.3)	(1.1)	(0.4)	(0.7)
(Alicona)											
Ti-6Al-4V	2.5	3.1			35.9						
‘rough’											
(Veeco)											

Table 3.2 shows that the Ti-6Al-4V ‘smooth’ surfaces had the smallest roughness ($S_q = 1.49$), Ti-6Al-4V ‘rough’ had the largest roughness ($S_q = 2.74$), with nickel pads (Udimet 720) having intermediate roughness ($S_q = 1.88$). The kurtosis values (≈ 3) of the pads indicate that the surfaces are nearly Gaussian. The results of the survey do show, however, that the difference in roughness between the three surface types is not as large as was hoped due to workshop limitations. Table 3.2 contains only surface amplitude information, and any indication of the spatial separation of surface roughness elements is absent. Therefore, an auto-correlation study was also undertaken using the Alicona surface profilometer with the same measurement setup as outlined above. Figure 3.9 shows three sample auto-correlation maps of the three surface types together with the overall means and standard deviations for the following parameters: Sal, the auto-correlation length, Str, the texture aspect ratio and Stdi, the texture direction index.

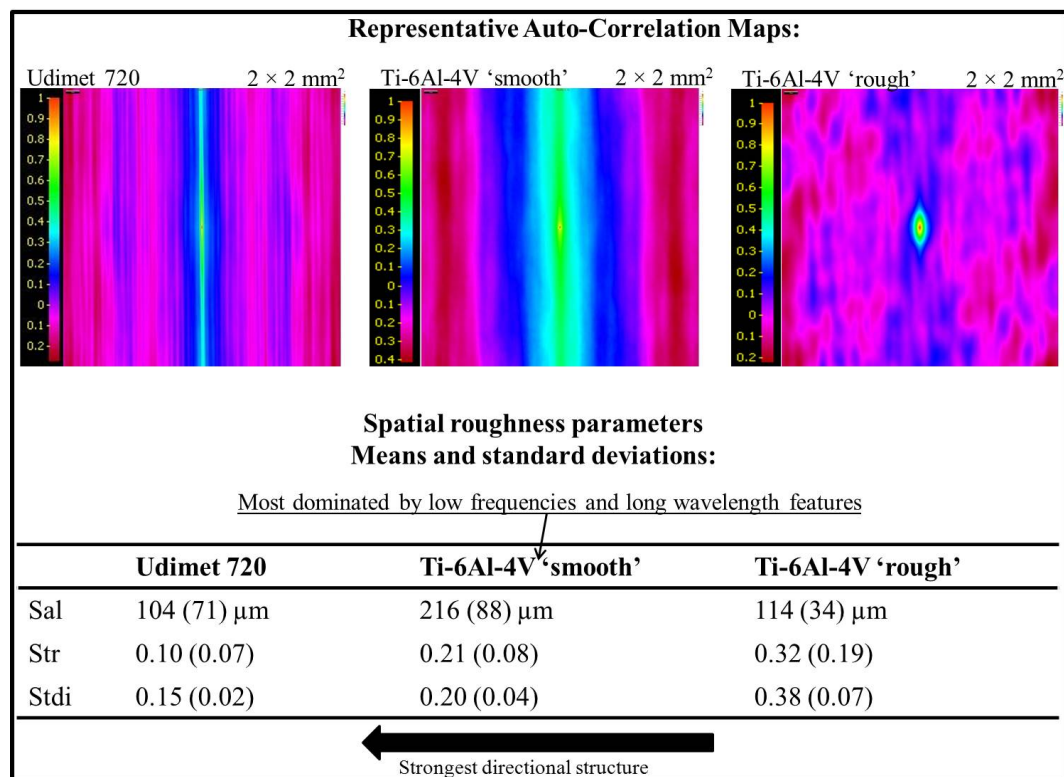


Figure 3.9: Pre-test spatial roughness parameters and accompanying representative auto-correlation maps. Bracketed values are standard deviations. Results are from five $2 \times 2 \text{ mm}^2$ scans taken on each of six different pad surfaces.

The Sal value is the shortest horizontal distance in which the autocorrelation function decays to 0.2 in any direction; thus, larger Sal values indicate low frequency surfaces with larger wavelengths (i.e. larger lateral features). Str (0–1) is the ratio of the shortest to the longest autocorrelation length: values closer to zero indicate directional anisotropy. The Stdi parameter (0–1) is a measure of how dominant the predominant direction is relative to the rest of the surface: surfaces with a predominant lay will have lower Stdi values closer to zero. Since Ti-6Al-4V ‘smooth’ has the largest Sal value: it is most dominated by low frequencies and long wavelength features, and since the Udimet 720 pad surfaces have the lowest value: they must be dominated by higher frequencies and shorter wavelength features. Also, lower Str and Stdi values indicate that the Udimet 720 pad surfaces have the strongest directional structure; whereas, the higher values obtained for Ti-6Al-4V ‘rough’ indicate less directional structure. These findings concerning frequency/wavelength and directional structure are confirmed by the autocorrelation maps of Figure 3.9, where this behaviour can be easily seen. Table 3.2 and Figure 3.9 will be of more interest when compared to corresponding post-test roughness measurements.

3.3.3 Test setup and methodology

A schematic diagram of the pads and specimen mounted in the Dartec servo-hydraulic tensile testing machine is shown Figure 3.10 and a photograph of the setup is shown in Figure 3.11. The specimen was screwed (and clamped by a wedge mechanism) into a specimen mount attached to the hydraulic actuator and given a sinusoidal oscillation frequency of 1 Hz during the tests. The amplitude of the applied peak-to-peak displacement amplitude $2\delta_{app}$ was held at 1mm. The pads were clamped in the ends of

cylindrical mounts which were free to slide along an axis perpendicular to the contact in a close-fit hole through a cast iron work-holding block. Two hydraulic pistons were then used to transmit an average normal pressure p_m of 70 MPa (via the pads) to the contact region, and a hydraulic accumulator was used to keep this pressure constant during the test. The tangential load ($2Q$ in this case) carried by the specimen was measured using a 125 kN capacity load cell mounted on the test machine. Specimen displacement δ was measured using a linear variable differential transformer (LVDT) mounted in the work-holding block – with its armature maintaining contact with the specimen mount. The resulting (actual) measured displacement amplitude δ_{actual} will, therefore, be somewhat less than that applied by the hydraulic actuator δ_{app} because of compliance between the actuator and the measurement location. The nominal accuracy of the LVDT was about 10 μm . The analogue signals from the two transducers (LVDT and load cell) were fed through a signal conditioning unit and into a National Instruments data acquisition system where a LabView computer program recorded both tangential force and displacement during the test.

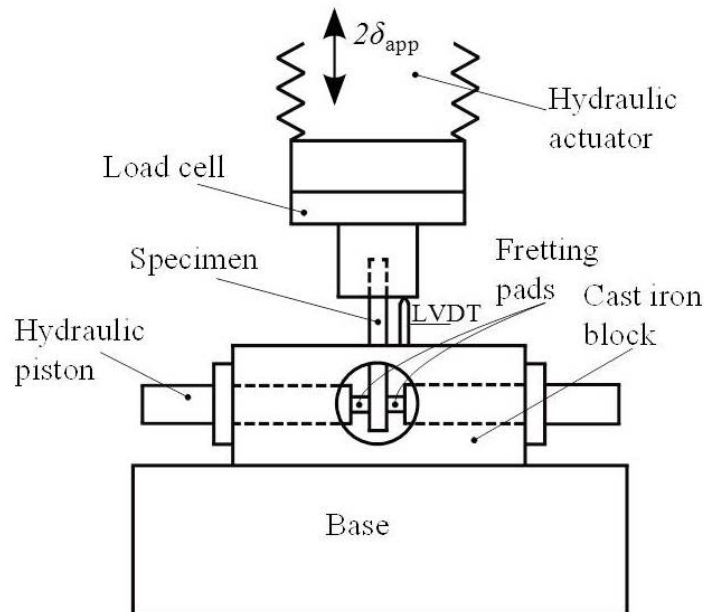


Figure 3.10: Schematic diagram of the pads and specimen mounted in the test machine.

The friction coefficient was simply calculated as half of the total tangential force at the onset of sliding in each cycle divided by the normal force (assuming the friction force at each contact is equal). Three tests with the same settings were carried out for the Ti-6Al-4V ‘smooth’ and Ti-6Al-4V ‘rough’ surface types, and two were carried out for the Udimet 720 surfaces. Tests were run until the total accumulated slip displacement had reached 5 m (about 5600 cycles). Surfaces were cleaned with acetone beforehand and testing was carried out in a ‘dry’ unlubricated condition with ambient conditions of temperature, pressure and humidity. The Dartec is a large test machine with associated safety issues. The ‘Dartec tests’ carried out for Section 3.3.4, and Section 5.3.3 were part of a collaborative research effort between the author and Dr Mehmet E. Kartal of the Solid Mechanics Group.



Figure 3.11: Photograph showing the Dartec servo-hydraulic tensile testing machine with the test fixtures attached.

3.3.4 Results and Discussion

Sample hysteresis loops taken after 2500 cycles (plotting total tangential force $2Q$ against LVDT displacement) are shown in Figure 3.12a for Ti-6Al-4V ‘rough’ and in Figure 3.12b for Udimet 720.

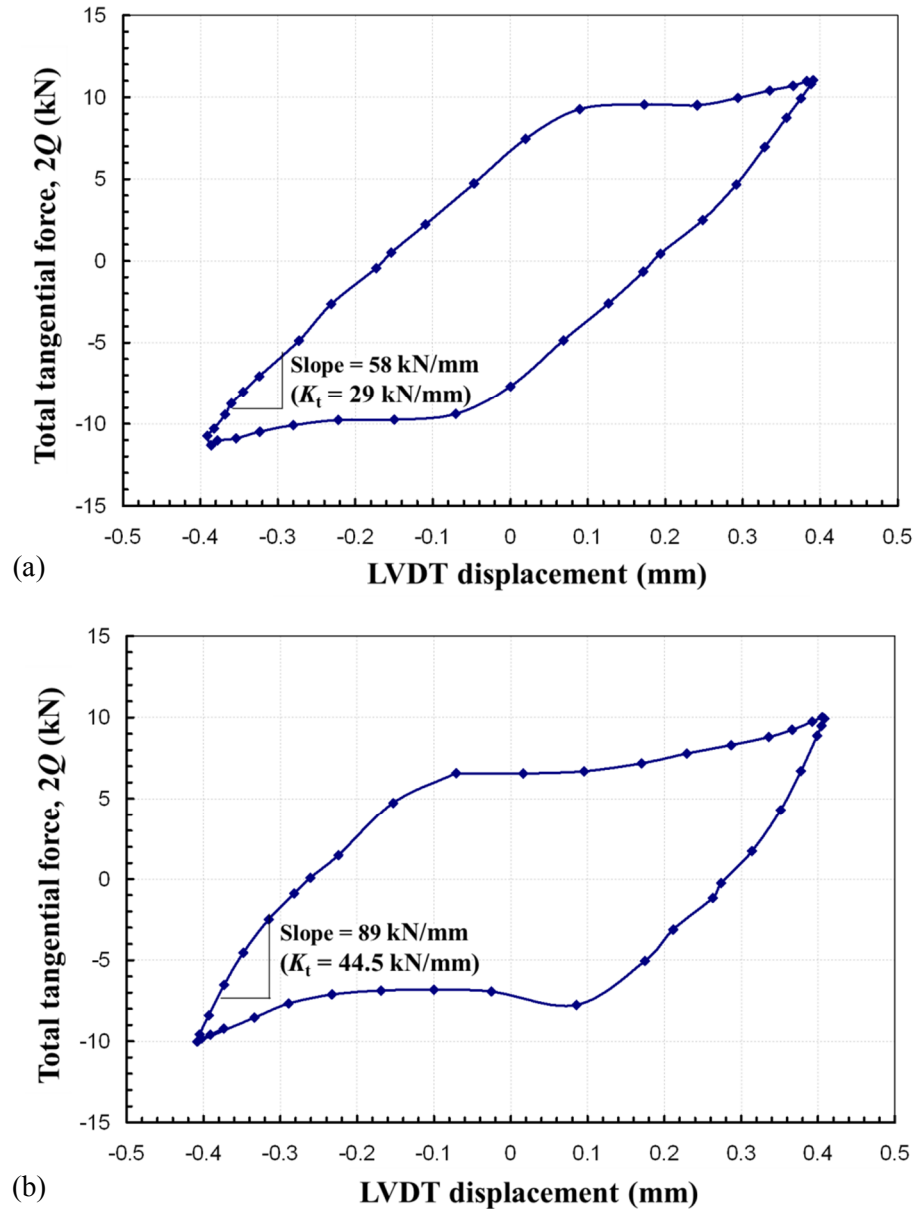


Figure 3.12: Hysteresis loops measured at $N_c = 2500$ cycles for: (a) Ti-6Al-4V ‘rough’ surface type, and (b) the Udimet 720 surface type.

It can be seen that both the titanium alloy (Ti-6Al-4V) and the nickel alloy (Udimet 720) exhibit the phenomenon discussed in Section 2.2.3 whereby the friction force varies

(usually increases) during the gross-slip phase of the fretting cycle. The increase in friction appears to be more distinct for the Udimet 720 than for the Ti-6Al-4V and this is representative of the other experiments also. Also shown in Figure 3.12 are the stiffness values K_t which are determined from the initial slope of the loop just after the direction of motion reverses. Since Figure 3.12 plots the total tangential force contribution from both pads, the equivalent stiffness values for a single contact are determined by halving the slope values shown. Stiffness values of 29 and 44.5 kN/mm for Ti-6Al-4V ‘rough’ and Udimet 720, respectively, were obtained. These stiffness values will inevitably be far less than the stiffness of the contact itself as they include bulk compliance contributions from both the pad and specimen as well as their holding fixtures. This is because the LVDT takes its measurement (remotely) from near the top of the specimen and is only a measure of specimen displacement rather than relative displacement. The stiffness for the Udimet 720 is 1.5 times higher than for the Ti-6Al-4V in Figure 3.12: this is probably because the Young’s modulus of the Udimet 720 is 1.9 times that of Ti-6Al-4V (see Table 3.1). In Section 3.4 an attempt is made to use these stiffness values to isolate the ‘true’ contact stiffness.

Figure 3.13 shows the evolution of the coefficient of friction (calculated at the onset of the sliding phase of each cycle) during all tests carried out on the three surface types. The results show that the nickel alloy, Udimet 720, (Figure 3.13c) gives somewhat higher friction coefficients than the titanium alloy, Ti-6Al-4V, (Figure 3.13a and b). Comparing the friction results for the Ti-6Al-4V ‘smooth’ (Figure 3.13a) and the Ti-6Al-4V ‘rough’ (Figure 3.13b) surface types, it is noted that the results are very similar (with Ti-6Al-4V ‘rough’ showing slightly higher friction coefficients). The similarity in results between ‘rough’ and ‘smooth’ Ti-6Al-4V is probably due to the fact that the difference in pre-test RMS roughness (S_q) was only about 1.25 μm (see Table 3.2), and because, as we will see,

severe wear occurs which removes any significant influence of the initial roughness. Figure 3.13 shows that the friction coefficient measurements from a particular surface type were reasonably repeatable when tests were repeated with the same settings. Also, the friction increase towards the beginning of the tests due to ‘bedding in’ can be observed. After about 800 cycles, the friction coefficient enters a ‘steady-state’ region and only changes moderately throughout the remainder of the tests.

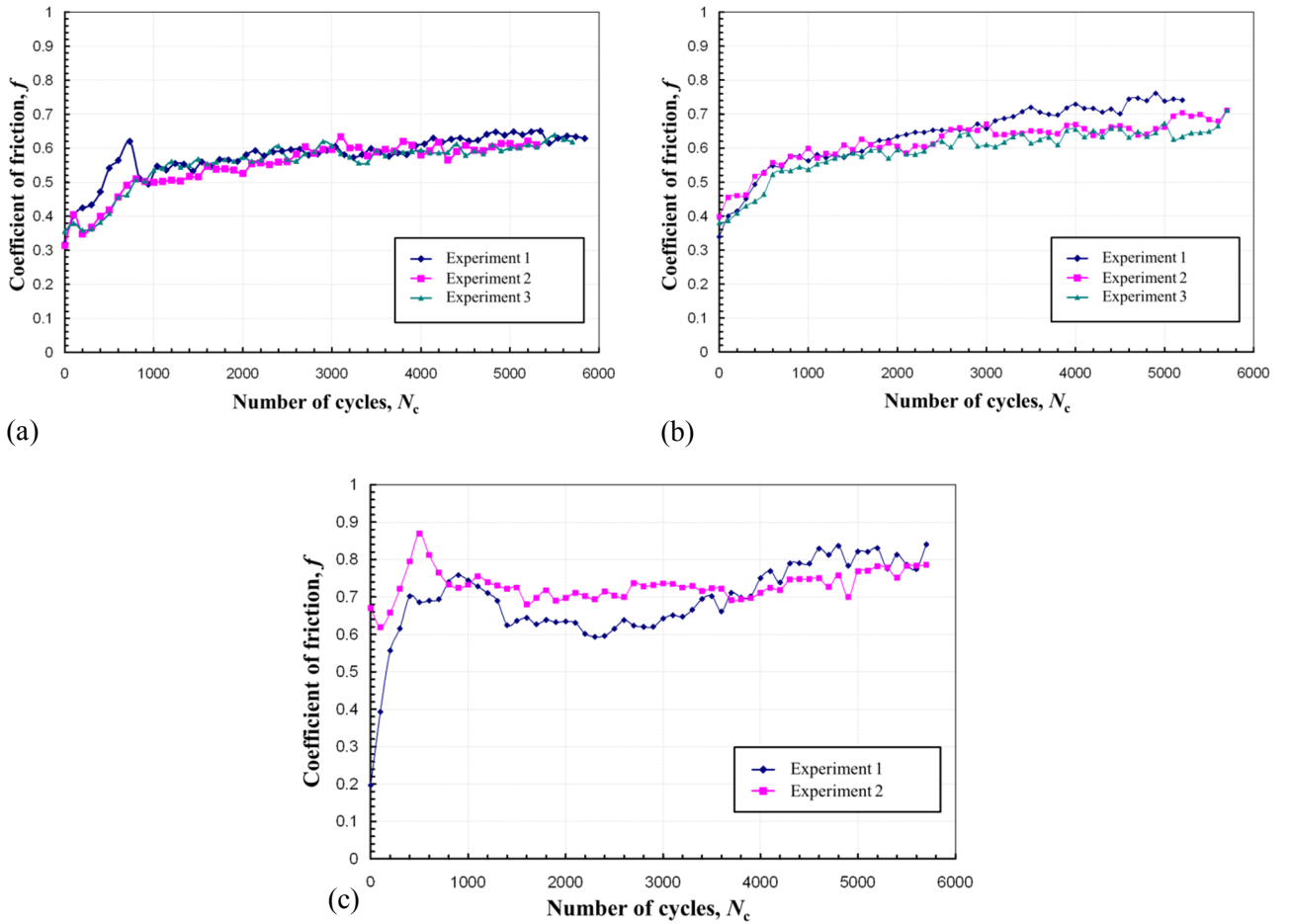


Figure 3.13: Evolution of the coefficient of friction (as calculated at the onset of sliding in each cycle) with number of test cycles for: (a) Ti-6Al-4V ‘smooth’, (b) Ti-6Al-4V ‘rough’, and (c) Udimet 720.

To obtain more information about the processes occurring during the tests, the post-test wear scars were imaged using the Alicona InfiniteFocus profilometer (Section 3.2.1). Following testing, the pads and specimens were carefully removed from the test rig and record was taken of which pads were in contact with which side of a given specimen. In

order to preserve the features existing during the test, the wear scars were not cleaned or disturbed in any way. The entire nominal contact area ($8 \times 10 \text{ mm}^2$) on the pads and specimens was scanned. Different settings to those set out in Section 3.2.3 for the measurement of pre-test surfaces were required because a larger scan area was being taken and because the surface texture was more challenging to image. A vertical resolution of $1.5 \text{ }\mu\text{m}$, horizontal resolution of $3.5 \text{ }\mu\text{m}$, and a $\times 5$ objective were used. The ring-light was used to improve lighting and the polariser was used to even out the brightness on the wear scars. Figure 3.14 and Figure 3.15 show post-test wears-scar pairs for Ti-6Al-4V ‘rough’ and Udimet 720, respectively. Each figure shows wear scars from two identical tests. For each test, the wear-scar pair from both the left-hand and right-hand contact is shown, and each pair is shown so that a rotation of the pad wear scar (following the arrow shown) would place it in contact with the specimen in the configuration that existed during testing.

It is immediately obvious that the wear-scar structure on the titanium alloy (Figure 3.14) is different from that for the nickel alloy (Figure 3.15) and the difference is repeatable as the four wear-scar pairs in each figure show. The Ti-6Al-4V tests produce W-shaped wear scars: for the specimen, a high ridge in the middle and two troughs at the sides (both spanning the width of the wear scar in the direction perpendicular to sliding) are visible; for the pads, a corresponding structure with a central low valley and two high ridges towards the edges can be observed. The Udimet 720 wear scars, on the other hand, exhibit a few randomly situated local high spots on the pads which have worn corresponding pits in the specimen wear scar. This difference in wear-scar geometry may explain the difference in how the friction force varies somewhat differently with sliding for the two alloys (see Figure 3.12). It is clear from the wear scars that a significant degree of material transfer and accumulation of wear debris must be occurring.

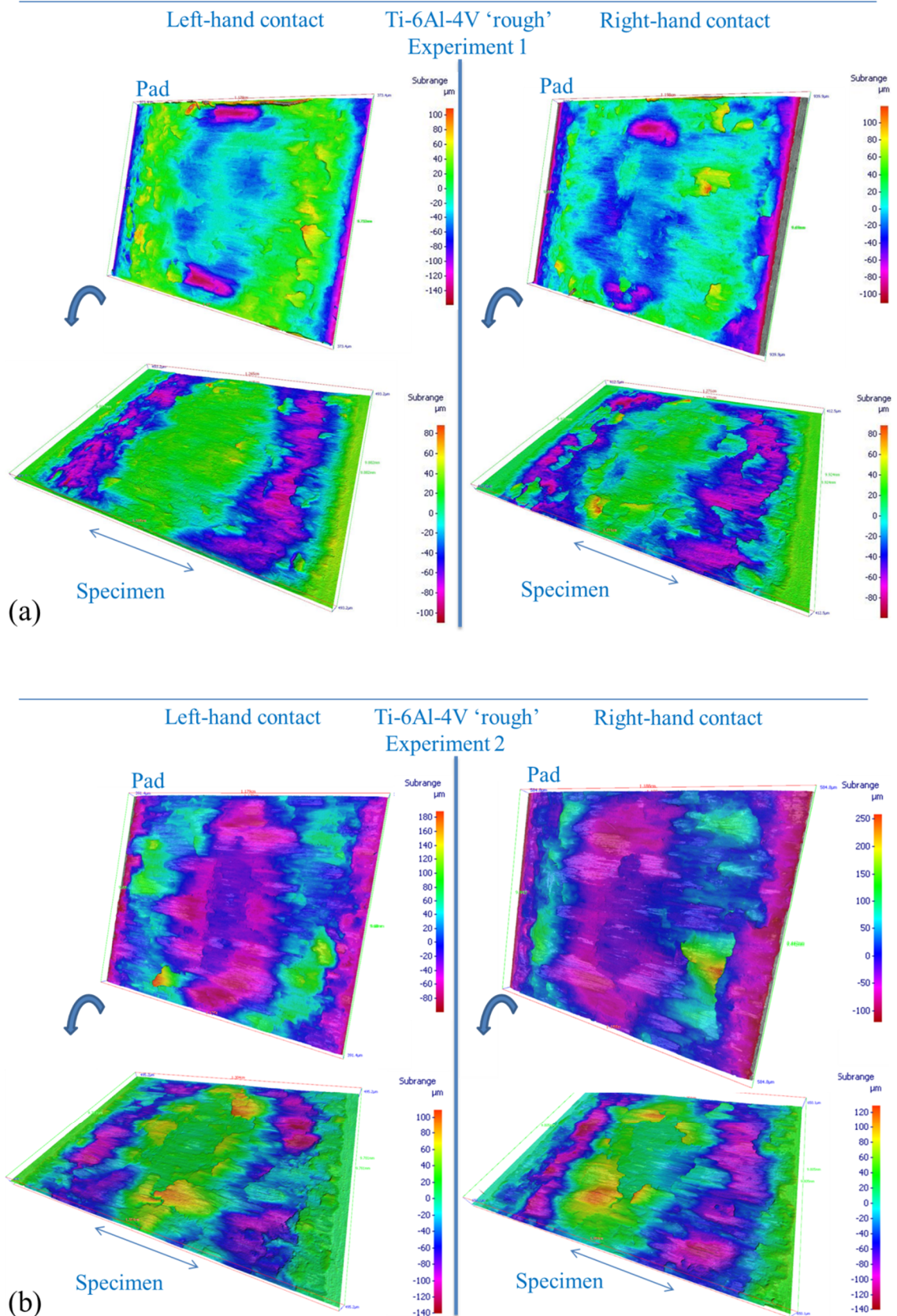


Figure 3.14: Post-test wear-scar pairs (after 5600 cycles) for the Ti-6Al-4V 'rough' surface type: (a) Experiment 1, and (b) Experiment 2. Experiment 1 and 2 are repeat tests with identical test settings. The double-headed arrows show the fretting direction.

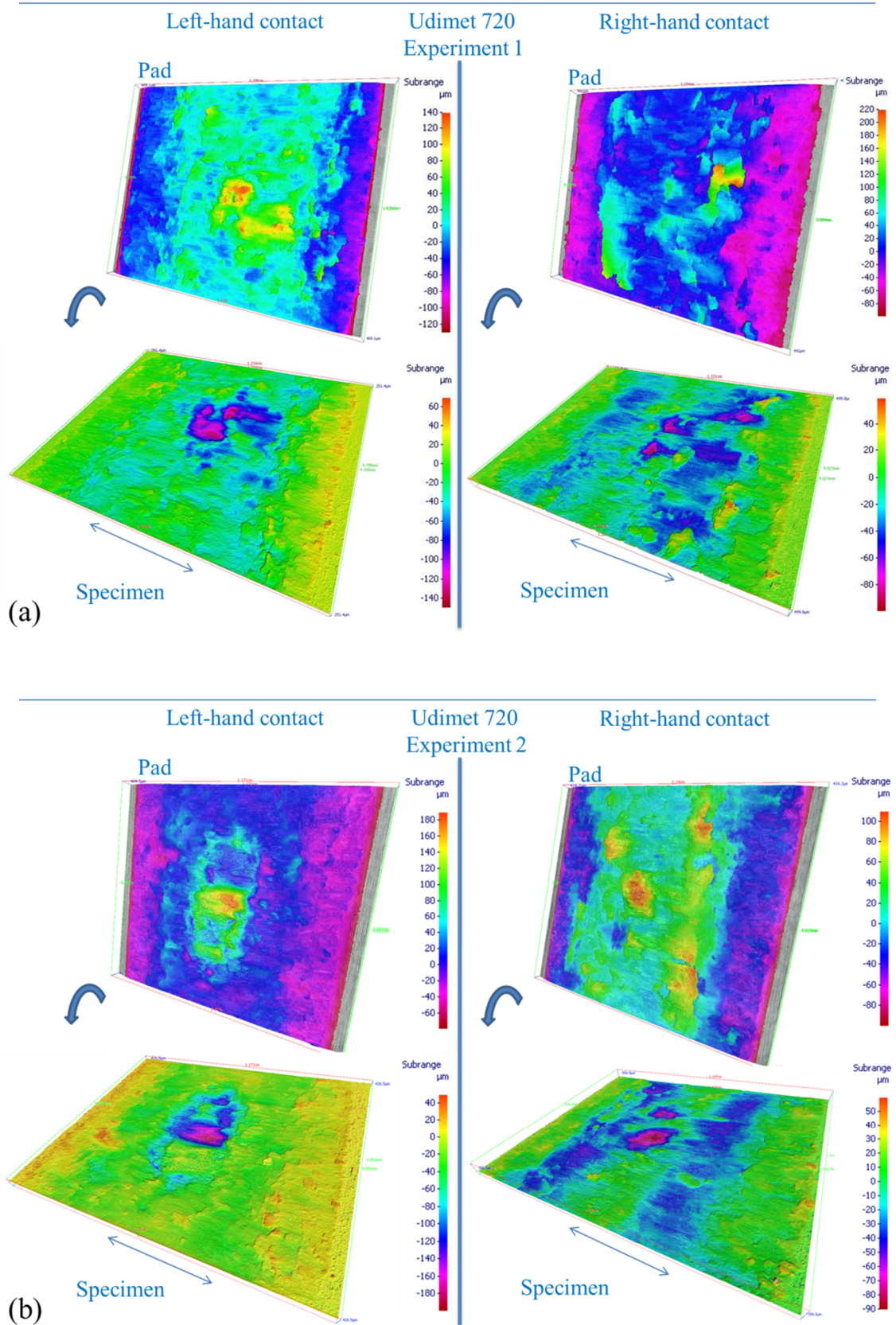


Figure 3.15: Post-test wear-scar pairs (after 5600 cycles) for the Udimet 720 surface type: (a) Experiment 1, and (b) Experiment 2. Experiment 1 and 2 are repeat tests with identical test settings. The double-headed arrows show the fretting direction.

Magaziner *et al.* [40], Mohd Tobi *et al.* [21] and Everitt *et al.* [22] have all observed the W-shaped wear scar in fretting-wear tests on Ti-6Al-4V using the cylinder-on-flat arrangement. The same type of behaviour occurs here with Ti-6Al-4V for a different contact configuration but does not occur for the Udimet 720: this suggests that the chemical composition of the material and its tribological properties are probably the deciding factors in producing the W-shape. Mohd Tobi *et al.* [21] had put forward an explanation based on the fact that the distribution of accumulated plastic strain across the contact in their continuum FE models also had a W-shape, while Magaziner *et al.* [40] had hypothesised that slight rotation of the pads might cause a ‘dig-in’ effect which would produce the two valleys in the W-shape. However, there were no difference in conditions for the tests carried out here on Ti-6Al-4V and Udimet 720 that would have changed the conclusion of either of these explanations, yet Ti-6Al-4V gives a W-shaped scar, while Udimet 720 does not.

The wear scars indicate that fretting wear during the tests was more significant than was originally envisaged when defining the experimental programme. Figure 3.16a and Figure 3.16b show optical micrographs of the post-test wear-scar surfaces of the right-hand pads from Experiment 1 in Figure 3.14a and Figure 3.15a, respectively (i.e. for Ti-6Al-4V and Udimet 720). Regions of wear debris together with some exposed metal patches can be seen. Some areas indicative of adhesive and abrasive wear can also be seen in the micrographs (the shiny silvered metallic areas). As the contact consisted of a large enclosed region (unlike, for example, a cylinder-on-flat test), debris will probably have found it difficult to escape with the result that a significant ‘third body’ layer is more likely to have developed in parts of the contact (see the purplish areas and black areas in Figure 3.16 below).

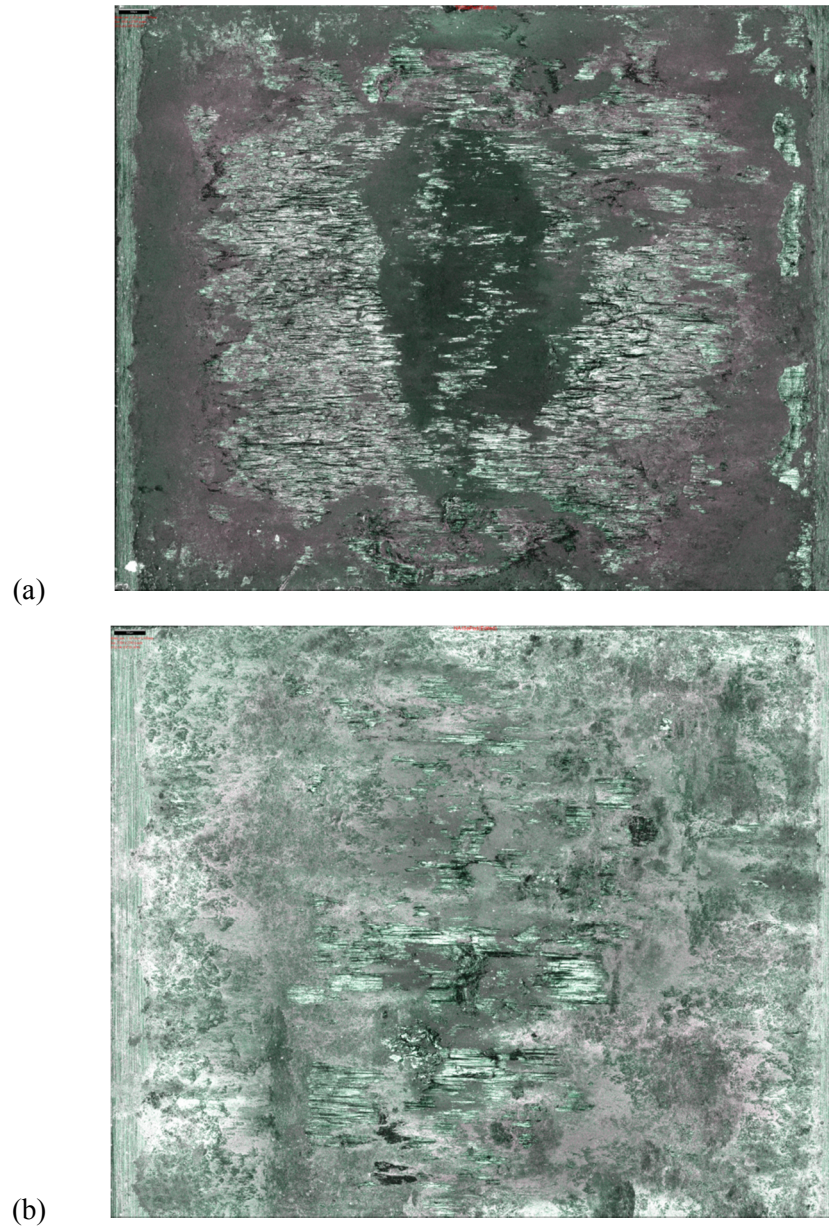


Figure 3.16: Optical post-test wear-scar images: (a) the right-hand Ti-6Al-4V 'rough' pad surface from Experiment 1 (Figure 3.14a), and (b) the right-hand Udimet 720 pad surface from Experiment 1 (Figure 3.15a). The approximate size of the images is 8 mm $\times$ 10 mm.

The same 3D areal roughness parameters that were measured prior to testing were then measured from the surface scans which were obtained as described above. The post-test amplitude based roughness parameters are given in Table 3.3.

Table 3.3: Post-test amplitude based primary roughness parameters measured using the Alicona InfiniteFocus profilometer. Results from two region types are given: the averages of five $2 \times 2 \text{ mm}^2$ areas on each pad, and the averages from a single $7 \times 7 \text{ mm}^2$ area on each pad. A total of eight pads were scanned. Bracketed values are standard deviations.

	Roughness Parameter										
	S_a	S_q	S_p	S_v	S_z	S_{10z}	S_{sk}	S_{ku}	S_k	S_{pk}	S_{vk}
	(μm)	(μm)	(μm)	(μm)	(μm)	(μm)			(μm)	(μm)	(μm)
Udimet 720	18	25	113	71	184	157	0.50	4	44	33	26
$2 \times 2 \text{ mm}^2$	(6.4)	(9.4)	(81.3)	(36.5)	(97.3)	(60.3)	(1.2)	(1.0)	(16.4)	(21.3)	(19.0)
region											
Udimet 720	22	33	190	111	298	255	0.52	5	53	42	38
$7 \times 7 \text{ mm}^2$	(5.1)	(9.5)	(117.2)	(59.8)	(124.3)	(82.9)	(1.8)	(1.7)	(9.5)	(27.7)	(25)
region											
Ti-6Al-4V	18	26	95	80	175	159	0.48	5	44	39	29
‘rough’	(7.5)	(11.1)	(46.7)	(39.9)	(70.5)	(61.1)	(1.3)	(1.9)	(15.7)	(27.8)	(18.4)
$2 \times 2 \text{ mm}^2$											
region											
Ti-6Al-4V	27	38	158	153	311	266	0.11	5	58	45	52
‘rough’	(6.5)	(9.1)	(56.3)	(50.3)	(80.1)	(31.1)	(1.5)	(0.9)	(14.9)	(22.3)	(25.2)
$7 \times 7 \text{ mm}^2$											
region											

Two values are reported for each parameter: one based on an area of $2 \times 2 \text{ mm}^2$ and the other on an area of $7 \times 7 \text{ mm}^2$. The $2 \times 2 \text{ mm}^2$ region size was chosen as this was the size of the pre-test scans and the $7 \times 7 \text{ mm}^2$ region was also included as it emerged that the 2

$\times 2 \text{ mm}^2$ size was too small to describe some of the longer wavelength features produced by the fretting wear process.

Comparing the post-test amplitude based roughness parameters in Table 3.3 to the equivalent pre-test values in Table 3.2 allows us to quantitatively assess the change in surface topography which occurred during the 5600 cycles of testing. It is clear from consideration of all of the roughness parameters taken as a whole that a dramatic increase in surface roughness has occurred during the tests. Taking the S_q parameter as a measure of surface roughness, and using the $7 \times 7 \text{ mm}^2$ scans for the post-test values, we see that the parameter increased from 2.7 to 38 μm for Ti-6Al-4V ‘rough’ and from 1.9 to 32 μm for Udimet 720. This is an increase by a factor of 14 for both Ti-6Al-4V ‘rough’ and Udimet 720. Similarly, taking S_{10z} as a robust measure of surface thickness (i.e. flatness), it is noted that for the Ti-6Al-4V ‘rough’ surfaces, S_{10z} increased from a value of 20.4 to 266 μm , while for the Udimet 720 surfaces, S_{10z} increased from 15.5 to 255 μm . This is a 13 fold increase for Ti-6Al-4V ‘rough’ and a 16.5 fold increase for Udimet 720. The other roughness parameters such as S_a , S_p , S_v , S_z , S_k , S_{pk} , and S_{vk} all reflect the same trend. The high surface thickness of the wear scars can be clearly observed from Figure 3.14 and Figure 3.15 by noting the difference between the lowest surface points and the highest surface points with the help of the height legend. The dramatic change in surface roughness occurring during the tests means that it is very difficult to notice any effect of pre-test surface finish on results derived from fretting tests of this kind.

Finally, Figure 3.17 shows the mean post-test spatial roughness parameters Sal , Str , and $Stdi$ (together with representative autocorrelation maps) for the Udimet 720 and Ti-6Al-4V ‘rough’ surfaces. The Sal parameter gives an indication of the dominant feature size in the lateral direction. Prior to testing, the mean Sal value was 104 μm for the Udimet 720 surfaces, and 114 μm for the Ti-6Al-4V ‘rough’ surfaces (Figure 3.9); whereas, after

testing, the Sal value was 1021 μm for Udimet 720, and 1043 μm for Ti-6Al-4V ‘rough’ (Figure 3.17). This is most interesting since the peak-to-peak applied amplitude $2\delta_{\text{app}}$ during testing was 1000 μm . The suggestion here is that features similar in size to the applied peak-to-peak fretting amplitude are being created by the fretting wear process. The possibility that these wear scar features are responsible for the friction variation phenomenon during the sliding phase of the fretting cycles (as observed in Figure 3.12) will be investigated thoroughly in the next Chapter (Chapter 4). The fact that the post-test dominant feature sizes (as estimated by the Sal values) are similar for both materials (Ti-6Al-4V and Udimet 720) is also noteworthy and suggest that the generation of features of the same size as the imposed peak-to-peak amplitude is possibly independent of material.

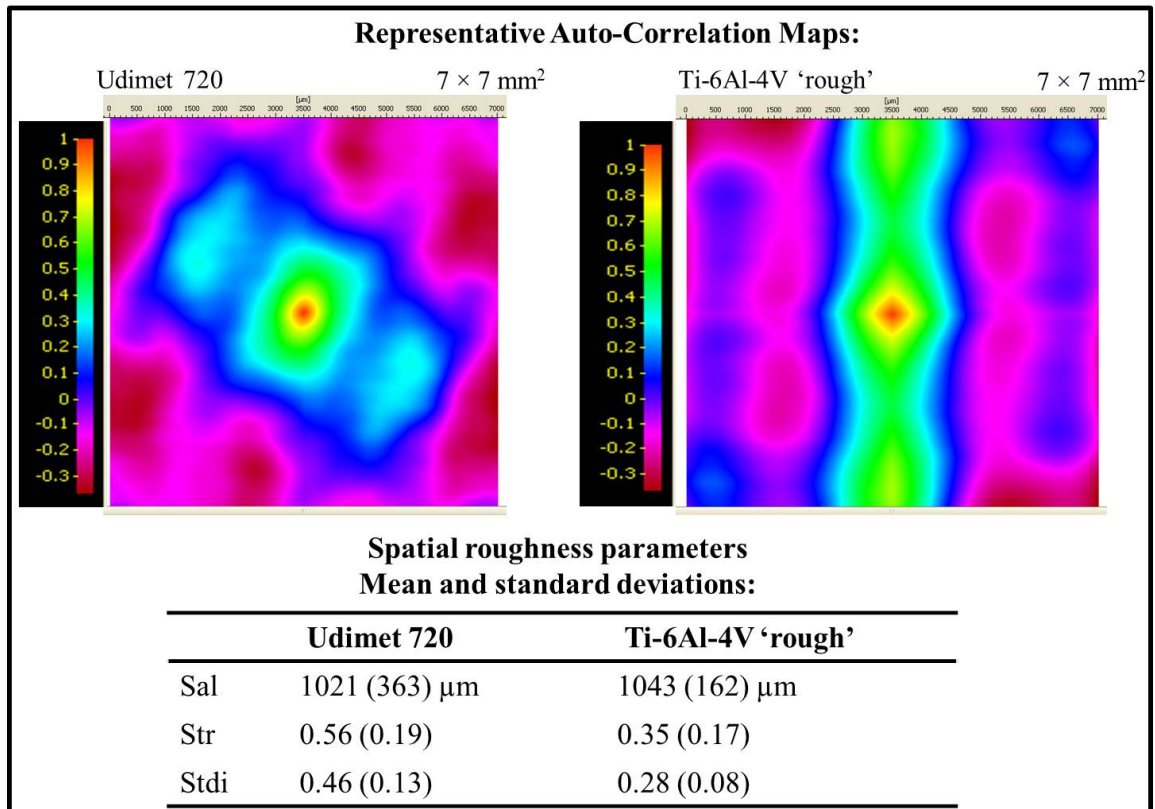


Figure 3.17: Post-test spatial roughness parameters and accompanying representative auto-correlation maps. Bracketed values are standard deviations. Results are from a single scan taken on eight pads.

3.4 Monolith experiment for isolation of contact stiffness

In the friction tests carried out in Section 3.3, tangential stiffness measurements were made based on LVDT measurements of displacement measured remote from the contact interface (i.e. near to the top of the specimen). One of the aims of the present work is to measure interface stiffness, however, these stiffness values (shown in Figure 3.12) also contain additional stiffness elements from bulk material compliance and from movements in the pad and specimen holding fixtures. In an attempt to use these results to isolate the tangential contact stiffness of the rough interface itself, a ‘monolith specimen’ was built where the pads and specimen constituted a single solid piece of material without any interfaces. The monolithic test specimen was manufactured from Ti-6Al-4V and is shown in Figure 3.18.

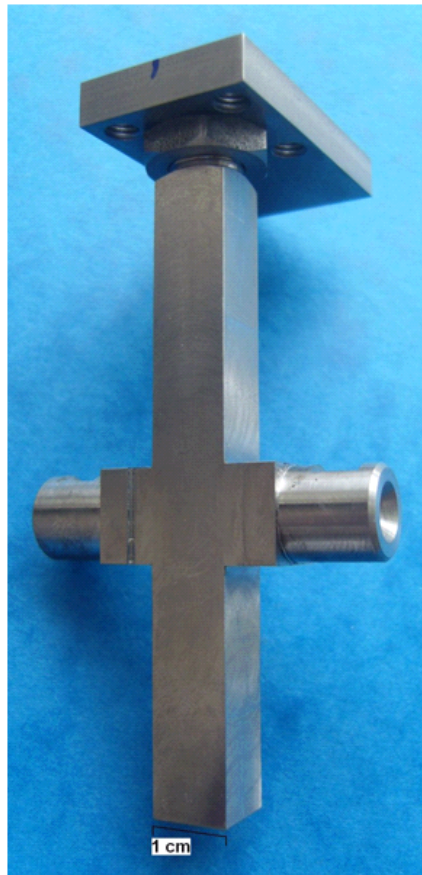


Figure 3.18: Photograph of the ‘monolith specimen’.

The idea behind this approach is that the tangential stiffness measured from the ‘monolith specimen’ (which contains no interfaces) can be compared to the tangential stiffness from the pad-specimen friction experiments in order to allow isolation of the interface stiffness – since the only difference between the two experiments is ‘apparently’ the two rough contact interfaces. If $K_{\text{Pad/Specimen}}$ and K_{Monolith} are the stiffness values measured remotely at identical positions using an LVDT for the pad/specimen and ‘monolith’ tests, respectively, then it should be possible to obtain the stiffness of the interface $K_{\text{Interface}}$ by considering the pad/specimen stiffness as being equivalent to the series sum of the interface stiffness and the monolith stiffness:

$$\frac{1}{K_{\text{Pad/Specimen}}} = \frac{1}{K_{\text{Monolith}}} + \frac{1}{K_{\text{Interface}}}. \quad (3.9)$$

The ‘monolith specimen’ was also tested in the Dartec servo-hydraulic tensile testing machine with the same test fixtures and apparatus as outlined for the pad/specimen experiments in Section 3.3. The distance from the LVDT measurement location to the pad axis was chosen to be the same as for the Ti-6Al-4V ‘rough’ pad/specimen experiment for which the resulting stiffness at 2500 cycles was shown in Figure 3.12a. Figure 3.19 shows the hysteresis loop recorded from the ‘monolith test’ which gives a stiffness K_{Monolith} of 49.6 kN/mm. It can be seen that this value is considerably stiffer than the result shown in Figure 3.12a for the Ti-6Al-4V ‘rough’ pad/specimen test which gave $K_{\text{Pad/Specimen}} = 29$ kN/mm after 2500 cycles of testing. Now, applying Equation 3.9 to the stiffness values from the ‘monolith’ and ‘pad/specimen’ experiments, gives an interface stiffness $K_{\text{Interface}} = 69.8$ kN/mm. Kartal *et al.* [94], however, have made measurements of tangential contact stiffness during gross-slip fretting based on relative displacements local to the contact using digital image correlation (DIC). These tests were

also on Ti-6Al-4V and used the same pad and specimen types as used in the present chapter with an $8 \times 10 \text{ mm}^2$ contact area.

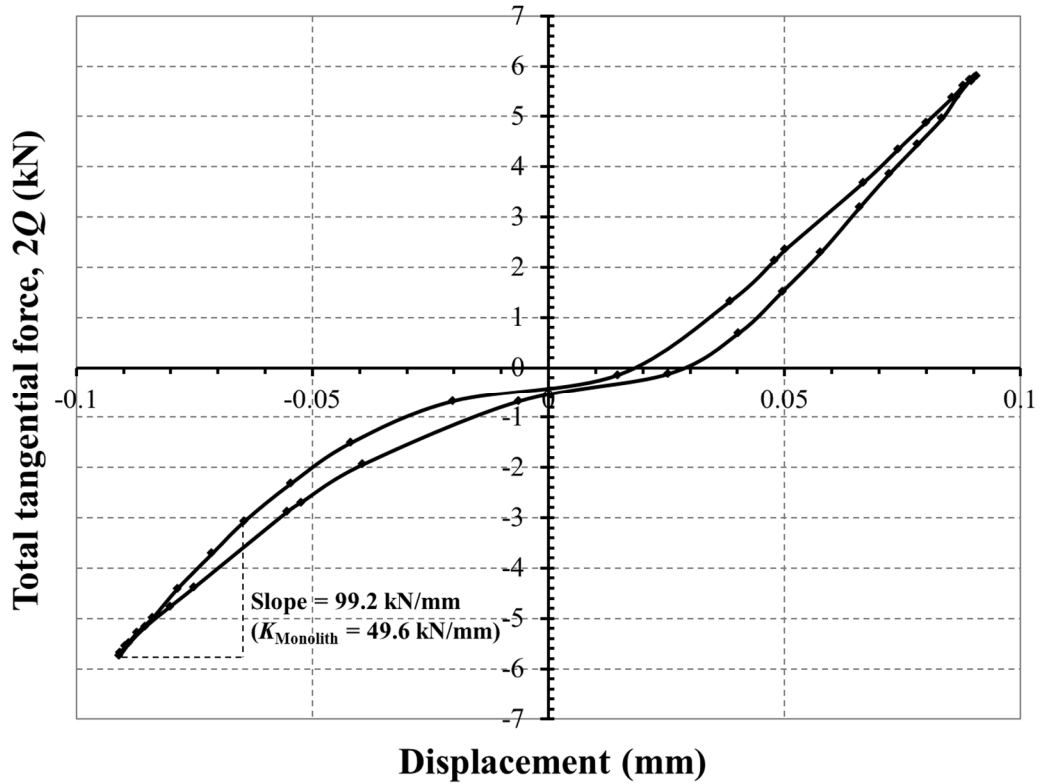


Figure 3.19: Hysteresis loop from the ‘monolith experiment’.

Kartal [124] measured a tangential contact stiffness value of 950 kN/mm (per single contact) by DIC at 2500 cycles during the same testing programme outlined in [94]: this is 13.6 times stiffer than the result predicted by the present analysis ($K_{\text{Interface}} = 69.8 \text{ kN/mm}$) using the ‘monolith test’. This suggests that there must be other differences in stiffness elements between the two tests other than the interface stiffness. This probably comes about because the pads behave stiffer when integrally attached to the specimen in the ‘monolith case’ than when they are separate from the specimen and free to move as in the case of the pad-specimen friction experiment. In any case, it seems that the interface stiffness is very difficult to isolate using this method since any such extra difference in stiffness contributions between the two tests will obscure the interfaces stiffness.

3.5 Conclusions

Initial precursory work has been carried out to investigate the nature of frictional behaviour in gross-slip fretting. Methods and settings to allow for the use of the new 3D areal roughness parameters and the latest focus variation based surface profilometry techniques in the analysis of surface texture evolution during fretting have been established. Both spatial and amplitude based primary roughness parameters were found to be useful in characterising pre and post-test surfaces. Tests were carried out on three surface types (Ti-6Al-4V ‘smooth’, Ti-6Al-4V ‘rough’, and Udimet 720) in order to examine the effect of surface finish and material on frictional properties. The applied normal contact pressure was 70 MPa, and the applied peak-to-peak displacement amplitude was 1 mm. Both material types displayed the phenomenon whereby the friction force varies (increases) during the sliding phase of individual fretting cycles – though the Udimet 720 showed, on the whole, a greater variation.

Examination of the post-test 3D wear scar profiles, as well as comparison of post-test roughness with pre-test roughness parameters, revealed that significant wear occurred during the tests. For example, after 5600 cycles of testing, the primary surface roughness parameter S_q was about 14 times larger for both Ti-6Al-4V and Udimet 720 than at the start of the tests. Wear-scar structure was distinctly different between Ti-6Al-4V and Udimet 720. Ti-6Al-4V formed W-shaped wear scars with the ridges and valleys of the “W” running across the width of the wear scars in the direction perpendicular to sliding. Udimet 720 wear scars, on the other hand, exhibited a few randomly positioned high spots on the pads which wore corresponding pits in the specimen wear scars. It is postulated that difference in wear-scar geometry, such as this, can explain differences in how the friction varies during the sliding stroke. It was also clear from the wear scars

that significant material transfer and wear debris accumulation was occurring in the contact region. A particularly interesting finding was that the post-test auto-correlation length (Sal) was not only similar for the two materials tested (1021.3 μm for Udimet 720, and 1042.6 μm for Ti-6Al-4V), but was similar in magnitude to the applied peak-to-peak displacement amplitude of 1000 μm which was used during the tests. Since auto-correlation length gives an indication of the prevailing size of surface features in the lateral direction, this suggests that wear-scar features of about the same size as the imposed peak-to-peak displacement amplitude developed during the tests as part of the fretting wear process. The notion that the interaction of these features on the pad and specimen surfaces is responsible for the friction variation during the sliding stroke, and that their shape influences the form of that variation, will be investigated thoroughly in the next chapter.

Friction coefficients were easily calculated as the tangential load at the initiation of sliding in each cycle divided by the constant normal load. The Udimet 720 gave slightly higher friction coefficients (≈ 0.7 on average) than the Ti-6Al-4V (≈ 0.6 on average) after the initial ‘bedding in’, but little difference was observed between Ti-6Al-4V ‘smooth’ and Ti-6Al-4V ‘rough’ since the initial difference in roughness was small, and because significant wear removed any influence of initial roughness. Hence, this type of test is not suitable for investigating the effect of pre-test roughness on frictional behaviour unless wear is reduced and the difference in pre-test surface roughness specification is increased. Good repeatability in the evolution of the friction coefficient was observed when tests were repeated with the same settings. In addition, the initial increase in friction due to ‘bedding in’ was noted. After about 800 cycles, the friction coefficient was relatively steady with a very gradual increase being observed thereafter for most of the tests.

An LVDT was used to measure tangential displacement during the tests, but this was done remote from the contact region. Therefore, compliance due to bulk material movement and flexibility in the pad and specimen holding fixtures will also have been included in the tangential stiffness which was calculated from the hysteresis loops. In order to try to isolate the interface stiffness from these remote contributions, a ‘monolith specimen’ comprising the pads and specimen constructed as a single solid piece was built and tested in the same way as with the pad/specimen friction experiment. This approach was predicated on the assumption that the only difference in stiffness between the two arrangements would be the stiffness of the contact interface. However, it was found that the monolith arrangement was stiffer than expected. The resulting calculations thus required an interface stiffness which was too low to be realistic in order to give the same combined result (monolith stiffness in series with the interface stiffness) as the stiffness measured from pad/specimen frictional test. This is thought to be due to the fact that the pads behave stiffer when integrally attached to the specimen than when free to move in the pad-holders. It is concluded that this type of approach to isolating interface stiffness is not very useful since any extra difference in the stiffness contributions (other than the contact stiffness) between the ‘monolith test’ and the ‘frictional test’ will obscure the interface stiffness. Therefore, it is clear that more local methods of determining contact stiffness are required. In Chapter 6, this issue is addressed through the use of ultrasound and digital image correlation.

Chapter 4

Investigation of non-Coulomb friction in gross-slip fretting

4.1 Introduction

Experiments in Chapter 3, as well as the literature surveyed in Section 2.2.3, confirm that it is common for the friction force to vary markedly during the gross-slip phase of individual fretting cycles. This variation usually involves a notable increase in the friction force as the sliding stroke proceeds and produces a ‘hook’ like feature on the frictional hysteresis loops. This is contrary to the ‘Coulomb view’ of friction which assumes a constant friction force during sliding (see Figure 4.1). As noted already, the frictional hysteresis loop determines the energy which is dissipated by vibration damping in a joint and converted to frictional work; therefore, the sliding friction behaviour is critical in influencing the damping and wear characteristics of a fretting joint. Despite the importance of this phenomenon, and the inconsistency between experiment and the commonly accepted ‘laws’ of friction, there appears to be a lack of experiments in the literature to determine the source of the increasing friction force during sliding. The

present chapter sets out to determine the physical cause of the behaviour. The proposition that the phenomenon is caused by wear-scar interaction effects is investigated. Further, if worn surface interaction is the underlying mechanism, it is important to determine whether this occurs mainly at the macroscopic edges of the wear scars, or whether it happens throughout the entire contact region at individual interlocking pairs of local surface peaks and depressions caused by the fretting wear process (see Figure 4.2). A series of fretting experiments (both linear and torsional) have been undertaken to investigate these issues. The subsequent worn surfaces were then analysed using both qualitative and quantitative means to determine if there are surface features on the wear scars which could be responsible for the effect. In addition, a simple one degree-of-freedom model is proposed to help explain the observed results.

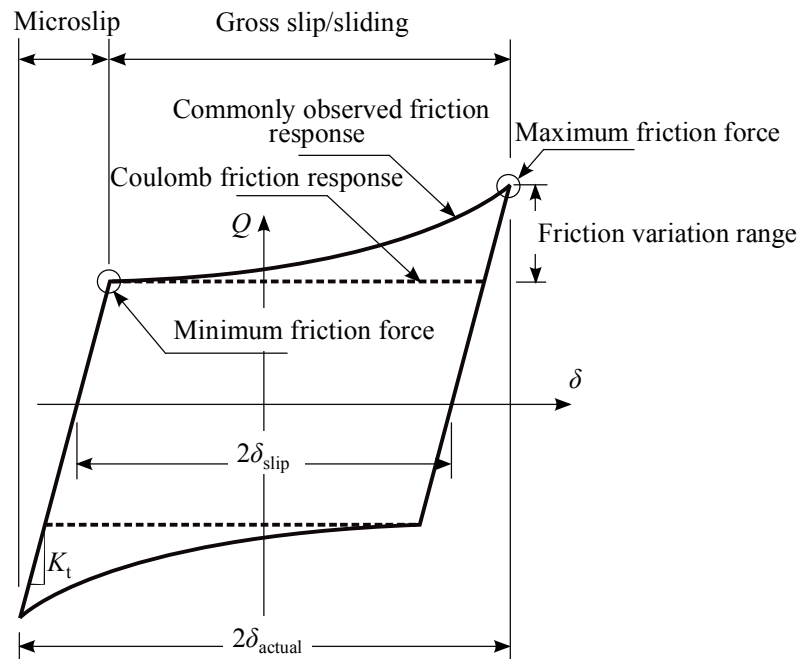


Figure 4.1 Hysteresis loops showing both the Coulomb model and the frictional behaviour commonly observed in gross-slip fretting tests.

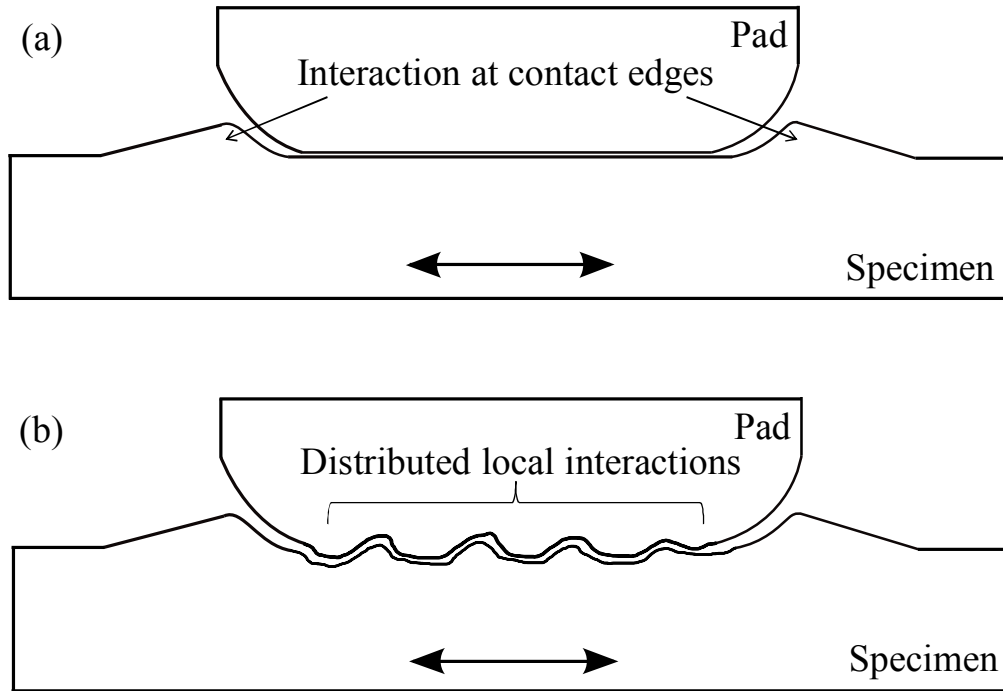


Figure 4.2: Schematic diagram of wear-scar interaction types: (a) interaction at contact edges and (b) distributed local interactions (as well as the ‘end interaction’).

4.2 Linear fretting test

As a prelude to more extensive experiments using a torsional rig, the results of a single linear fretting test carried out using the Dartec servo-hydraulic tensile testing machine are reported here. The setup is the same as that used in Chapter 3. The purpose of this experiment is to establish that the variable friction phenomenon arises from interaction of the worn surfaces, rather than from a ‘velocity effect’. The experiment was carried out (under displacement control) at an applied fretting stroke length $2\delta_{app}$ of 1.0 mm (peak-to-peak) for 400 cycles, until a significant ‘hook feature’ (i.e. an increase in tangential force during sliding) had developed in the measured hysteresis loops. The test was then stopped and restarted at an increased applied displacement stroke of 1.6 mm. If the ‘hook feature’ is a result of wear-scar interaction, then changing the imposed displacement amplitude should significantly affect the measured response.

4.2.1 Material and specimens

The material used for testing throughout this chapter is the nickel alloy Udimet 720, and was supplied by Rolls-Royce plc. This alloy is noted for its strength at high temperatures, and is used extensively in high temperature parts of aero-engines such as the turbine. It was selected for this study because of its aerospace applications (where fretting is a major issue), but also because earlier work (Chapter 3) showed that this material produces more distinct increases in friction force during the sliding phase of a fretting cycle than the titanium alloy Ti-6Al-4V (also used extensively in aero-engines, and investigated in Chapter 3). The mechanical and chemical properties of Udimet 720 were given previously in Table 3.1.

The geometry of the fretting pads and specimen used for the linear test here is the same as that used in Chapter 3 and was shown schematically in Figure 3.6. Two flat pads with rounded edges were again pressed against two opposite sides of a square-sectioned specimen with flat sides. The pre-test surface condition of the pads and specimen was a ground unidirectional surface with an areal root mean square (RMS) roughness $S_q \approx 1.94 \mu\text{m}$. Pre-test surface topography was measured using the Alicona ‘Infinitefocus’ optical surface profilometer according to the settings established in Section 3.2.3. Figure 3.6 also shows the loads acting on the pads and specimen during the test.

4.2.2 Test setup

A schematic diagram of the pads and specimen mounted in the servo-hydraulic tensile testing machine has already been given in Figure 3.10. The specimen was screwed (and clamped) into a specimen mount attached to the hydraulic actuator and oscillated at a

frequency of 1 Hz during the test. Two hydraulic pistons were then used to transmit an average normal pressure p_m of 70 MPa (via the pads) to the contact region (nominal area: $8 \times 10 \text{ mm}^2$). The tangential load ($2Q$ in this case) carried by the specimen was measured using a 125 kN capacity load cell mounted on the test machine. Specimen displacement δ was measured using a linear variable differential transformer (LVDT) mounted in the cast iron block – with its armature maintaining contact with the specimen mount. Surfaces were cleaned with acetone beforehand and testing was carried out in a ‘dry’ unlubricated condition with ambient conditions of temperature, pressure and humidity.

4.2.3 Friction results

A selection of the tangential force hysteresis loops arising from the linear fretting test are shown in Figure 4.3.

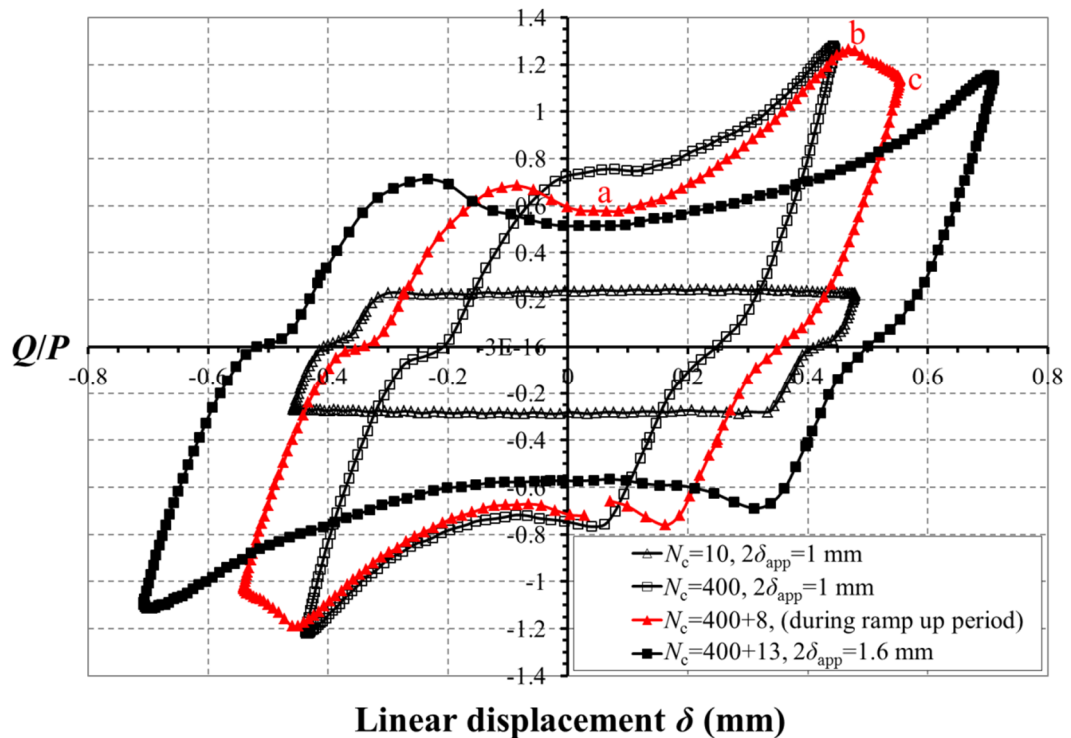


Figure 4.3: Frictional hysteresis loops derived from the linear fretting test. The test was stopped, restarted, and ramped to an increased displacement amplitude after cycle 400. The figure shows two loops recorded before the change in amplitude ($N_c = 10$ and $N_c = 400$), and two loops recorded after the change in amplitude ($N_c = 400+8$ and $N_c = 400+13$).

It can be seen that after ten cycles of testing at an applied displacement stroke, $2\delta_{\text{app}} = 1$ mm, no increase in friction force is observed during sliding (if a ‘velocity effect’ were driving the increasing friction behaviour, we would have expected to see the behaviour in all cycles). However, after 400 cycles a significant ‘hook feature’ has developed. The loop (in red) recorded eight cycles after the test was restarted at the higher displacement amplitude ($N_c = 400+8$) is particularly interesting as it shows an increase in friction force only up to the point where the previous maximum displacement occurred, followed by a decrease until the new maximum displacement is reached. This appears to indicate that the friction increases as the ‘old’ wear scars interfere (a–b) and then drops off (b–c) as fresh unworn surface is encountered and the topography of the previous wear scars no longer conform or interlock. This is further evidence that the sliding friction response is dictated by interaction of the worn surfaces rather than by a velocity effect. The reason for the slight mismatch between the start and end point of this loop is that the hydraulic actuator was still ramping up (from zero) to the new applied displacement stroke, $2\delta_{\text{app}} = 1.6$ mm, when this loop was recorded. Indeed, by the time loops at the new applied stroke length were recorded, this transition feature had disappeared from the hysteresis loop and a new conventional ‘hook feature’ had developed – as shown by the loop recorded 13 cycles after the restart ($N_c = 400+13$). The unusual shape of the loop recorded eight cycles after the change of amplitude and its rapid evolution into a conventional type loop, together with the fact that no ‘hook feature’ is observed in the initial cycles, presents strong evidence that the variation in friction force during sliding is indeed a wear-scar interaction effect, and not a velocity effect. A small amount of backlash is evident from the loops, but this does not affect the sliding response.

4.3 Torsional Fretting Tests

Although the experiment discussed above suggests that the friction variation during sliding results from wear-scar interactions, it provides no information on whether the interaction is taking place between the macroscopic ends of the wear scars (Figure 4.2a), or whether it occurs throughout the contact region at local corresponding surface peaks and troughs on the pad and specimen wear scars (Figure 4.2b). To probe this question further, it is necessary to compare hysteresis loops from a fretting contact with no macroscopic ends in the direction perpendicular to sliding with loops from a similar test where the contact patch has the usual edges perpendicular to the sliding direction. A series of fretting experiments were carried out to implement this idea using a torsional configuration. Worn surface topography was then examined after the tests with the aim of identifying corresponding features on the wear scars by both qualitative and quantitative means.

4.3.1 Material and specimens

In order for a contact region to have no macroscopic ends, it must take a circular form; therefore, a cylindrical specimen with annular or *continuous ring* shaped ends was chosen for one of the contacting bodies. This was compared with a specimen with *segmented ring* shaped ends (each containing three ring sectors). Each of these specimens was tested in contact with a flat counterbody or ‘pad’ under reciprocating torsion. The pads and specimens for this ‘ring-on-flat’ design are shown in Figure 4.4 together with dimensioned drawings of the two contact regions to be compared (the pads are shown attached to the pad holders).

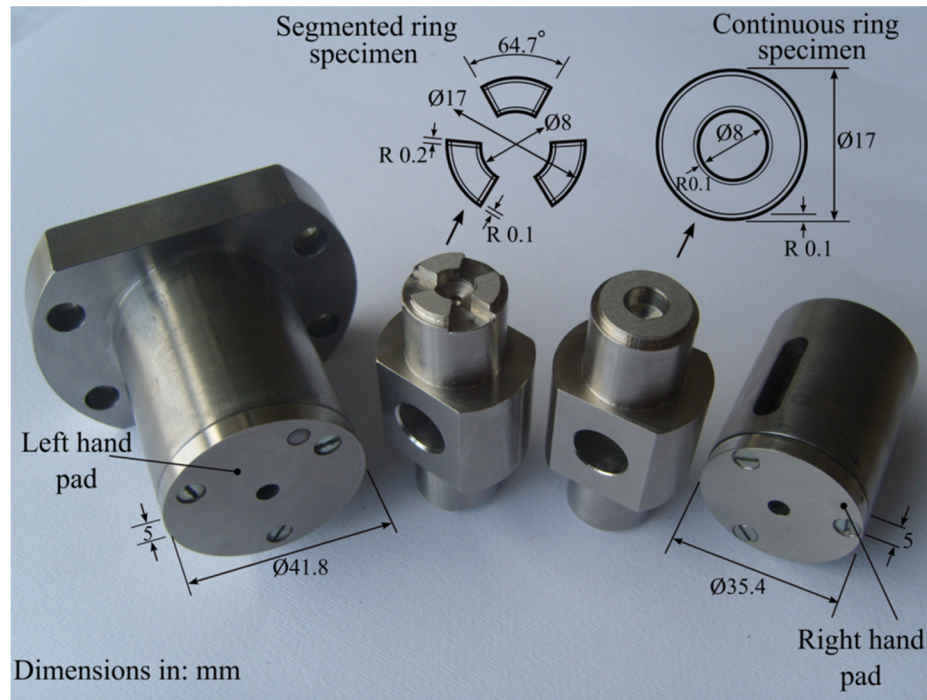


Figure 4.4: Pads and specimens for the torsional fretting tests.

Both the pads and specimens were also made from the nickel alloy Udimet 720 (see Section 4.2.1). These pads and specimens were specially designed and manufactured for the study outlined in this chapter. The pad and specimen surfaces were subject to unidirectional grinding, and the resulting pre-test areal RMS surface roughness values S_q are reported in Table 4.1 (together with post-test roughness values) as measured by the Alicona surface profilometer (see Section 3.2.1).

Table 4.1: Pre-test and post-test primary areal root mean square (RMS) roughness S_q in μm . These values are each based on the average of eight different scans of the pad and specimen surfaces: each scan being over a $2 \times 2 \text{ mm}^2$ scan area. p_m is the average normal contact pressure in each test.

	Segmented ring		Continuous ring	
	Test 1: $p_m = 30 \text{ MPa}$	Test 2: $p_m = 70 \text{ MPa}$	Test 3: $p_m = 30 \text{ MPa}$	Test 4: $p_m = 70 \text{ MPa}$
Pre-test	1.02	1.04	1.30	1.11
Post-test	16.06	23.06	22.75	29.12

4.3.2 Test setup

A torsional fretting fixture was designed and built to enable torsional (or rotational) fretting of the ring contacts described above. This fixture was designed so that it could be easily fitted to conventional in-line testing machines. For the present work, the rig was attached to the Dartec servo-hydraulic tensile testing machine which was used in Section 4.2. A sectional plan view and side elevation of the assembled fixture is shown in Figure 4.5.

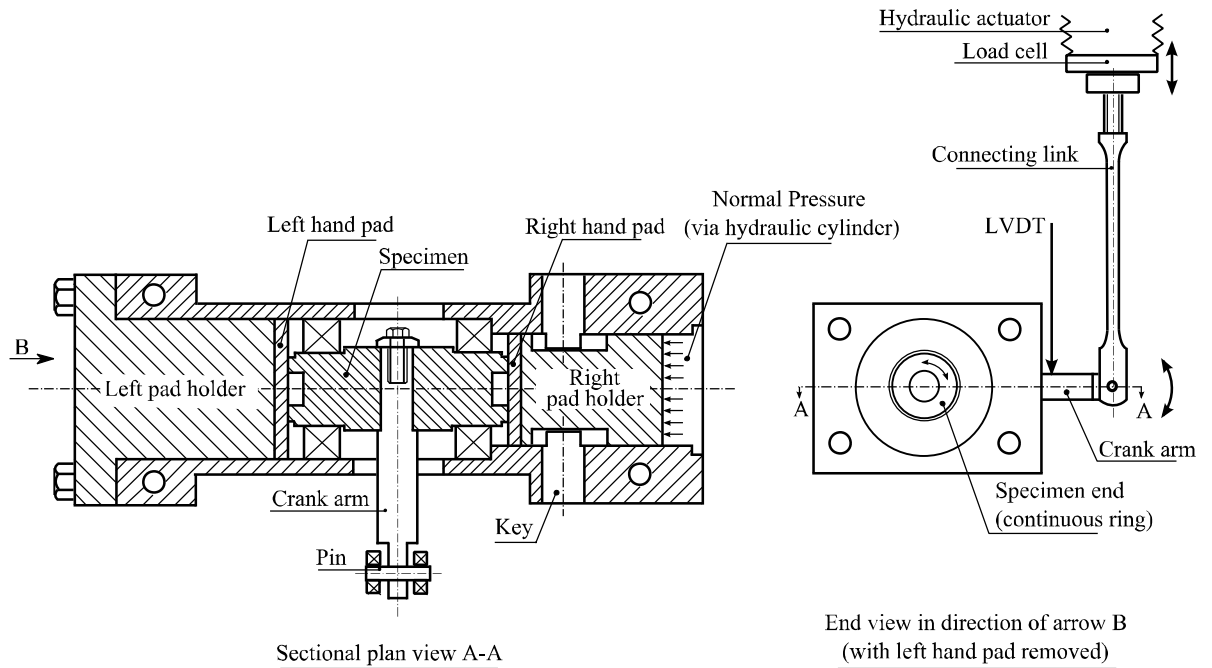


Figure 4.5: Sectional plan-view and end-view of the torsional fretting fixture.

The specimen is supported on two bearings, and linear motion from the hydraulic actuator is transferred to rotational oscillating motion of the specimen via a connecting link and a crank arm. The two pads are rigidly fixed to cylindrical pad holders by both screws and dowels. The left hand pad holder is fixed to the housing by screws, while the

right hand pad holder is free to move only axially in two key-ways to allow transmission of normal pressure applied by hydraulic piston to the two contact regions. Accurate alignment between the pads and specimen during testing is crucial to the investigation; hence, the rig was bored from one solid block of material and accommodates two symmetrically located fretting contacts, instead of one, for mechanical balance (if only one contact were present, the specimen might tend to tilt out of alignment).

During a test, the force transmitted through the connecting link is measured by the machine load cell, and a measure of linear displacement of the crank arm is given by an LVDT. The analogue signals from the load cell and LVDT are processed in the same way as was described previously in Chapter 3 (Section 3.3.3). However, the resulting force must be converted to a torque T by multiplication by the crank arm length, and LVDT displacements must be converted to angular displacement δ_θ using the perpendicular distance between the LVDT and specimen axes. A photograph of the rig mounted on the Dartec testing machine, with the housing bolted onto the lower platen and the connecting rod screwed (and clamped) to the actuator, is shown in Figure 4.6.

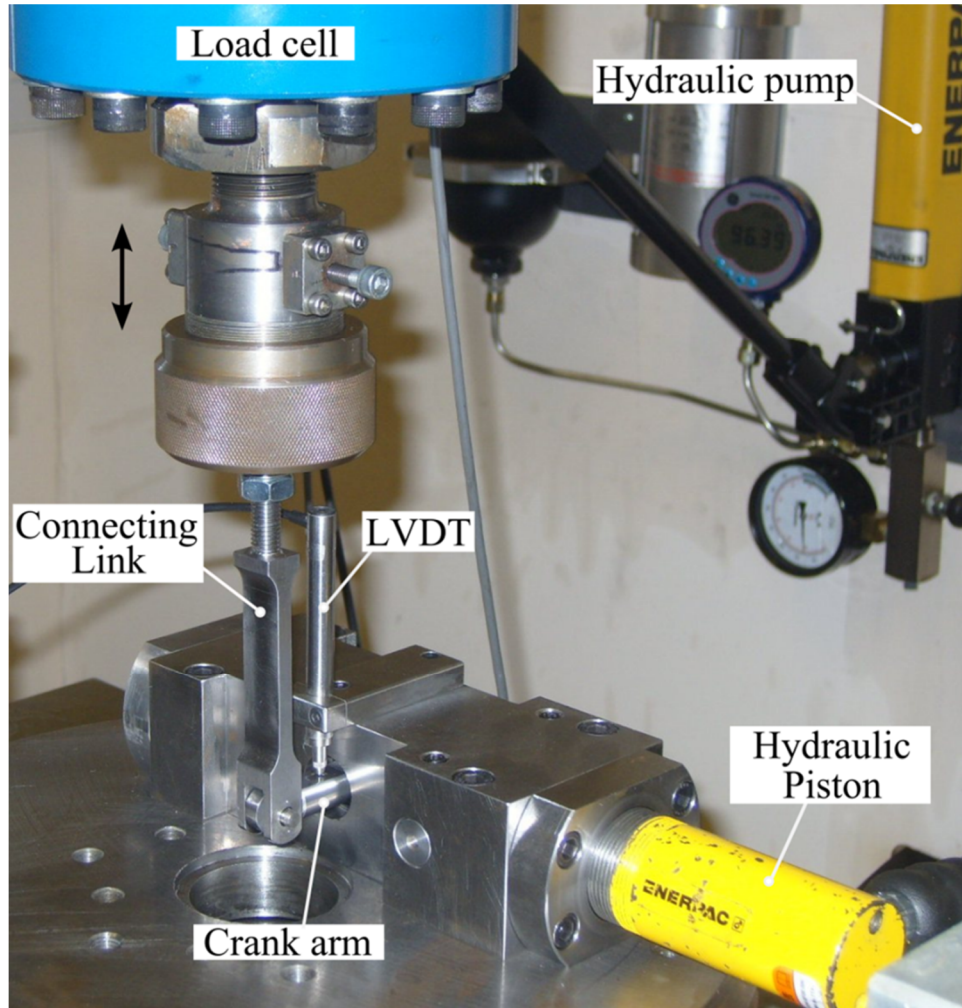


Figure 4.6: Torsional fretting fixture assembled on the servo-hydraulic tensile testing machine.

A total of four experiments were undertaken with the torsional fretting rig: the continuous ring and the segmented ring contact were each tested at mean normal contact pressures of $p_m = 30$ MPa and $p_m = 70$ MPa. An applied angular fretting oscillation stroke of $2\delta_{\theta,app} = 8.82^\circ$ (peak-to-peak) was chosen to produce a displacement oscillation amplitude of 1 mm (peak-to-peak) on the median circle of the ring, and in each case, tests were carried out up to 5030 fretting cycles at a frequency of 0.75 Hz. Surfaces were again cleaned with acetone beforehand and testing was carried out in a ‘dry’ unlubricated condition with ambient conditions of temperature, pressure and humidity.

4.3.3 Friction results

Hysteresis loops showing the quantity q_m/p_m against angular displacement, where q_m and p_m are the mean tangential and normal shear tractions respectively, are shown in Figure 4.7 for tests carried out at $p_m = 30$ MPa, and in Figure 4.8 for tests carried out at $p_m = 70$ MPa. It should be noted that the quotient q_m/p_m corresponds to the friction coefficient during the sliding phases of the hysteresis loops. The average tangential traction q_m for a single continuous ring contact pair is calculated from:

$$q_m = \frac{T}{4\pi \int_{r_i}^{r_o} r^2 dr}, \quad (4.1)$$

and for the segmented ring contact by:

$$q_m = \frac{T}{4\pi\zeta \int_{r_i}^{r_o} r^2 dr}, \quad (4.2)$$

where, T is the total torque arising from both contact pairs, ζ ($=0.52$) is the fraction of the full ring which is occupied by the three flats on the segmented ring, and r_i , r_o , and r are the inside, outside, and variable radii of the ring geometry, respectively. Both Figure 4.7 and Figure 4.8 show hysteresis loops which are each an average representation of 30 test cycles.

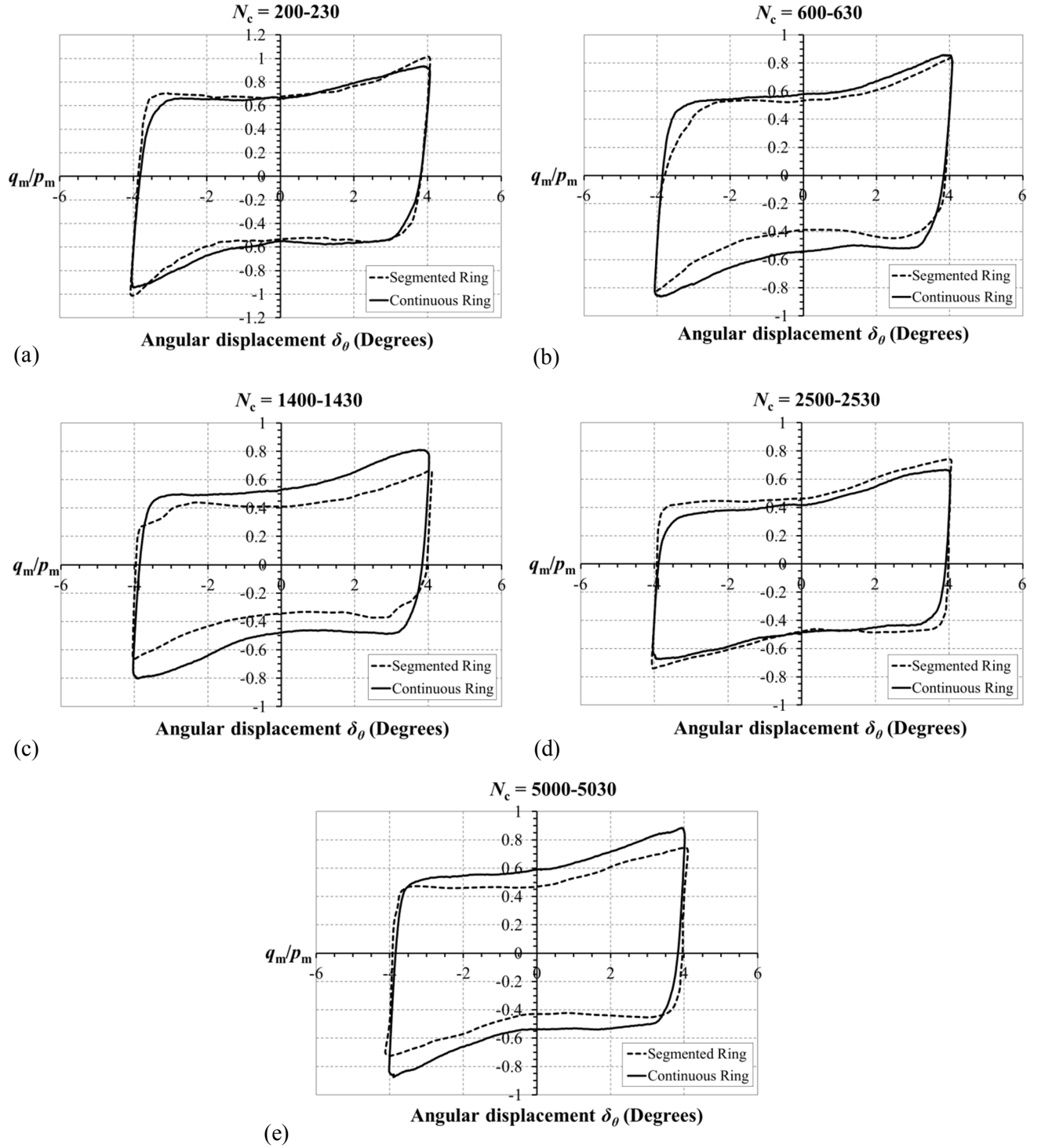


Figure 4.7: Frictional hysteresis loops derived from the torsional fretting tests at an average normal pressure $p_m = 30$ MPa for cycles: (a) $N_c = 200-230$, (b) $N_c = 600-630$, (c) $N_c = 1400-1430$, (d) $N_c = 2500-2530$ and (e) $N_c = 5000-5030$.

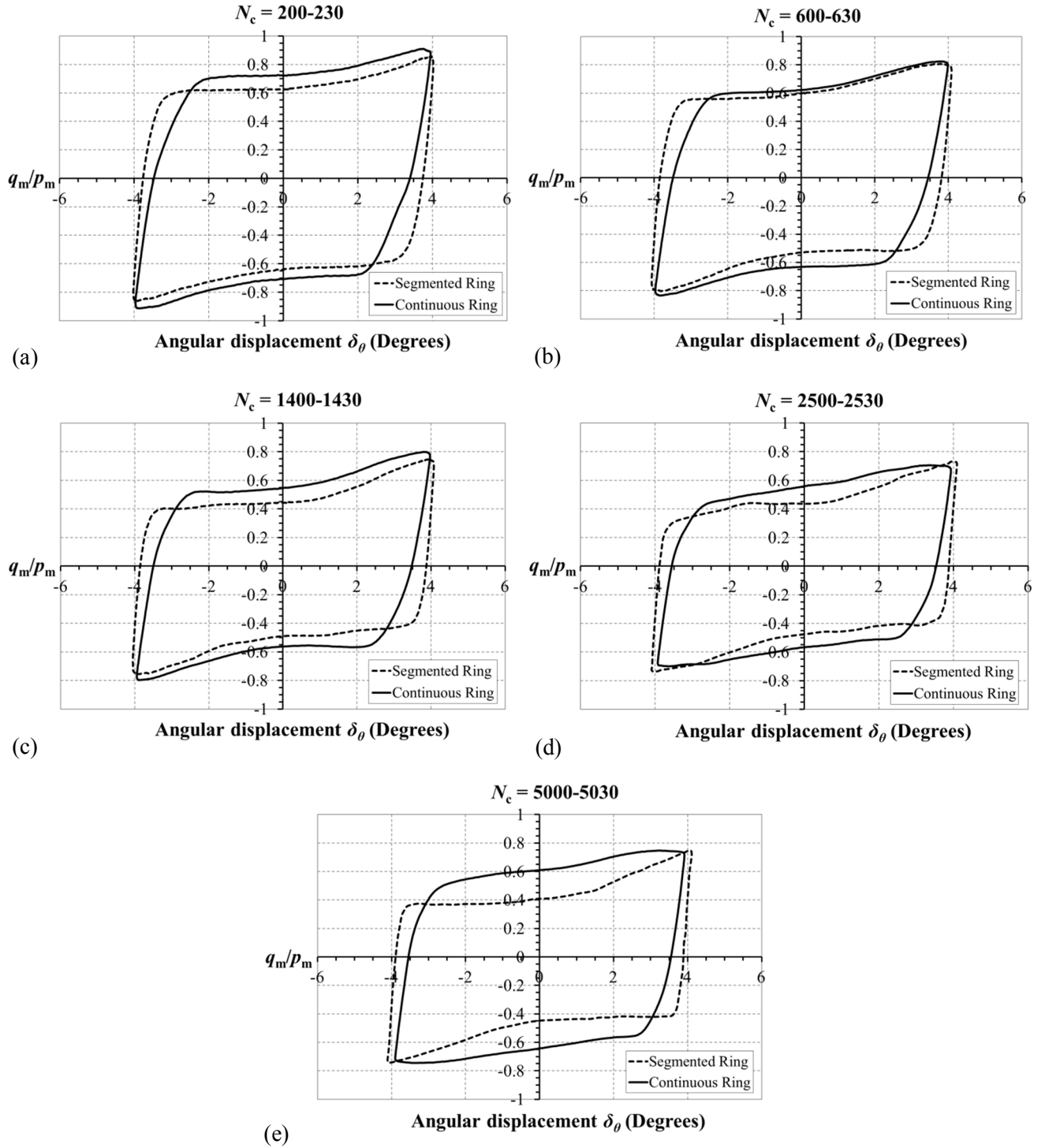


Figure 4.8: Frictional hysteresis loops derived from the torsional fretting tests at an average normal pressure $p_m = 70$ MPa for cycles: (a) $N_c = 200-230$, (b) $N_c = 600-630$, (c) $N_c = 1400-1430$, (d) $N_c = 2500-2530$ and (e) $N_c = 5000-5030$.

These average loops were each determined from a set of 30 test cycles by a moving average smoothing technique implemented in MATLAB. This procedure was carried out

since individual loops showed a small degree of measurement noise, and does not significantly change the results since the variation in the measured loops occurring over just 30 fretting cycles is small (once the initial ‘run-in’ cycles are complete). Figure 4.7 and Figure 4.8, therefore, show representative average loops for cycles: 200-230, 600-630, 1400-1430, 2500-2530 and 5000-5030. Examination of these loops shows that the friction increases during sliding for *both* the continuous and segmented ring experiments. Since a macroscopic *end effect* cannot be responsible for the increasing friction in the case of the continuous ring contact, this suggests that corresponding pairs of local interlocking wear-scar features (peaks and troughs) on the pad and specimen must be interacting throughout the contact region to produce this effect. However, it is possible that macroscopic wear-scar edges, when present, (as with the segmented ring) also contribute to the ‘hook’ effect.

Figure 4.9 shows the evolution of the maximum, average and minimum sliding friction coefficients for all four torsional tests. In determining these friction coefficients from the hysteresis loops a MATLAB script was used, and the values reported are an average which takes into account sliding in both directions for a given test cycle. These graphs show that the difference between maximum and minimum friction coefficients (denoted as the ‘friction variation range’ in Figure 4.1) can be as much as 0.4 (see Figure 4.9c). Although the overall *average* range of variation of the friction coefficient for the $p_m = 70$ MPa tests was slightly larger for the segmented ring (0.34) than for the continuous ring (0.23), the two $p_m = 30$ MPa tests showed very similar ranges (0.31 for the segmented ring and 0.32 for the continuous ring). The above results suggest that the *primary* wear-scar interaction effect is at the scale of peaks and troughs distributed within the contact region, rather than at the macroscopic contact edges – at least for the stroke lengths, pressures, material and contact arrangements investigated in this work. Figure 4.10

compares the evolution of the average sliding friction coefficient for all four tests. It can be seen (for this nickel alloy at least) that the friction coefficient tends to rise sharply to a maximum after the initial ‘run in’ cycles and then gently reduces somewhat before increasing slightly again. It is also noticed that friction coefficients for both the continuous and segmented rings are similar and no significant differences in friction coefficient with normal pressure can be discerned between the results for the $p_m = 30$ and $p_m = 70$ MPa tests (which is consistent with the pressure independence part of the Amontons-Coulomb friction model).

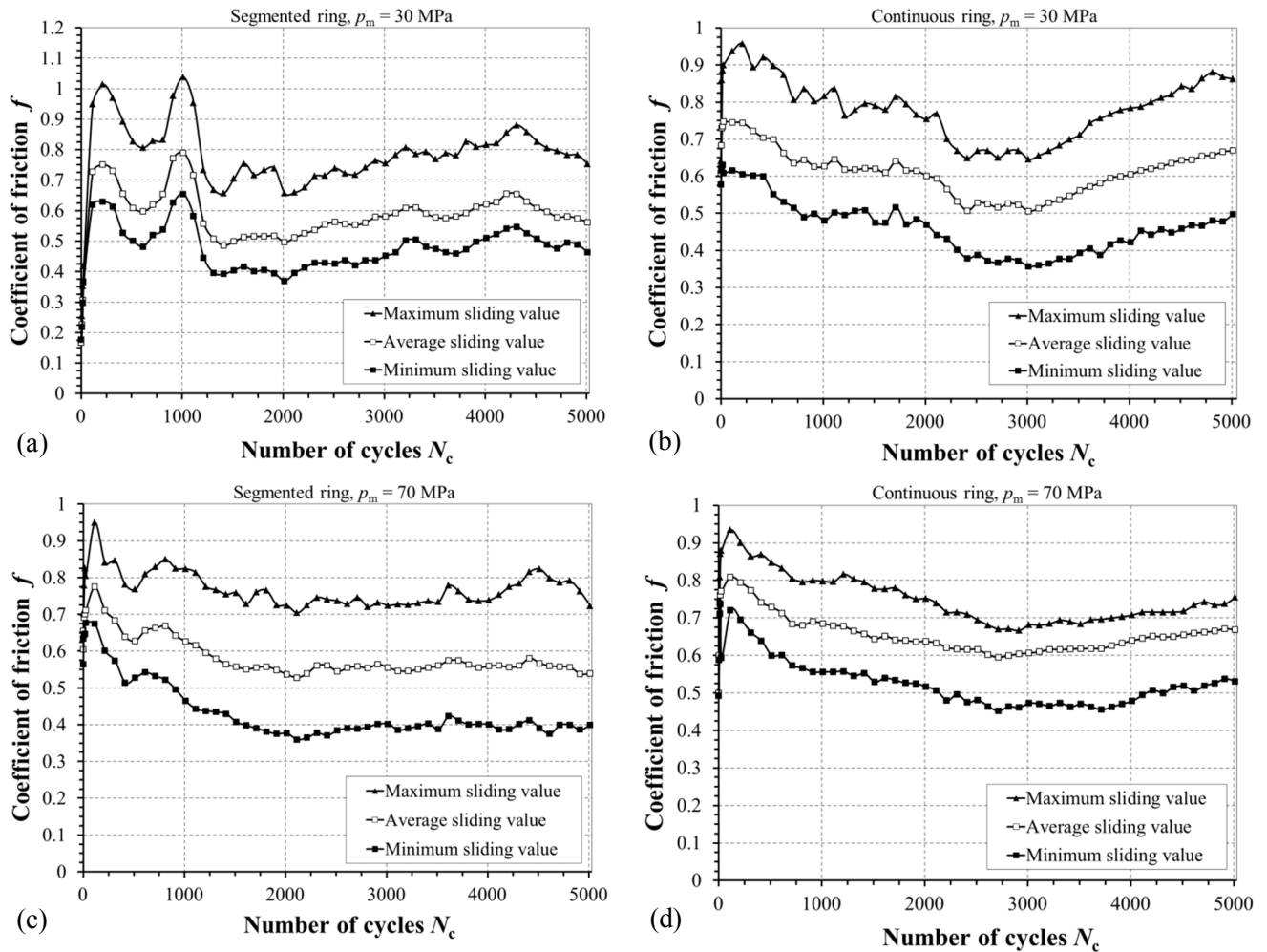


Figure 4.9: Evolution of the maximum, average, and minimum sliding friction coefficient with number of test cycles for: (a) segmented ring contact at $p_m = 30$ MPa, (b) continuous ring contact at $p_m = 30$ MPa, (c) segmented ring contact at $p_m = 70$ MPa and (d) continuous ring at $p_m = 70$ MPa.

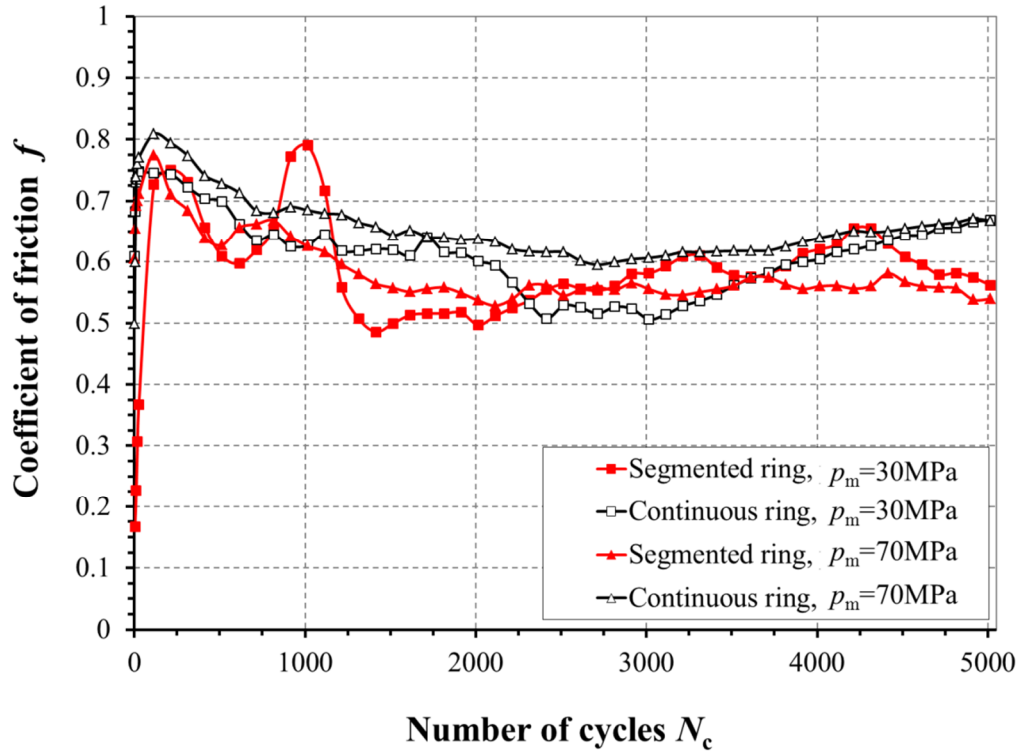


Figure 4.10: Comparison of the evolution of the average sliding friction coefficient for all four torsional fretting tests carried out.

4.3.4 Wear-scar analysis

The results obtained from the torsional fretting tests suggest that the friction variation during gross-slip is a result of wear-scar interaction at a local level within the contact. A detailed examination of the worn surfaces was therefore carried out to determine if such distributed local features exist which could cause such interaction. At the end of each torsional fretting test (after 5030 cycles), the pads and specimen were carefully removed from the test rig, and the full wear-scar surface topography was then scanned using the Alicona optical surface profilometer (see Section 3.2). The worn surfaces were not cleaned before imaging in order to preserve the nature of the surfaces which existed during the test itself. Figure 4.11 and Figure 4.12 show the worn surface topography of

both ring configurations (continuous and segmented) for the tests carried out at normal pressures of 30 and 70 MPa, respectively. In each case, the specimen wear scar and the corresponding pad wear scar are shown such that a 180° rotation of the specimen wear scar about the axis shown would place the pair of wear scars into the contact configuration which existed during the test. This method of presentation allows identification of corresponding features on the specimen and pad wear scars which may have been interacting during fretting.

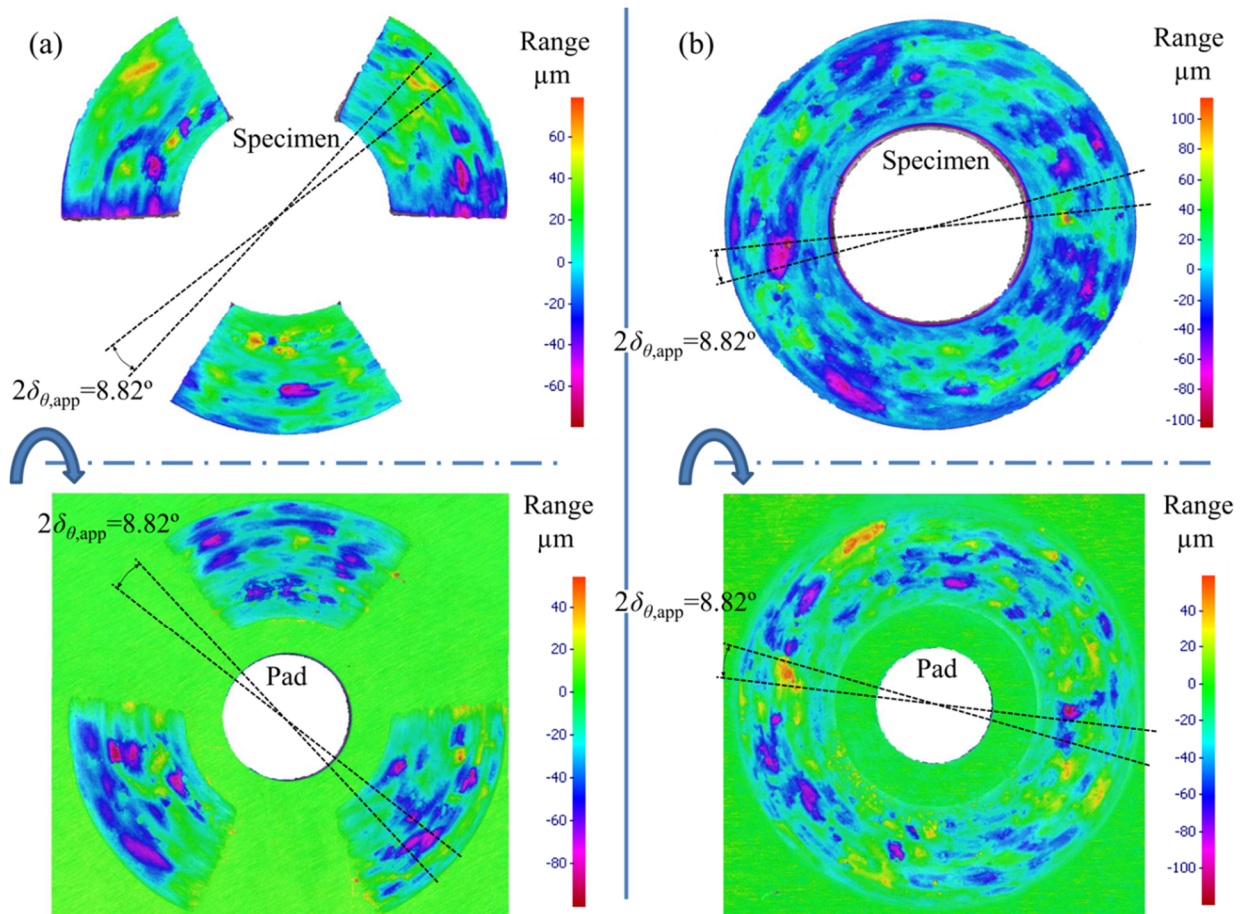


Figure 4.11: Pad and specimen wear-scar surface topography after 5030 torsional test cycles at an average normal pressure $p_m = 30$ MPa for: (a) the segmented ring and (b) the continuous ring. (Note: these wear scars are taken from the left hand contact pair – see Figure 4.5)

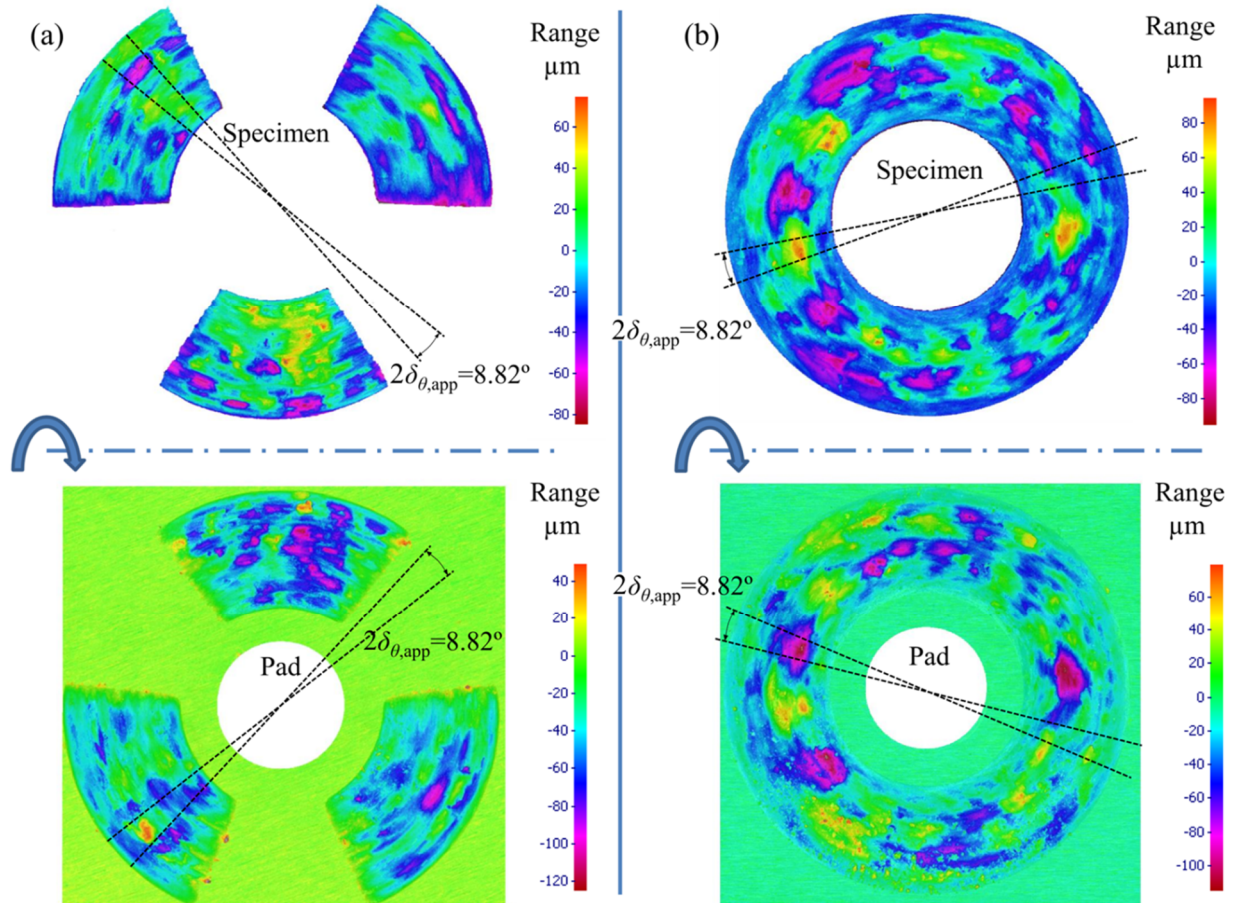


Figure 4.12: Pad and specimen wear-scar surface topography after 5030 torsional test cycles at an average normal pressure $p_m = 70$ MPa for: (a) the segmented ring and (b) the continuous ring. (Note: these wear scars are taken from the left hand contact pair – see Figure 4.5)

Both Figure 4.11 and Figure 4.12 show the existence of many corresponding peaks and troughs distributed throughout the surfaces of both the segmented and continuous ring wear scars: this is consistent with the finding above that the friction variation during gross slip also occurs when no macroscopic terminal wear scar ends exist. In each figure, a pair of intersecting dotted lines shows the applied fretting stroke angle $2\delta_{\theta,app}$, and also highlights corresponding features on the specimen and pad wear scars. It can be seen that the size of these features is very similar to the size of the applied fretting stroke (see those features captured between the dotted lines in Figure 4.11 and Figure 4.12). Although the worn surface profiles show the existence of such distributed local features, a quantitative method is required to estimate the dominant feature size. A cross-correlation and auto-

correlation approach was therefore developed, for the continuous ring configuration, using a MATLAB script. The surface data obtained from the Alicona surface profilometer, in the form of a rectangular grid, were converted to a polar grid of $M \times N$ surface heights $z(r_j, \theta_k)$ by interpolation (where $z = 0$ is the surface mean plane). The angular and radial separation of grid points was uniform at 0.19° and $15 \mu\text{m}$, respectively and the rings were cropped to an inner and outer radius of 4.5 and 8 mm, respectively, to eliminate difficulties caused by edge effects. A cross-correlation function was then evaluated for a number of different relative orientations of the two surfaces (defined by a relative angle $\Delta\theta$) until results for a full 360° rotation (in increments of 0.19°) were determined. The normalised cross-correlation function (CCF) for the corresponding specimen and pad wear-scar pair was determined as:

$$\text{CCF}(\Delta\theta) = \frac{1}{S_{q,\text{pad}}S_{q,\text{spec}}} \frac{1}{MN} \sum_{k=1}^N \sum_{j=1}^M z_{\text{pad}}(r_j, \theta_k) z_{\text{spec}}(r_j, \theta_k + \Delta\theta), \quad (4.3)$$

where $S_{q,\text{pad}}$ and $S_{q,\text{spec}}$ are the RMS roughness values (standard deviations of heights) of the pad and specimen surfaces, respectively. Auto-correlation was also carried out for the individual pad and specimen worn surfaces according to the normalised auto-correlation function (ACF):

$$\text{ACF}(\Delta\theta) = \frac{1}{S_q^2} \frac{1}{MN} \sum_{k=1}^N \sum_{j=1}^M z(r_j, \theta_k) z(r_j, \theta_k + \Delta\theta), \quad (4.4)$$

where S_q is the RMS roughness of the single worn surface (pad or specimen) for which the ACF is being calculated. Figure 4.13 and Figure 4.14 show plots of the cross-correlation and auto-correlation functions against $\Delta\theta$ ($-180^\circ > \Delta\theta < 180^\circ$) for the same continuous ring wear scars as are shown in Figure 4.11b and Figure 4.12b, respectively.

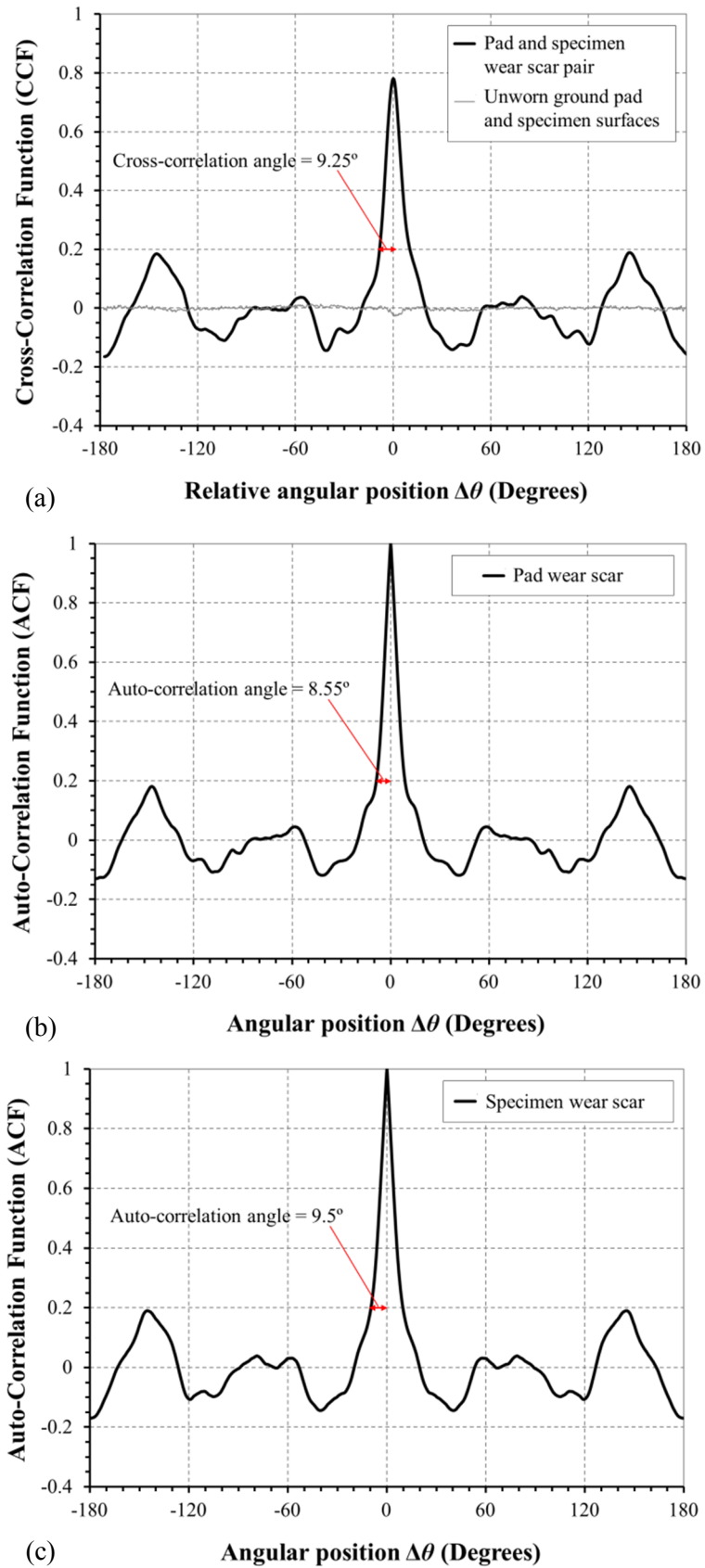


Figure 4.13: Correlation functions against angular position for the continuous ring wear scars tested at $p_m = 30$ MPa: (a) cross-correlation of the pad and specimen wear-scar pair (also showing result for the unworn surfaces), (b) auto-correlation of the pad wear scar and (c) auto-correlation of the specimen wear scar.

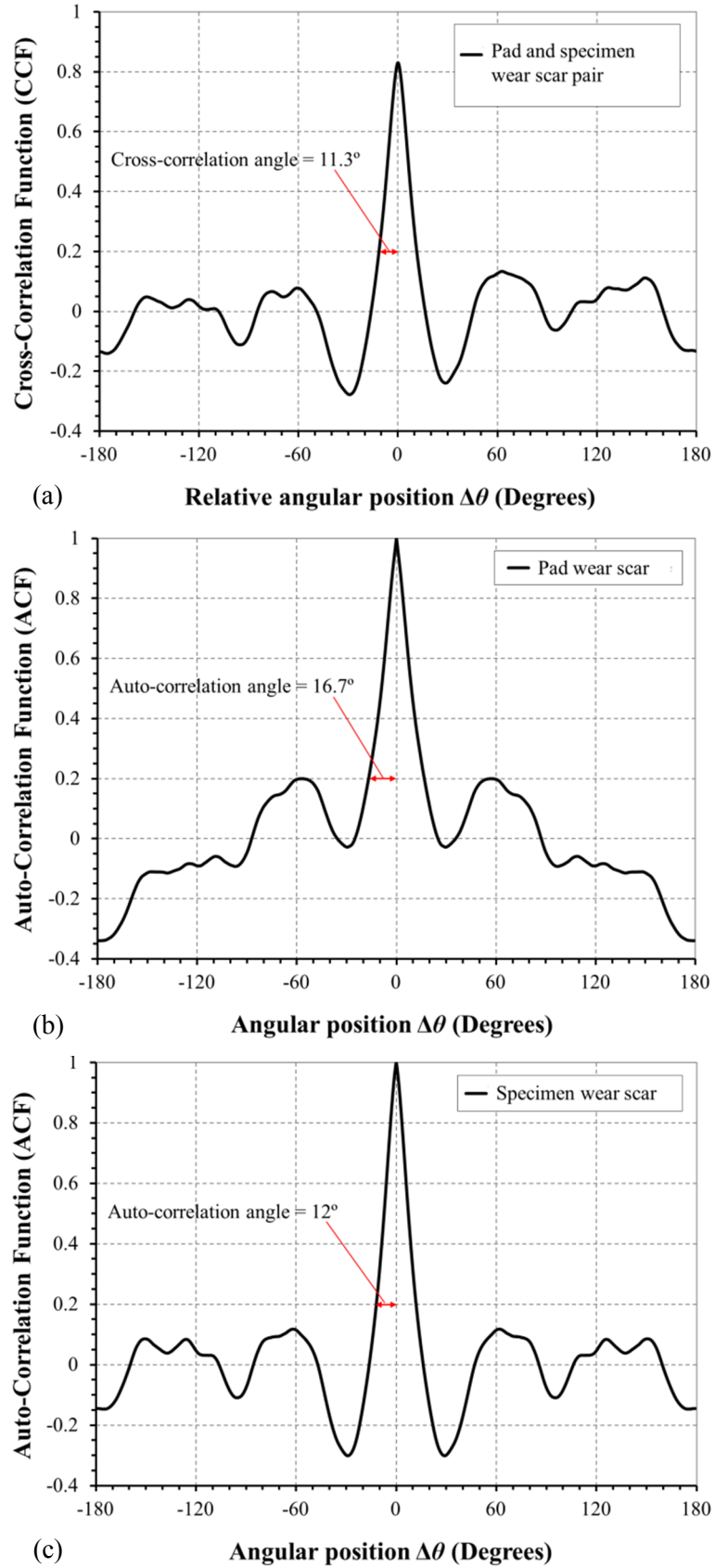


Figure 4.14: Correlation functions against angular position for the continuous ring wear scars tested at $p_m = 70$ MPa: (a) cross-correlation of the pad and specimen wear-scar pair, (b) auto-correlation of the pad wear scar and (c) auto-correlation of the specimen wear scar.

The plots show strong cross-correlation of the surfaces in a clearly defined region around the 0° position which corresponds to the orientation in the experiment. At angles of rotation which are much greater than the fretting amplitude, there is very little correlation between the surfaces as expected (since the features are randomly distributed). An estimate of the dominant feature size can be gained by determining half of the angle for which the surfaces remain reasonably well correlated: referred to here as the *cross-correlation* (or auto-correlation) *angle*. The definition of the onset of correlation is somewhat arbitrary, but the point at which the correlation function has decayed to 0.2 is often taken as the point at which the surfaces become uncorrelated [122]. This value is therefore used to define the cross-correlation (or auto-correlation) angle. The corresponding cross-correlation and auto-correlation angles are marked on Figure 4.13 and Figure 4.14.

Bearing in mind that the applied fretting stroke angle was 8.82° during the tests, the size of the correlation angles highlighted in Figure 4.13 and Figure 4.14 show that the wear-scar features are similar in size to the applied fretting stroke angle for the $p_m = 30$ MPa case and somewhat larger for the $p_m = 70$ MPa test. In Figure 4.13a, the CCF for the two surfaces in the pre-test ‘as manufactured’ state is also shown. This confirms, as expected, that there were no significant correlating features prior to testing. The maximum cross-correlation value of the wear scars, which is close to 0.8 for both the 30 MPa and 70 MPa tests (Figure 4.13a and Figure 4.14a), is of interest since it indicates quite a high level of registration or conformity between the two worn surfaces which began the test as two uncorrelated surfaces (a value of unity would indicate perfect conformity as with the auto-correlation results). Finally, referring back to Table 4.1, we note that the RMS areal surface roughness of the contact pairs increased significantly during the tests (on average

from 1.1 to 22.7 μm : this is consistent with the formation of the local wear-scar features discussed above.

4.4 Wear-scar interaction model

In order to help explain how the interaction of corresponding local interlocking surface bumps and depressions (due to wear) in fretting tests produces a variation in the measured friction force during gross slip, a simple one degree-of-freedom model was developed. The model, shown schematically in Figure 4.15, is based on a rigid round-ended body (representing a surface peak) undergoing oscillatory interaction with a groove in a rigid pad (representing a depression worn into the mating surface).

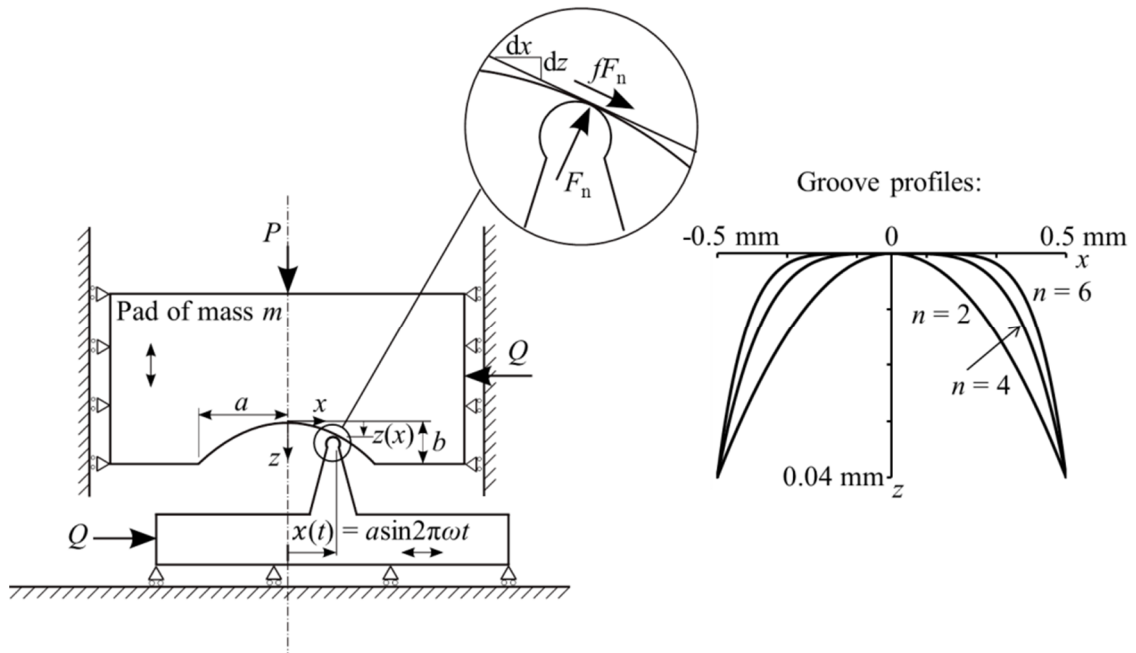


Figure 4.15: Schematic diagram illustrating the proposed wear-scar interaction model.

The oscillating pad is assumed to have effective mass m , and is free to move in the direction perpendicular to the imposed fretting motion. The position of the moving body (relative to the pad) is given by $x(t) = a \sin 2\pi\omega t$, where a the groove half-width, is also the amplitude of the imposed motion and ω is the angular frequency. The groove in the mating surface has a profile, $z(x)$, which takes the form of a polynomial:

$$z = b \left(\frac{x}{a} \right)^n, \quad (4.5)$$

where b is the groove depth. Analysis was carried out with three different groove geometries: $n = 2$, $n = 4$, and $n = 6$ (the $n = 4$ and $n = 6$ grooves have progressively steeper edges than $n = 2$). The normal force developed at the contact between the two surfaces is defined as F_n , and this produces a frictional force fF_n tangential to the groove surface. A direct load P is applied to the pad in the direction orthogonal to the fretting direction, and Q is the resulting tangential force, which is required to prevent motion of the pad in the x -direction as the moving body slides along the groove. The equation of motion in the vertical direction (for the pad) is:

$$fF_n \frac{\frac{dz}{dx}}{\sqrt{1 + \left(\frac{dz}{dx}\right)^2}} \frac{|\dot{x}|}{\dot{x}} - F_n \frac{1}{\sqrt{1 + \left(\frac{dz}{dx}\right)^2}} + P = m\ddot{z}. \quad (4.6)$$

This equation may be solved (for a given form of groove) to find the local normal contact force, F_n . Similarly, the equation of motion of the pad in the tangential direction is:

$$Q = F_n \frac{\frac{dz}{dx}}{\sqrt{1 + \left(\frac{dz}{dx}\right)^2}} + fF_n \frac{1}{\sqrt{1 + \left(\frac{dz}{dx}\right)^2}} \frac{|\dot{x}|}{\dot{x}}, \quad (4.7)$$

which may be solved² for Q . For the purposes of the present analysis, the calculation was carried out without including the mass, m , (i.e. a quasi-static analysis), since dynamic effects are unlikely to be significant at the test frequencies employed. The aspect ratio of the groove ($2a:b = 25:1$) was chosen to correspond to the features measured in the analysis of the ring wear scars in Section 4.3.4. A friction coefficient, $f = 0.6$, was chosen based on the friction results of Section 4.3.3. The resulting predicted local hysteresis loops, as determined by Equations 4.6 and 4.7, of Q/P against normalised sliding distance x/a are shown in Figure 4.16(a, b, and c) for groove profiles of: $n = 2$, $n = 4$, and $n = 6$, respectively.

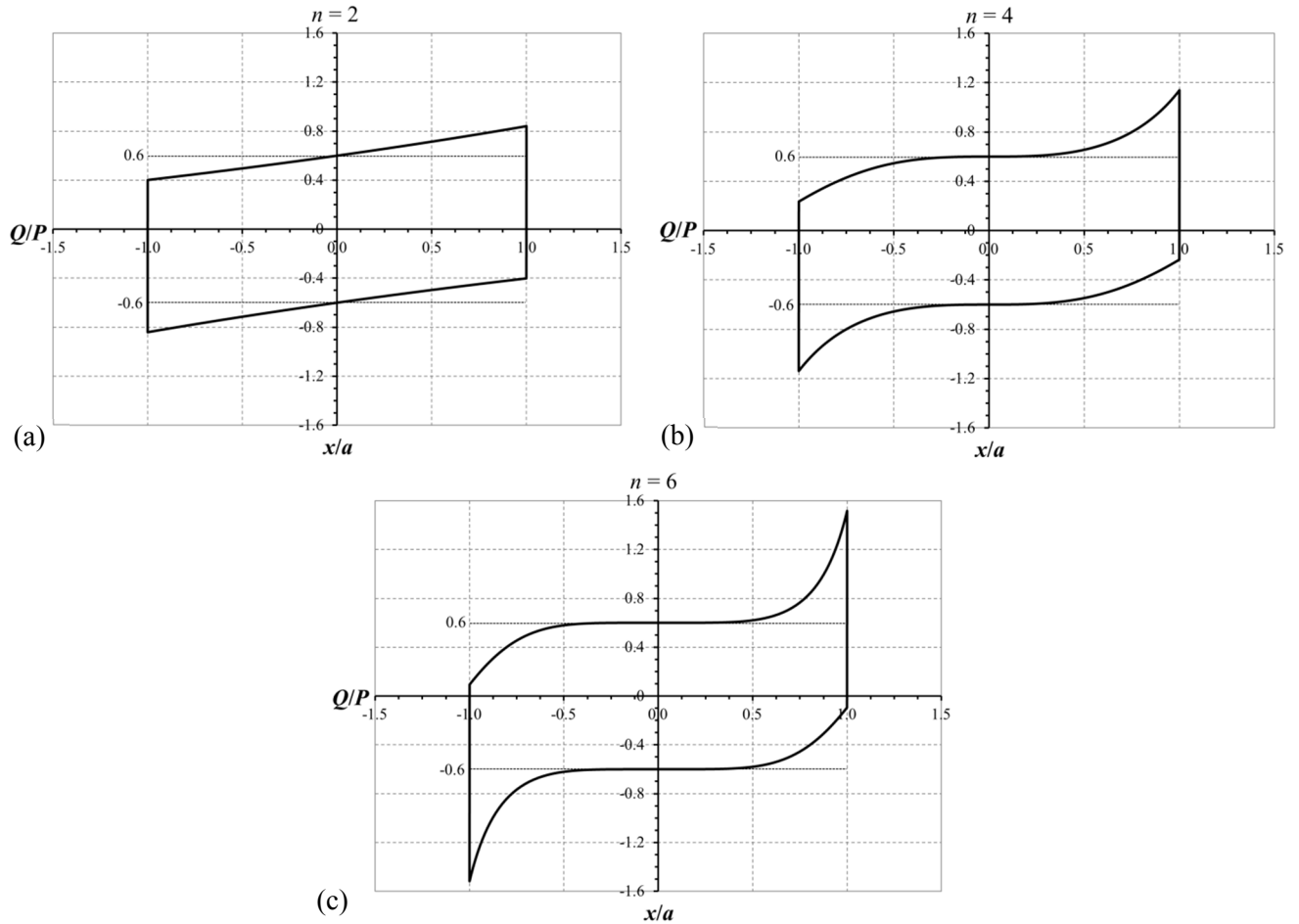


Figure 4.16: Normalised tangential force hysteresis loops derived from the wear-scar interaction model for a worn surface groove defined by polynomials of order: (a) $n = 2$, (b) $n = 4$ and (c) $n = 6$. The corresponding Coulomb loops (with $Q/P = \pm 0.6$) are also indicated.

² There is no dynamic term in Equation 4.7 as the pad is constrained to have no motion in the tangential direction (see Figure 4.15).

All three profile shapes produce the continuous increase in friction force during slip which is commonly observed in real fretting test conditions. For $n = 2$, the increase is linear, and for $n = 4$ and $n = 6$, the increase becomes more pronounced at the beginning and end of sliding (because of the groove steepness at the ends). Therefore, the actual shape of the tangential force variation during sliding is likely to be affected by the geometry of the interacting worn surface features, and different forms of variation have indeed been observed in the literature: sometimes a linear increase is observed [22, 125], and in other work a more curved variation is noted [23, 24].

This idealised model does not precisely capture what happens at a real peak-trough interaction, but it does show how such an interaction can produce the type of behaviour which is commonly measured in real test conditions. The underlying reason for the continuous increase in tangential force Q during sliding (based on the present model) is that, when resolved in the (global) tangential direction, the local normal force F_n acts in the opposite direction to the friction force fF_n over the first half of the sliding phase, but in the same direction in the second half.

As a result of how the model is formulated, the energy dissipated in a single cycle will be somewhat greater than the energy dissipated by a corresponding ‘Coulomb hysteresis loop’. For the parameters used here, this difference is small, and is barely apparent in Figure 4.16 where the area of the predicted loops adding to the Coulomb friction loop (a rectangle with $Q/P = \pm 0.6$) is slightly greater than the area subtracting from the Coulomb loop. In the case of the experiments, it is not easy to determine whether the friction increase during sliding causes a higher dissipation of energy than the corresponding Coulomb prediction as it is difficult to determine the appropriate Coulomb friction coefficient to use; however, some level of increase in energy dissipation is likely.

The model outlined here is somewhat similar to the Tabor model [126] which attempted to model friction by a narrow slider passing over a single rigid triangular shaped asperity with a local friction coefficient imposed at the interface. The Tabor model was later adapted to the rough surface situation by Ford [127] using a Greenwood-Williamson approach. The present model adapts the Tabor model to describe contact between a slider and a local surface depression described by an appropriate polynomial curve. The model outlined here can also accommodate dynamic effects, and is set up for reciprocating sliding in order to explain how worn surface features cause the type of frictional behaviour which is frequently observed. Despite the simplicity of the model, it is possible to produce hysteresis loops which are a good qualitative match to those which are commonly measured in experiments. The guidance of Prof. Andrew Olver is acknowledged in helping develop the model.

4.5 Conclusions

This chapter has described an experimental programme aimed at investigating the significant variation in friction force which is often observed during the gross-slip phase of individual fretting test cycles. A linear fretting test, where the applied fretting stroke length was abruptly increased during the test, indicated that the phenomenon is a result of wear-scar interaction rather than a ‘velocity effect’. The question of whether this wear scar interaction results from interaction at the macroscopic ends of the worn surfaces, or whether it may also be occurring at local interlocking peaks and troughs distributed throughout the contact area was then investigated by a series of torsional fretting tests. A new rig, designed to allow for the adaption of conventional in-line test machines for torsional fretting testing, was constructed and shown to work effectively. These tests

revealed that the friction also increases during the gross-slip phase of fretting cycles even when no macroscopic wear-scar ends are present (i.e. for the continuous ring configuration). This suggests that local corresponding peaks and troughs on the two worn surfaces involved in a fretting pair must be forming throughout the contact, and causing this type of frictional behaviour. Analysis of the post-test worn surfaces has clearly shown the existence of such local interacting features, and cross-correlation and auto-correlation methods were used to identify the characteristic size of these features. This feature size was found to be similar to the applied fretting stroke for the tests at $p_m = 30$ MPa but was somewhat larger than the applied fretting stroke for the $p_m = 70$ MPa tests. It should be noted that if macroscopic wear-scar edges do exist, or if the overall worn surface geometry develops a curved profile (e.g. concave), then these are also likely to contribute to the measured variation in tangential force. A one degree-of-freedom model of a single surface peak undergoing fretting interaction with a corresponding surface groove was used to show how such interactions of the worn surfaces can produce the continuous increase in friction which was observed during the gross-slip phase.

The phenomenon of increasing friction with sliding may increase the area of the tangential force hysteresis loop, which would have energy implications for both frictional damping and fretting wear; however, additional inquiry would be needed to investigate this possibility. Further work is required to determine the effect of variables such as normal pressure and stroke length on the resulting local wear-scar feature sizes. Investigation of the relative importance of distributed local wear-scar features, as opposed to macroscopic wear-scar ends, in contributing to the friction variation may also be required. The work presented here has implications for researchers attempting to validate velocity dependent friction models using reciprocating sliding tests as, unless proper care is taken, interaction of the worn surfaces will produce a friction variation

which could easily otherwise be attributed to the velocity variation. Finally, the phenomenon discussed in this paper may need to be included in predictive models of frictional joints which undergo significant fretting wear.

Chapter 5

Asperity modelling of friction

5.1 Introduction

In Chapters 3 and 4, experimental aspects of sliding friction were explored. In this chapter, attention is turned towards the equally important task of modelling sliding friction. There are two overall aims in this: the first is in working towards developing predictive methods of determining friction, and the second is in advancing understanding of the physical mechanisms responsible for friction. The relevant background introducing the topic, as well as, previous research in this field has already been presented in Chapter 2. This chapter develops a model of sliding friction based on the sliding interaction of cylindrical and spherical metal asperities (with properties corresponding to Ti-6Al-4V). Essentially, by moving two elastic-plastic asperities over each other with a prescribed overlap using FE software (ABAQUS), friction coefficients can be predicted. A range of asperity overlaps (5–30% of the radius) larger than previously addressed in friction modelling studies are investigated, and a range of constant interface shear strengths (from zero up to the bulk material yield strength) are imposed at the junction interface to incorporate the ‘adhesion theory of friction’ proposed

by Bowden and Tabor. Overall effective friction coefficients are calculated for the asperity junction interactions and compared to experimental values derived from sliding friction tests on Ti-6Al-4V. To facilitate the high asperity overlaps, and interface shear strengths, asperity failure is also accounted for in the modelling. In addition, friction coefficients derived from cylindrical and spherical asperities are compared, and friction results from asperities with a failure criterion are compared to those with infinite ductility. A statistical generalisation of the results from the single asperity junction to a representative rough surface is also undertaken and compared with experiment. The goal is to see whether this mechanism of explaining friction by plasticity and tangential interface adhesion is capable of predicting realistic friction coefficients compared to those determined experimentally. It was shown in Chapter 2, how previous elastic-plastic models of sliding asperity interaction have under-predicted the friction coefficients (which were more representative of lubricated contact). The aim with this work is to determine whether modelling higher asperity overlaps, and including tangential adhesion can correct this defect.

5.2 Modelling Approach

The finite element software ABAQUS 6.9 [128] was used throughout the modelling work. Initially, it was intended to analyse 3D spherical asperities using the ‘implicit’ solver ‘ABAQUS Standard’, which provides a quasi-static result. However, given the demanding contact conditions and asperity overlaps being attempted, convergence difficulties were encountered when modelling with higher overlaps and interfacial shear strengths was attempted. For this reason, recourse was made to the ‘explicit’ solver ‘ABAQUS Explicit’ which is more robust in handling difficult contact problems.

‘ABAQUS Explicit’ is usually used in the context of solving dynamic problems in real time, but methods for the modelling of quasi-static problems using the explicit solution procedure are well established [128]. As a result of the change to an explicit code, it was decided to use cylindrical asperities rather than spherical ones for the majority of the modelling work so that a satisfactory methodology could be developed in ABAQUS Explicit with reasonable computation times. Although the main sequence of modelling was carried out with cylindrical asperities using the explicit solution procedure, a spherical model was also used in the present work for outlining the initial implicit modelling work, and also as part of a comparison (using ABAQUS Explicit) with the cylindrical model for the least severe asperity overlap. Hence, details of both the cylindrical and spherical asperity models are included below. Likewise, although the explicit FE solution procedure is used in obtaining the main results, in certain cases an implicit analysis was also carried out on both asperity types in order to verify the ‘explicit’ results. Thus, details of both types of analysis are described below.

5.2.1 Problem formulation

A schematic diagram of the sliding asperity interaction, which applies equally to the cylindrical and spherical asperities, is shown in Figure 5.1. The friction coefficients predicted by this single asperity junction model apply equally to the interaction of two surfaces that both have a constant asperity height and where all asperity junctions have the same overlap and interact independently. This was explained in Chapter 2 (Section 2.3.3).

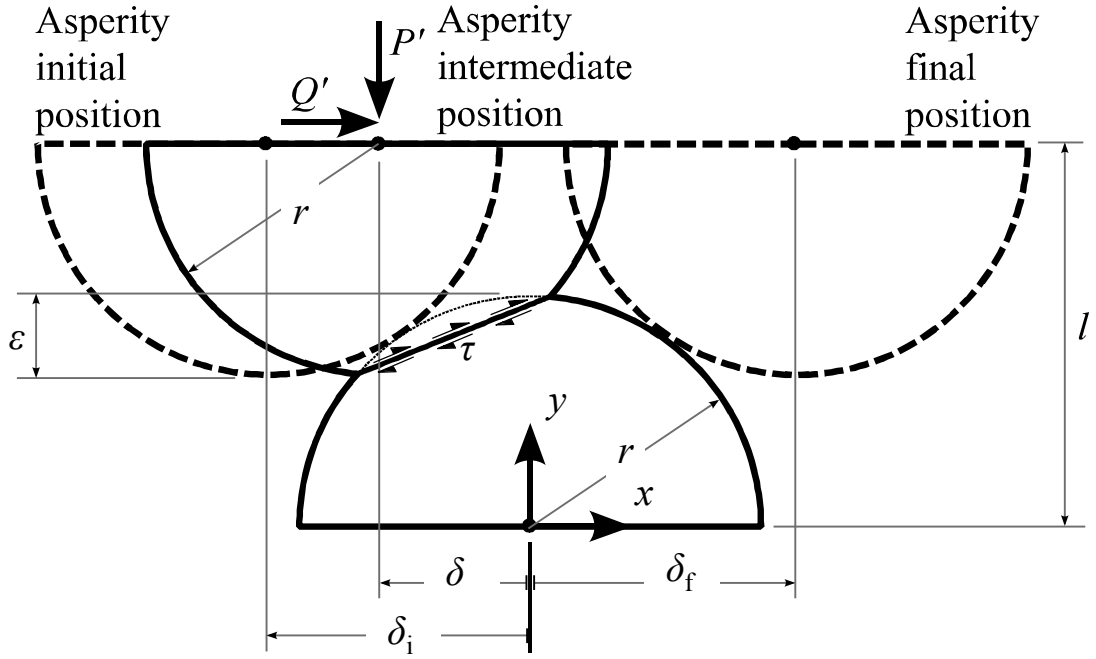


Figure 5.1: Schematic diagram of the asperity junction model.

In accordance with the observations made by Green [70] (see Section 2.3.3), the asperities are moved over each other with the vertical separation between asperity bases l held constant. During the analysis, the normal and tangential reaction forces (P' and Q' , respectively) are recorded. The average normal force is then given by:

$$\bar{P} = \int P' d\delta / \Delta\delta, \quad (5.1)$$

while the average tangential force is:

$$\bar{Q} = \int Q' d\delta / \Delta\delta, \quad (5.2)$$

where δ is the horizontal position of the upper asperity relative to the lower (the x - y axes are fixed to the lower asperity). The predicted friction coefficient is then defined as:

$$f = \frac{\bar{Q}}{\bar{P}}. \quad (5.3)$$

The quantities δ_i and δ_f denote the initial value of δ (when the asperities are just touching), and the final value of δ (when the asperities part contact), respectively. For the main modelling programme (involving cylindrical asperities in ‘ABAQUS Explicit’), analysis was carried out for six different overlaps ε ranging from 5 to 30 percent of the asperity radius r , and for each overlap, simulations were run at six different interfacial shear strength values τ ranging from 0 to σ_Y , where σ_Y is the yield strength of the bulk asperity material in simple tension. In addition, the initial modelling work using spherical asperities with the ‘implicit’ solver was carried out at an overlap of $(1\%)r$, while the comparison between cylindrical and spherical asperities (in ‘ABAQUS Explicit’) was undertaken at an overlap of $(5\%)r$.

5.2.2 Finite element model

Figure 5.2a and Figure 5.2b show the FE mesh used for the cylindrical asperity model, and the spherical asperity model, respectively. The cylindrical asperity-junction model consisted of a total of 67064 linear hexahedral elements (C3D8R), while the spherical model contained a total of 31104 linear hexahedral elements (C3D8R) concentrated around the contact region and a total of 157180 linear tetrahedral elements (C3D4) throughout the remaining part of the model. The element numbers quoted are for the whole junction model including both asperities. Since the continuum mechanics model used involves no scale dependent properties, a generic asperity radius of 100 mm was used. In the case of the cylindrical asperity model, it would be usual to conduct the analysis as a 2D plane strain problem. However, since asperity failure is being allowed for, it may be necessary in the course of a simulation for element surfaces which were originally within the interior of the asperity to interact as contact surfaces. In ABAQUS

6.9, this requires the use of the ‘general contact algorithm’ which is only available for 3D models; hence, a 3D model of the cylindrical asperities was constructed. Plane strain was simulated in the 3D model by constraining the front and back planar faces of the cylinder model to remain ‘in plane’ (i.e. $u_z = 0$, $ur_x = 0$, $ur_y = 0$), so that the model behaves as a plane strain slice. A relatively small thickness of 4 mm was chosen for the cylinders in order to minimise total element numbers. For the spherical asperity model, advantage was taken of symmetry, and only ‘half-hemispheres’ were modelled, therefore requiring the planar front face of the asperities to be constrained in the same way as the front and back faces of the cylinder model. Nodes on the top and bottom surfaces of the asperities (at $y = 0$ and $y = l$) were prevented from moving relative to each other, and were allowed only rigid body motion in the x -direction.

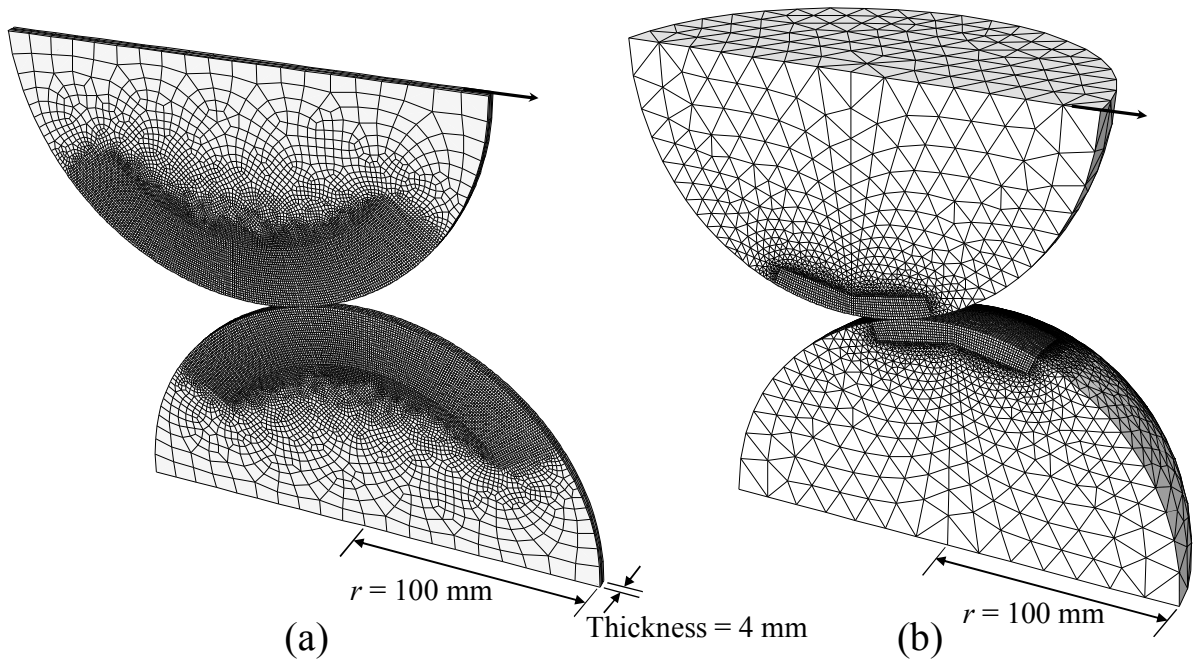


Figure 5.2: Finite element mesh for: (a) the cylindrical asperity junction, and (b) the spherical asperity junction.

Prescribed displacement was used for the ‘implicit’ models, while prescribed velocity was used in the ‘explicit’ models. When using the implicit solver, the lower asperity was

fixed at its base, while the upper asperity was given a prescribed displacement sufficient to complete a full junction interaction. However, for ‘ABAQUS Explicit’, it was found that a more symmetric result (between the two asperities) was obtained if the two asperities were given equal but oppositely directed velocity profiles.

At this point, an important aspect of attempting quasi-static modelling using an explicit solution procedure should be mentioned. Namely, without scaling of the mass and/or the loading rate, the analysis is likely to take a prohibitive length of time to solve. This is because the minimum stable time increment Δt (the smallest element dimension divided by the wave speed in the material) is likely to be very small, and a very large number of increments would be required to complete the analysis in a given (unscaled) time. However, the wave speed in the material can be reduced by scaling up the density (known as ‘mass scaling’), and this allows a larger value of Δt to be used. In addition, the loading rate can be increased so that the total analysis time decreases. This reduces the number of increments required. Using these measures, the analysis time can be vastly reduced.

Both mass and load-rate scaling, however, contribute to increasing the inertial (or dynamic) forces in the model. Therefore, it is important to choose a loading rate and mass scaling factor that keep the inertial forces within a relatively small fraction of the static forces. Choosing the optimum combination of loading rate and mass scaling can be a difficult task involving much ‘trial and error’. However, as long as a check on the energy outputs from the model reveals that the kinetic energy is a small fraction of the internal strain energy (less than 10% according to [128]), a quasi-static result can generally be assumed. For the asperity models with overlaps ranging from 10–30% of the radius, a mass scaling factor of 10^6 applied to all elements was used. In the case of the models with $(5\%)r$ overlap, the kinetic energy with this mass scaling factor was

slightly higher in proportion to the internal strain energy than recommended. Therefore, a slightly different approach was used for the $(5\%)r$ case: a minimum target time increment of 1.17×10^{-5} s was specified, and only elements producing time increments lower than this value had their mass scaled. This produced an equivalent overall mass scaling factor of 1.159×10^4 and reduced the kinetic energy to an acceptable level. In fact, the difference in overall predicted friction results between the two mass scaling options for the $\varepsilon = (5\%)r$ case was negligible, but the lower mass scaling was used to comply with recommended levels of kinetic energy.

The loading rate was defined in all cases by a special ‘strain rate’ defined as the ratio of the strain at the top of the asperity at vertical alignment ($\frac{1}{2}\varepsilon/r$) to the ‘time to reach vertical alignment’ (vertical alignment occurs when the axes of the two asperities coincide). This ratio was held constant at 0.0135s^{-1} for all models. This approach was taken in order to adjust the loading rates to account for the higher strains that occur as the asperity overlap is increased: the objective being to prevent increased strain rates at higher overlaps. Within this requirement, a velocity profile was defined (for both asperities in the junction) so that the velocity was smoothly increased from an initial value of zero to a final value occurring when the upper asperity had travelled a distance of $2\delta_i$ relative to the lower asperity. Had the asperities been given a constant velocity, the initial impact would be likely to induce the propagation of undesirable stress waves. Thus, it is important to increase the velocity in the smoothest possible way: this was done by using the ‘smooth amplitude curve’ option available within ABAQUS. The above conditions fully define the mass scaling and loading rates for all of the ‘explicit’ simulations analysed. With these loading conditions, the cylindrical asperity models with an overlap of $(5\%)r$ each took about 3 hours to complete, while the models with $(30\%)r$

overlap took in the region of 11 hours on a PC with a 3.00 GHz ‘Intel Core 2 Duo’ processor and 3.25 GB of RAM.

To enforce the required contact conditions on the interface, the ‘general contact algorithm’ was used for the ‘ABAQUS Explicit’ models, while the ‘contact pair algorithm’ was used for the ‘implicit’ models. In the case of the implicit model, surface-to-surface discretisation was used together with the ‘finite sliding formulation’ for the two defined ‘master-slave’ contact surfaces. For the ‘ABAQUS Explicit’ models, contact was defined for all element faces (internal and external): this measure was taken to allow for the correct contact enforcement of exposed element faces should the failure of nearby elements occur. The contact conditions are only enforced if a particular element face is exposed to the exterior. In order to enforce different values of constant shear strength τ on the interface, the ‘penalty’ friction formulation was used for all models. An interfacial shear strength cannot be defined directly with the penalty formulation; therefore, a very large friction coefficient of 1000 in conjunction with a ‘shear stress limit’ was used. This provision ensures that the interface will remain stuck until the shear stress limit is reached, whereupon sliding occurs. This shear traction is subsequently applied uniformly throughout the contact area as sliding proceeds. A third requirement for the penalty formulation is an elastic slip tolerance parameter (or factor): the ABAQUS default value of 0.005 was used. Automatic time incrementation, and the ‘NLGEOM’ nonlinear geometry option (see ref. [128]) was used for all models.

5.2.3 Material model

Material properties for the titanium alloy Ti-6Al-4V were chosen for the asperity models. This material was chosen because it is frequently used in aero-engines where it may be

subject to fretting. Therefore, experimental capabilities were available to perform friction tests so that the asperity model predictions could be compared to experimental sliding friction results. A plot of equivalent von Mises stress versus equivalent strain for the chosen material model is shown in Figure 5.3. The Young's modulus and Poisson's ratio were 115 GPa and 0.31, respectively, and a yield stress $\sigma_Y = 1000$ MPa was also defined for the material. These properties are easily obtained in the literature [129]. However, the hardening and subsequent failure behaviour of Ti-6Al-4V are more difficult to obtain, and experiments carried out by Petrinic and Wang [130] were used in specifying these aspects of the material model. Following yield, the model allows for a phase of linear isotropic strain hardening (A–B). When a certain level of strain is reached, ductile failure will occur in the material: for this reason, a failure criterion is included. Definition of a failure criterion in ABAQUS involves two separate aspects: specification of the point of 'failure initiation', and a suitable damage evolution law that governs the subsequent degradation of the material stiffness.

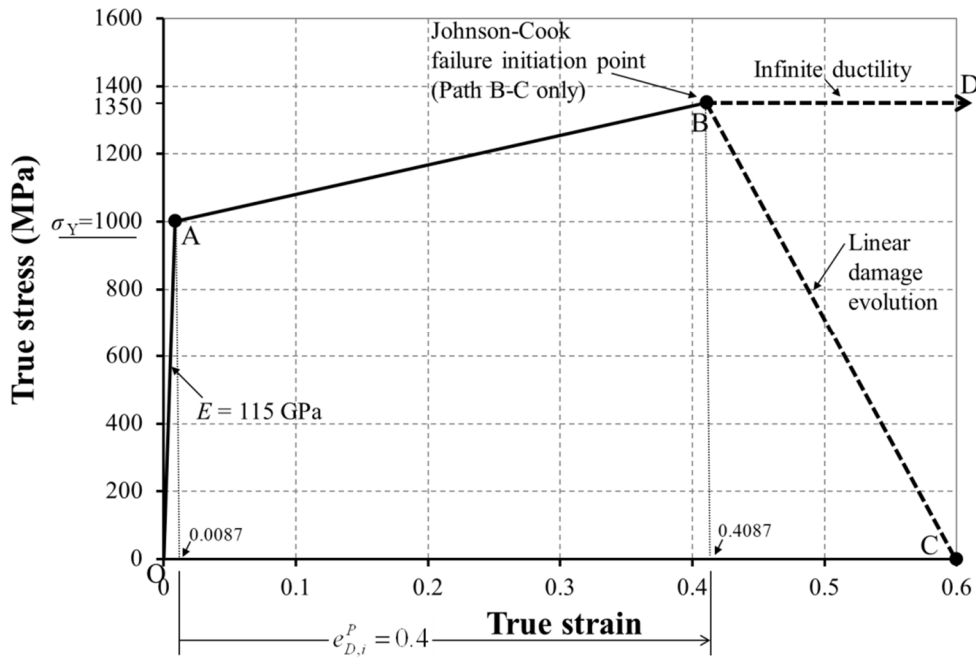


Figure 5.3: Material model of the titanium alloy Ti-6Al-4V used for the asperity modelling showing equivalent true stress versus equivalent true strain.

The Johnson-Cook (JC) failure criterion was used to define the point of failure initiation (point B in Figure 5.3). According to this model, the equivalent plastic strain at the initiation of damage is a function of stress triaxiality η given by:

$$e_{D,i}^P = D_1 + D_2 \exp(-D_3 \eta), \quad (5.4)$$

where the JC failure parameters $D_2 = 0.183$ and $D_3 = 0.452$ that define the shape of the ‘failure initiation strain’ versus stress triaxiality curve were obtained from characterisation work carried out by Peirs *et al.* [131]. The failure parameters determined in [131] predict damage initiation at relatively low strains (0.1–0.3 approx.) for the quasi-static loading case. However, Petrinic and Wang [130] found that the localised failure strains were significantly higher (0.5–0.65 approx.) than the overall average values. For this reason, the first damage parameter D_1 was adjusted from -0.05 in [131] to 0.242 in order to match the results obtained in [130]. As a result of the dependence on stress triaxiality, the point of damage initiation will vary somewhat according to the state of stress (in Figure 5.3, it is shown for the case of uniaxial tension). In the absence of any experimental data for the damage softening part of the curve (B–C), a simple linear damage evolution law was used to degrade the material stiffness to zero where the equivalent total strain reaches 0.6 (point C in Figure 5.3). When an element reaches point C on the curve, its stiffness has fully degraded, and the element is removed from the mesh. Gradual damage evolution to point C is required since complete failure at point B would lead to the propagation of stress waves and possible severe dynamic effects in the remaining part of the simulation. In addition to modelling the asperities with the failure criterion included, all simulations were repeated for the case where the asperities have infinite ductility: defined by path O–A–B–D in Figure 5.3.

5.2.4 Model verification

As with all finite element work, it is important to verify that the mesh gives convergent results. In addition, since ‘explicit’ solution procedures involving mass scaling were used in the present work to simulate quasi-static modelling, it was necessary to show that the results are indeed quasi-static. Also, in the case of the cylindrical asperities (which had to be modelled in 3D), it is useful to ensure that the results are in line with the 2D plane strain case. The results of the verification steps are outlined in Figure 5.4 and Figure 5.5. Figure 5.4a and b shows the evolution of the normalised tangential and normal forces for the cylindrical asperity junction with an overlap of $(5\%)r$ and $(10\%)r$, respectively, while Figure 5.4c shows the evolution of forces for a spherical asperity junction with $(5\%)r$ overlap. The forces are normalised by P_c , the critical normal load required to cause first plastic yield in the asperity under an orthogonal frictionless approach: this is calculated from elasticity theory as outlined in [132].

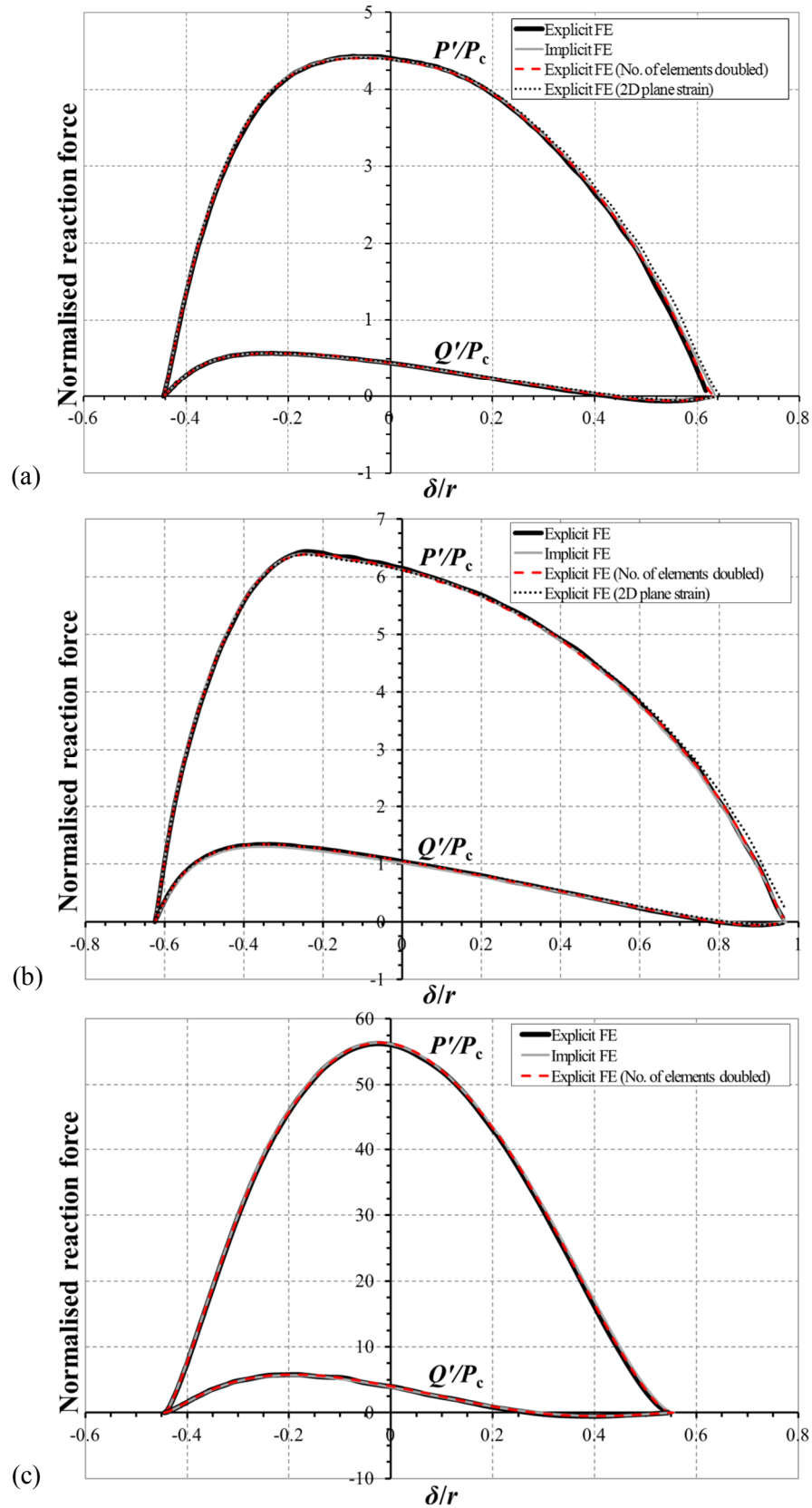


Figure 5.4: Model verification comparing junction forces from various models ('Explicit FE', 'Implicit FE', 'Explicit FE' with the number of elements doubled, and (for a and b only) 2D plane strain using 'Explicit FE') for: (a) cylindrical asperities with $\varepsilon = (5\%)r$, (b) cylindrical asperities with $\varepsilon = (10\%)r$, and (c) spherical asperities with $\varepsilon = (5\%)r$. The interface was frictionless in all cases.

The results in Figure 5.4a and b were obtained from four different models: the ‘explicit’ model itself, the ‘explicit’ model containing twice the number of elements, the equivalent model analysed using the ‘implicit’ solver, and the equivalent 2D plane strain model. Since the implicit solver guarantees a quasi-static result, its close agreement with the explicit solver in Figure 5.4a, and b shows that the explicit model is indeed free from significant dynamic effects. The good agreement with results from the models with double the number of elements indicates that mesh convergence has been achieved, while agreement with the results from the ‘explicit 2D plane strain’ case shows that the models are behaving as a 3D plane strain slice. Similar agreement was also obtained when verification was carried out at other cylinder overlaps. In Figure 5.4c, the normalised reaction forces for the spherical asperity junction with $\varepsilon = (5\%)r$ show good agreement between the different models – again indicating a converged quasi-static result. It should be noted that the verification ‘runs’ in ‘ABAQUS Implicit’ were only possible because they were run at the least severe overlaps, with $f = 0$, and with no material damage criterion.

Another useful test, used to show that an explicit analysis is essentially quasi-static, is to ensure that the kinetic energy during the analysis is a small fraction ($<10\%$ according to [128]) of the internal strain energy for most of the analysis. Figure 5.5 shows a plot comparing the kinetic energy with the internal strain energy for the model with $\tau = 0$, and $\varepsilon = (10\%)r$.

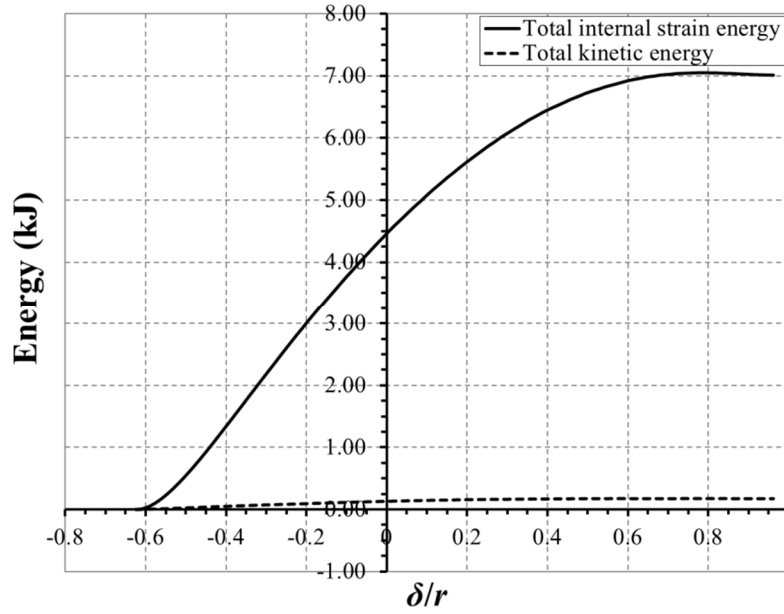


Figure 5.5: Comparison of total internal strain energy with total kinetic energy for the cylindrical asperity interaction with $\varepsilon = (10\%)r$ and $\tau = 0$.

This case is presented since the $(10\%)r$ models had the highest fraction of kinetic energy of all the cases investigated. Even here, Figure 5.5 shows that the kinetic energy remains within the recommended fraction of the internal energy. As outlined above (Section 5.2.2), the $(5\%)r$ model originally did not meet the requirements on the kinetic energy, and its mass scaling was reduced so that the resulting energy comparison was even better than for the $(10\%)r$ case shown in Figure 5.5. In summary, all models presented were shown to produce results which can be considered to be quasi-static.

5.3 Results and Discussion

In this section, the friction modelling results are analysed and compared with experimentally measured friction coefficients. A statistical adaptation of the results to the modelling of rough surfaces is also presented.

5.3.1 Friction modelling results

As mentioned above, an initial attempt was made to model *spherical* asperities using an ‘implicit’ FE solver; however, convergence difficulties were encountered when running models with the higher interface shear strengths and asperity overlaps. Some results *were* obtained from this modelling approach, and a sample of these is shown in Figure 5.6. Figure 5.6a and b show the evolution of the normalised tangential and normal junction forces, respectively, for an asperity overlap of $(1\%)r$. In each case, results from three different models are shown: fully elastic asperities with a frictionless interface, elastic-plastic asperities with a frictionless interface, and elastic-plastic asperities with an interface shear strength of $\tau = 0.1\sigma_Y$. It is clear from Figure 5.6a, that the average tangential force is zero for the interaction of the fully elastic asperities; since, the energy invested in deforming the asperities is recovered in equal measure as they relax. Applying Equations (5.1–5.3) above, this leads to a friction coefficient of zero. When elastic-plastic behaviour is introduced, not all of the energy is recovered, hence, there is a non-zero average tangential force. This gave a small friction coefficient of 0.021 which is entirely due to plasticity. Finally, when the elastic-plastic model is run with the interface shear strength present, the average tangential force increases (Figure 5.6a), but the average normal force (Figure 5.6b) changes little from the frictionless case. This caused the predicted friction coefficient to increase to 0.069. In this case, the friction coefficient arises both from the combination of increased energy loss due to plastic deformation and from interfacial energy dissipation.

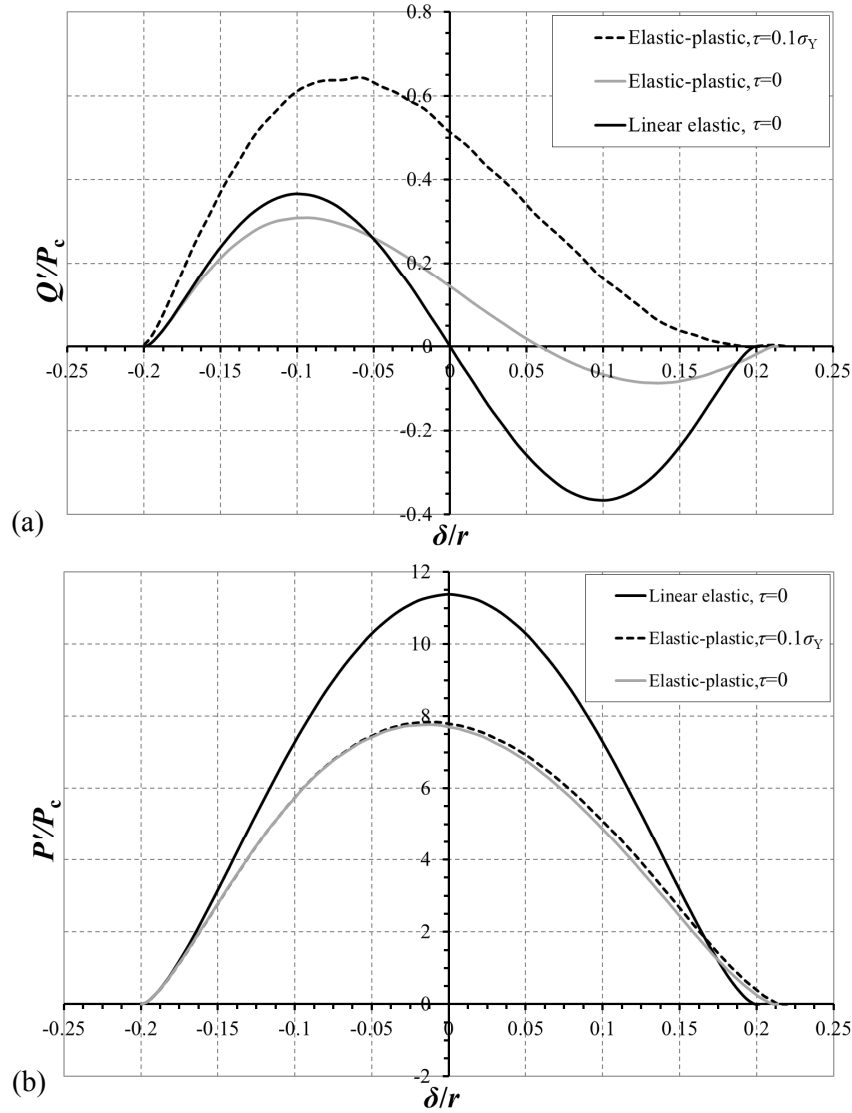


Figure 5.6: Comparison of: (a) the dimensionless tangential junction force Q'/P_c , and (b) the dimensionless normal junction force P'/P_c , for the spherical asperities modelled in ‘ABAQUS Implicit’ for $\varepsilon = (1\%)r$. Results are shown for: the purely elastic case with $\tau = 0$ (predicted friction coefficient: 0), the elastic-plastic case with $\tau = 0$ (predicted friction coefficient: 0.021), and the elastic-plastic case with $\tau = 0.1\sigma_Y$ (predicted friction coefficient: 0.069).

As discussed previously, in order to analyse models with higher asperity overlaps and interfacial shear strengths, the remaining modelling was carried out using ‘ABAQUS Explicit’. Also, to allow reasonable computation times, cylindrical asperities were used for the main analysis. Cylinder overlaps ε of 5, 10, 15, 20, 25 and 30 percent of the radius were analysed, and for each overlap, interface shear strengths equal to the following fractions of the bulk yield stress σ_Y were applied: 0, 0.2, 0.4, $1/\sqrt{3}$, 0.8 and 1. Figure 5.7 shows the normalised force histories for the $\varepsilon = (10\%)r$ case. Results are

shown for four values of interface strength τ ranging from the frictionless value of $\tau = 0$ to $\tau = \sigma_Y/\sqrt{3}$ (the yield strength of the bulk material in shear).

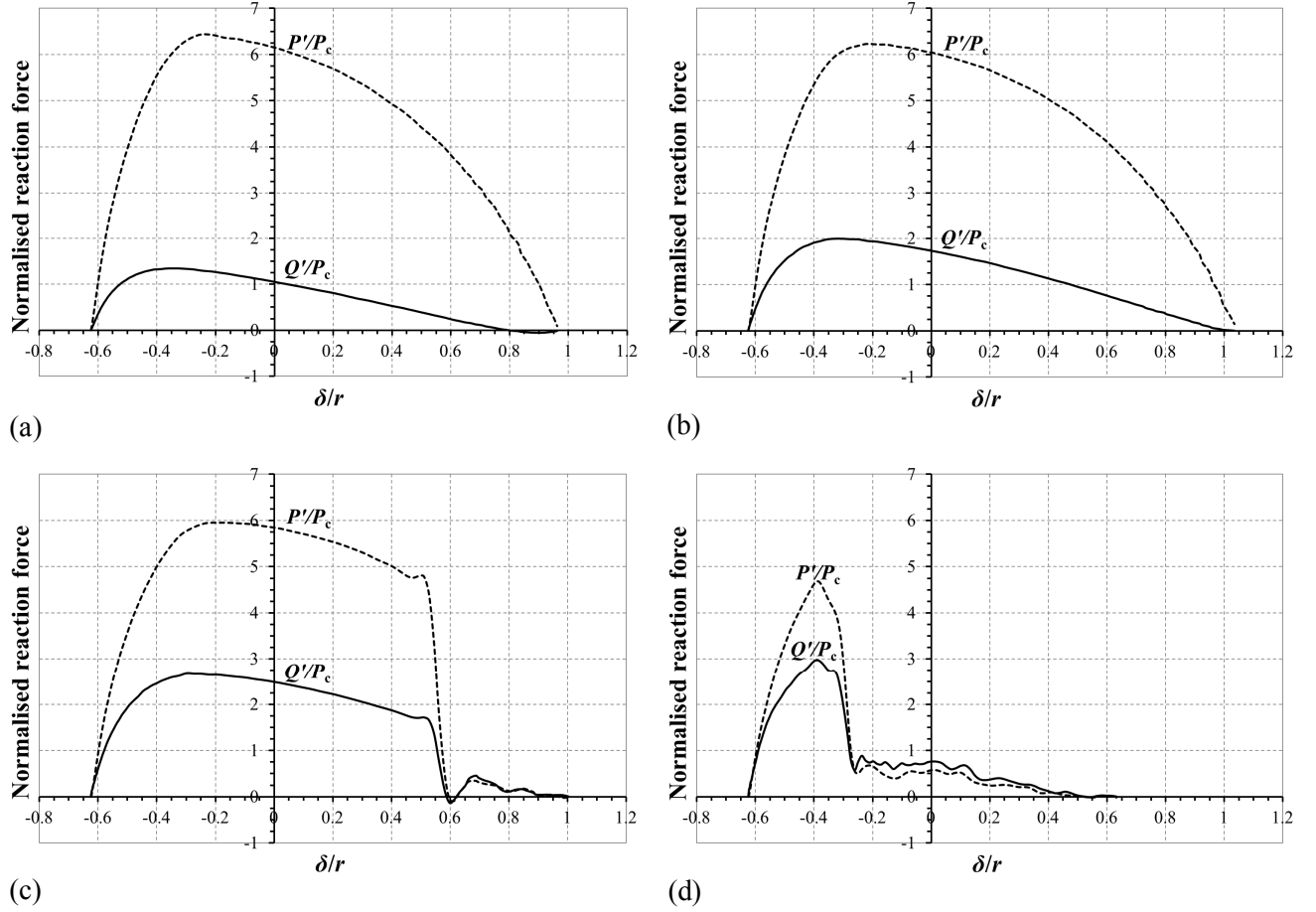


Figure 5.7: Evolution of the normalised tangential (Q'/P_c) and normal (P'/P_c) junction forces for the cylindrical asperity junctions (material model O–A–B–C in Figure 5.3) with $\varepsilon = (10\%)r$ for: (a) $\tau = 0$, (b) $\tau = 0.2\sigma_Y$, (c) $\tau = 0.4\sigma_Y$, and (d) $\tau = 1/\sqrt{3}\sigma_Y$.

For $\tau = 0$ (Figure 5.7a) and $\tau = 0.2\sigma_Y$ (Figure 5.7b), no material failure takes place, but when τ is increased further (as in Figure 5.7c and Figure 5.7d), material failure occurs and the forces subsequently drop off sharply. As the interface strength increases in Figure 5.7, the average tangential force increases in relation to the average normal force, thus raising the effective friction coefficient.

Contour plots of equivalent plastic strain (ABAQUS terminology: PEEQ) for the $\varepsilon = (10\%)r$ case discussed above, are shown in Figure 5.8 and Figure 5.9 for the interface

strength values of $\tau = 0$, and $\tau = 0.4\sigma_Y$, respectively. In each figure, three positions of the asperity junction interaction are shown. For the $\tau = 0$ case in Figure 5.8 (frictionless case), no asperity failure occurs as noted from Figure 5.7a. However, as shown in Figure 5.7c, an interface shear strength $\tau = 0.4\sigma_Y$ is sufficient to cause asperity failure, and the corresponding failure geometry is shown in Figure 5.9. Failure begins beneath the surface at the region of highest plastic strain, and propagates until part of the asperity becomes detached. In cases of higher interfacial shear, failure begins at the interface itself. This detachment of asperity fragments, or ‘wear’, is a common feature of frictional sliding. Good symmetry between the strain distributions of the two interacting asperities is apparent from Figure 5.8. In Figure 5.9, however, an asymmetry seems to develop near to the onset of failure. Both asperities are identical in terms of mesh, material model etc., and one might expect to see a symmetric response. However, it is likely that the accumulation of small numerical rounding errors associated with the ‘explicit’ solution procedure causes failure to occur in one asperity, rather than in the other. This is similar to the situation that would occur with real asperities, where a small defect might cause a similar response. The maximum equivalent plastic strains, during, and after failure (0.64 and 0.62 at positions $\delta/r = 0.55$ and $\delta/r = 1$, respectively) shown in Figure 5.9 are in good agreement with the material model described earlier in Figure 5.3 which defines an equivalent plastic strain to complete failure of approximately 0.6.

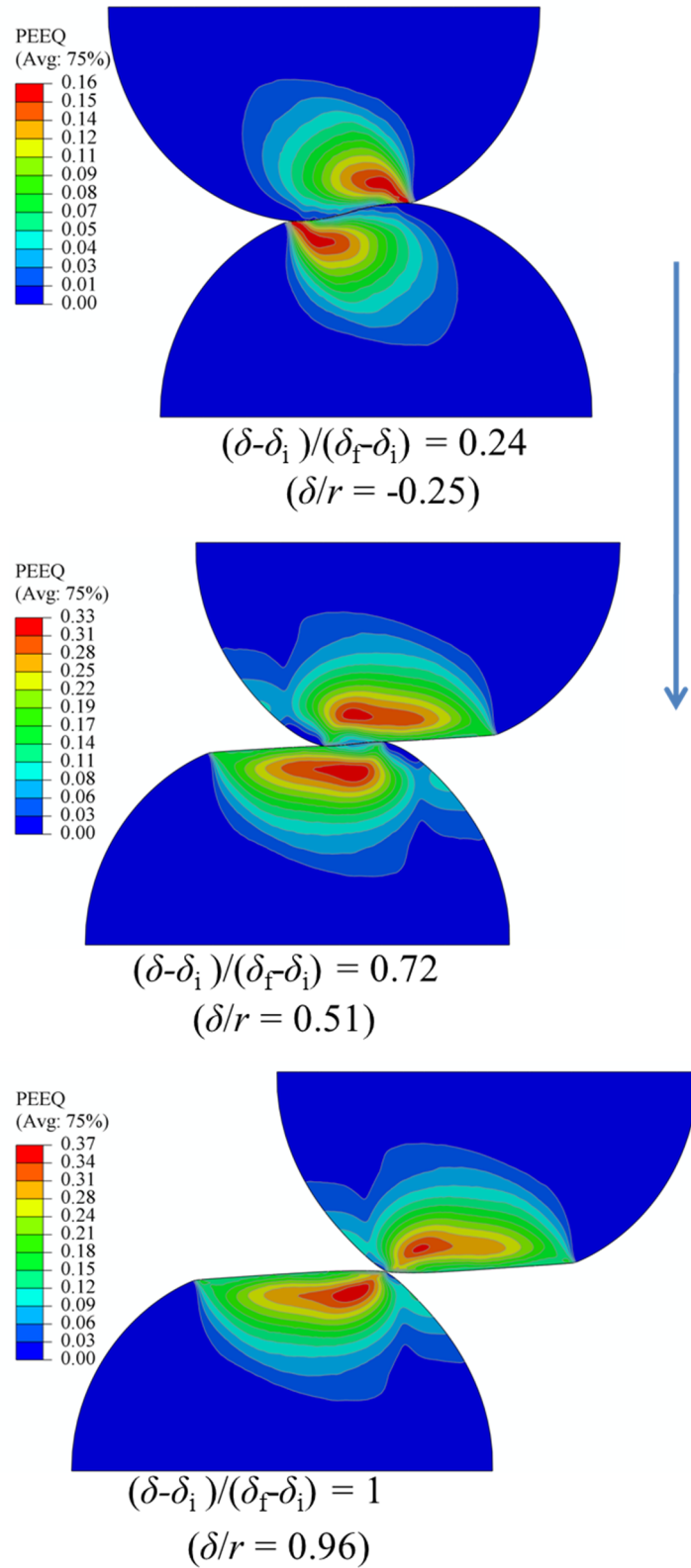


Figure 5.8: Contour plots of equivalent plastic strain (PEEQ) for three different positions within the interaction of the cylindrical asperity junction with $\varepsilon = (10\%)r$ and $\tau = 0$. The quotient $(\delta - \delta_i)/(\delta_f - \delta_i)$ is the fraction of the complete junction interaction which has occurred, whereas δ/r is the conventional normalised position of the upper asperity relative to the lower. Material model O–A–B–C was used (see Figure 5.3), but no material failure occurred for this model. The junction force histories for this case are given in Figure 5.7a.

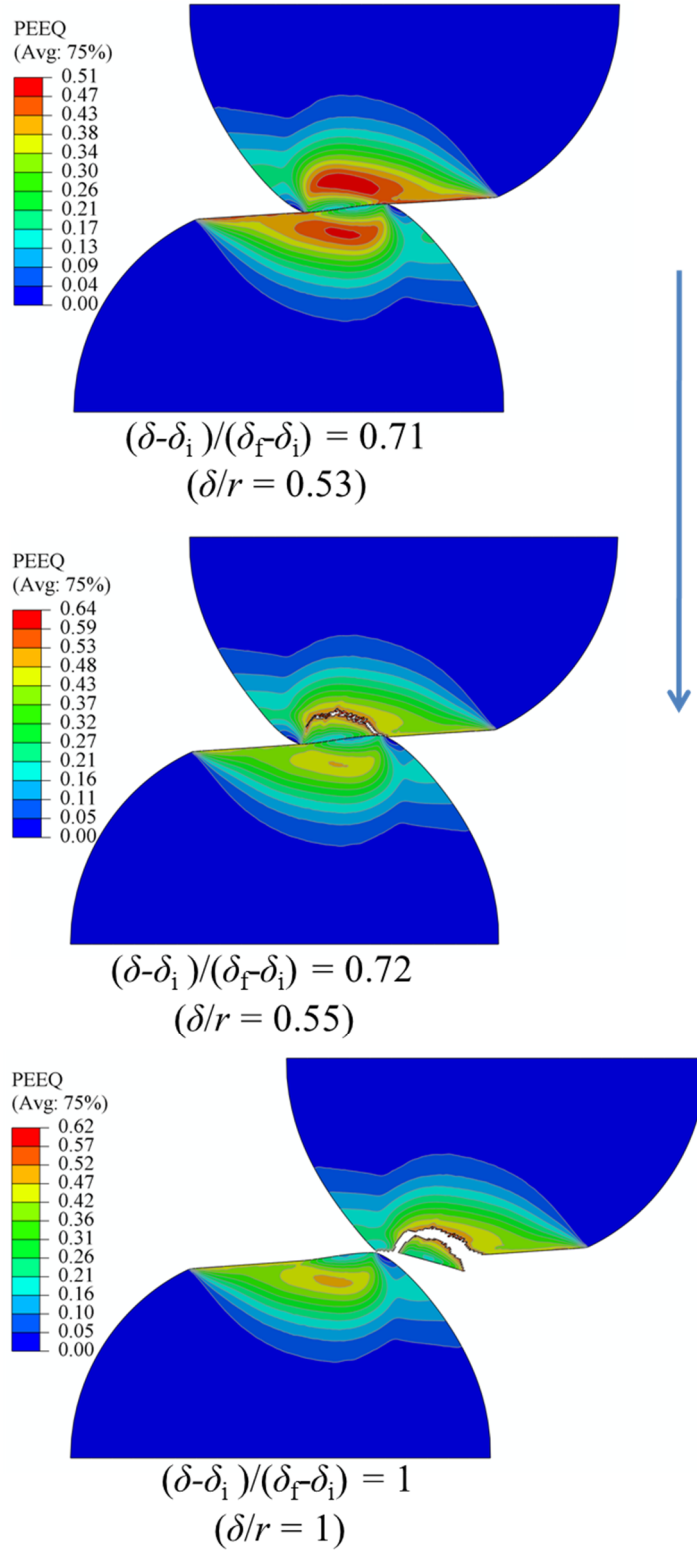


Figure 5.9: Contour plots of equivalent plastic strain (PEEQ) for three different positions within the interaction of the cylindrical asperity junction with $\varepsilon = (10\%)r$ and $\tau = 0.4\sigma_Y$. The quotient $(\delta - \delta_i)/(\delta_f - \delta_i)$ is the fraction of the complete junction interaction which has occurred, whereas δ/r is the conventional normalised position of the upper asperity relative to the lower. Material model O–A–B–C was used (see Figure 5.3), and failure occurs between the first position ($\delta/r = 0.53$) and the second position ($\delta/r = 0.55$). The junction force histories for this case are given in Figure 5.7c.

The main results for cylindrical asperities are given in Figure 5.10. Figure 5.10a and b show predicted friction coefficients for the case when the material failure criterion is included (path O–A–B–C in Figure 5.3), while Figure 5.10c shows the results for asperities with infinite ductility (path O–A–B–D in Figure 5.3).

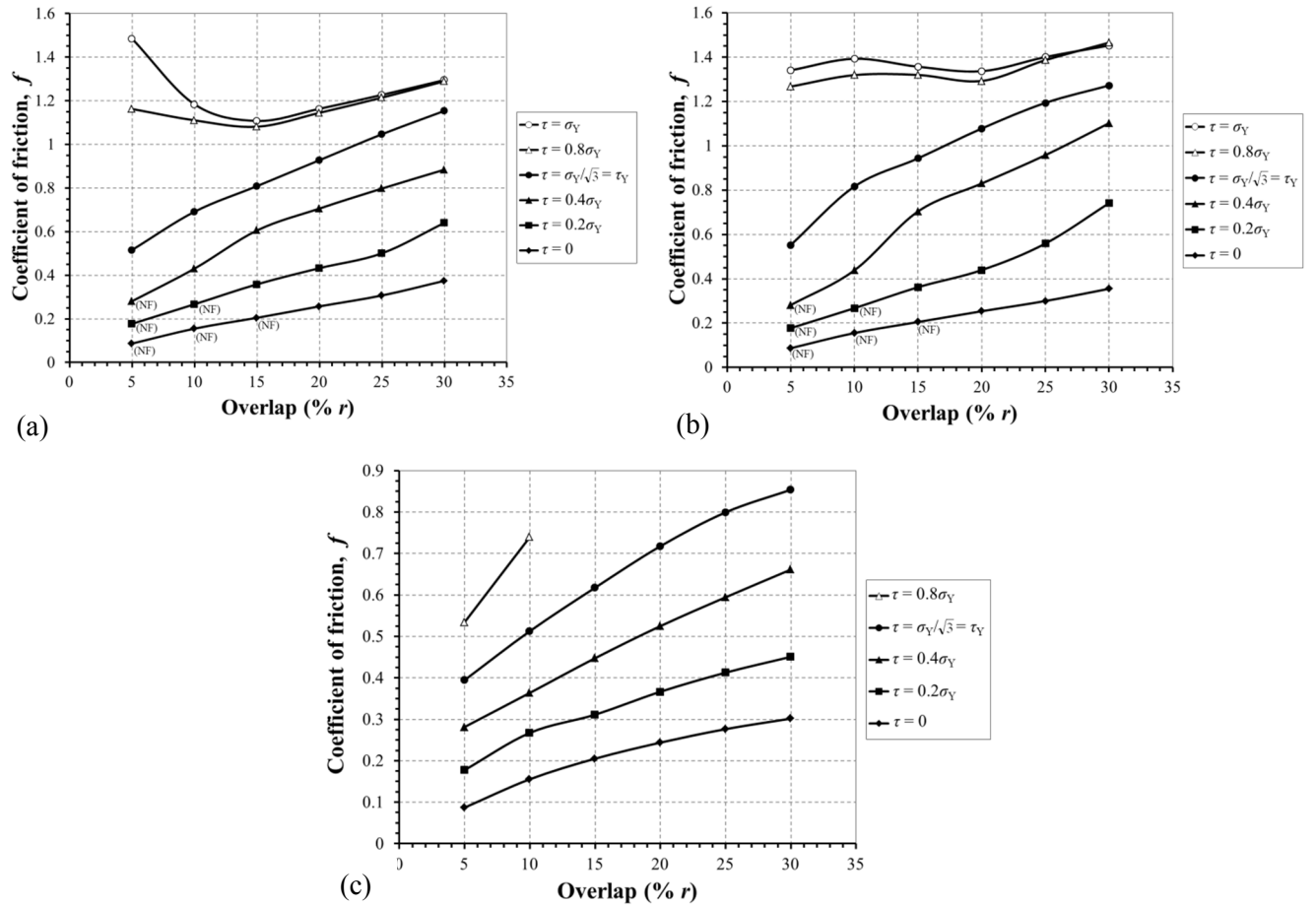


Figure 5.10: Predicted sliding friction coefficients versus overlap for the cylindrical asperity junctions showing results for the full range of interface shear strengths investigated: (a) friction coefficients calculated based on the force evolutions up to first element failure for material model O–A–B–C in Figure 5.3, (b) friction coefficients calculated based on the force evolutions for the complete asperity junction interaction using material model O–A–B–C in Figure 5.3, and (c) friction coefficients calculated based on the force evolutions for the complete asperity junction interaction using material model O–A–B–D in Figure 5.3. In (a) and (b), failure has occurred for all data points other than those denoted NF (No-Failure), and in (c) no failure occurs for any of the models since the material is infinitely ductile.

In general, friction increases with both asperity overlap ε and interface shear strength τ . In all cases, the predicted friction coefficients were calculated from the force evolution curves using Equations. (5.1–5.3). In Figure 5.10a, the average forces were calculated

based on the evolution of the forces only up to the point of first element failure within the asperities. Taking the force evolution curves in Figure 5.7c as an example, this means that the integrals in Equations (5.1 and 5.2) would be evaluated only up to the point at which the sharp drop off in the forces initiates. In both Figure 5.10b and Figure 5.10c, the friction coefficients were calculated based on the complete evolution of the asperity junction. This is an important distinction in models where failure occurs, because, after the point of failure initiation, the reaction force curves can become difficult to interpret (see Figure 5.7c and Figure 5.7d). This is because failure and removal of surface elements can produce a ‘stepped’ surface which may not be physically realistic. Despite this, the predicted friction coefficients based on the evolution up to first failure (Figure 5.10a) show much the same trends and magnitudes as those based on the complete evolution (Figure 5.10b).

There is a slight difference in behaviour for the $\tau = 0.8\sigma_Y$ and $\tau = \sigma_Y$ curves where the friction coefficient reduces as the overlap is increased from about 5 to 15% in Figure 5.10a, but not in Figure 5.10b. This is thought to be due to the fact that asperity failure occurs very quickly in the analysis for these cases – due to the very high interface strengths. In both Figure 5.10a and Figure 5.10b, the $\tau = 0.8\sigma_Y$ and $\tau = \sigma_Y$ curves seem to be converging on an upper-bound solution. This is because the interface shear strength has been increased beyond the shear strength of the bulk material. The plot for the case of the infinitely ductile asperities (Figure 5.10c), predicts slightly lower friction coefficients, but the same trend is apparent. In Figure 5.10c, it was only possible to determine two results for the $\tau = 0.8\sigma_Y$ case, and no results were obtainable at $\tau = \sigma_Y$. This is because, at high interface shear strengths, the interface remains stuck, and since there is infinite ductility, the elements near the surface become very distorted. In general, the effective friction coefficients in Figure 5.10 vary from about 0.1 to 1.5 depending on

the overlap and interface shear strength. These are of the same order as measured for metals sliding in air (0.2 to 1.5 [83]).

It is clear that the effective friction coefficients derived for $\tau = 0$ arise purely from the dissipation of energy due to plasticity; however, when interfacial shear strength τ is introduced at the interface, energy will be dissipated in both plasticity and in overcoming the interface shear strength. This energy dissipated at the interface is equivalent to the energy required to break the adhesion bonds envisaged by Bowden and Tabor [62]. Their theory of friction involved the combination of ‘ploughing’ and ‘shearing’ where ‘ploughing’ is equivalent to the plastic deformation in the present modelling approach, and ‘shearing’ is simulated by the interface shear strength τ . In determining the effective friction coefficient f using Equations (5.1–5.3), the work done by external forces $U_E = \int Q' d\delta$ is required in calculating the average tangential force $\bar{Q}$. This work (which is done only by Q' , since P' undergoes no displacement) arises from the dissipation of energy due to plasticity U_P , and the dissipation of energy due to shearing at the interface U_S , so that:

$$U_E = U_P + U_S. \quad (5.5)$$

It is useful to compare the relative contributions to the predicted friction coefficients of both U_P , and U_S . These were extracted from the ABAQUS output results.

Table 5.1 shows the values of these energies, as well as the ratio U_S/U_P , for the case of infinitely ductile asperity junctions (material model O–A–B–D in Figure 5.3) corresponding to the friction predictions in Figure 5.10c. This case was chosen for the energy comparison as the asperity failure involved in the other models would introduce complications.

Table 5.1 shows that U_P , U_S , and the ratio U_S/U_P all generally increase as the interface shear strength τ is increased. The energy dissipated in plasticity increases with τ because of the increased deformation caused by the higher shear traction. Both U_P , and U_S also increase as the asperity overlap is increased due to increased deformation and contact area, respectively; however, the ratio U_S/U_P reduces as the overlap is increased (as plastic deformation begins to dominate).

Table 5.1 shows that, in general, either plastic dissipation, or interfacial shear dissipation may dominate, depending on the amount of interface shear strength and asperity overlap.

Table 5.1: Comparison of the energy U_P dissipated in plasticity, with the energy U_S dissipated in overcoming the interface shear traction τ . These values apply to the friction results in Figure 5.10c (i.e. the complete evolution of cylindrical asperity junctions with no failure criterion included – path O–A–B–D in Figure 5.3).

Overlap (% r)	Energy parameter	$\tau=0$	$\tau=0.2\sigma_Y$	$\tau=0.4\sigma_Y$	$\tau=\sigma_Y/\sqrt{3}$
5	U_P (kJ)	1.76	2.06	2.69	3.99
	U_S (kJ)	0	1.91	3.98	5.54
	U_S/U_P	0	0.93	1.48	1.39
10	U_P (kJ)	6.87	8.02	9.65	12.75
	U_S (kJ)	0	4.40	7.58	10.07
	U_S/U_P	0	0.55	0.79	0.79
15	U_P (kJ)	13.33	15.07	18.08	22.38
	U_S (kJ)	0	5.19	9.98	12.77
	U_S/U_P	0	0.34	0.55	0.57
20	U_P (kJ)	20.56	23.07	27.09	31.33
	U_S (kJ)	0	6.46	11.96	14.53
	U_S/U_P	0	0.28	0.44	0.46
25	U_P (kJ)	28.49	31.68	36.75	40.58
	U_S (kJ)	0	7.57	13.72	16.16
	U_S/U_P	0	0.24	0.37	0.39
30	U_P (kJ)	36.90	40.97	46.09	53.56
	U_S (kJ)	0	8.67	15.13	18.39
	U_S/U_P	0	0.21	0.33	0.34

It is probably true that spherical asperities are more representative of the asperities that exist on real engineering surfaces. These are computationally more time consuming to analyse, and cylindrical asperities were, therefore, used in deriving the plots in Figure

5.10. It is instructive to compare friction predictions from each of these asperity types at a particular overlap. This comparison was carried out for the $\varepsilon = (5\%)r$ case where analysis times are reasonable. The result is shown in Figure 5.11.

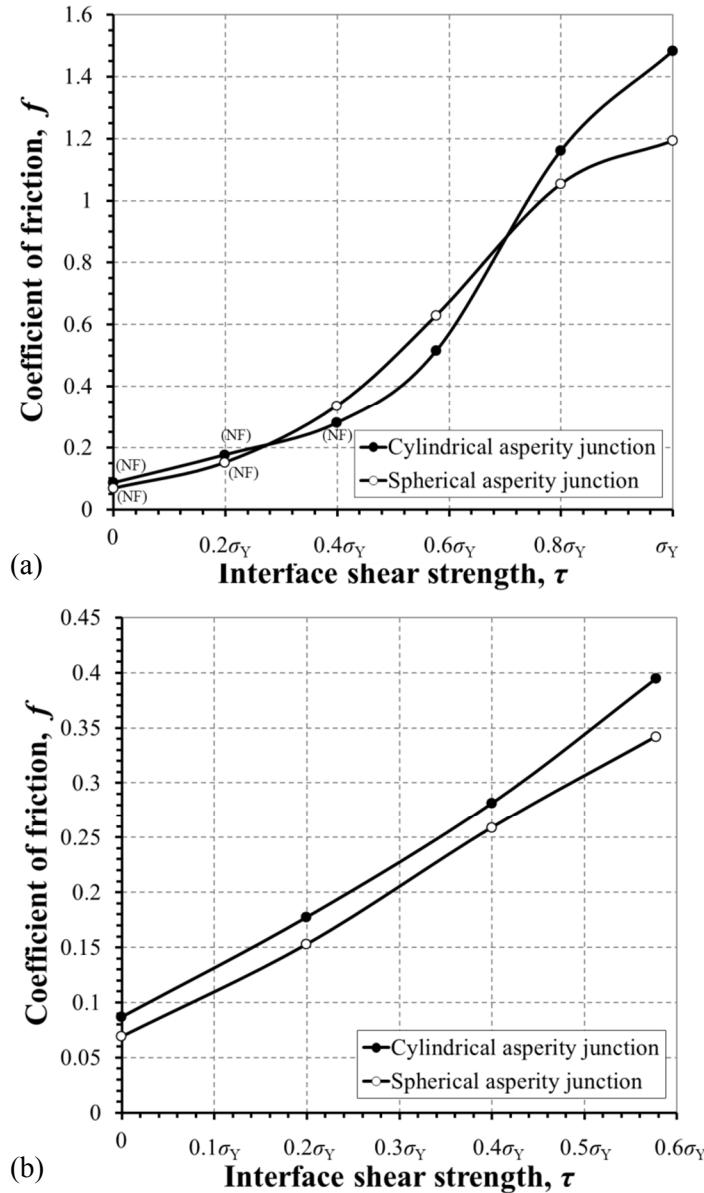


Figure 5.11: Comparison of predictions of the coefficient of friction from cylindrical and spherical asperity junctions for the $\varepsilon = (5\%)r$ case: (a) friction coefficients calculated based on the evolution of the junction forces up to first element failure using material model O-A-B-C in Figure 5.3, and (b) friction coefficients based on the complete junction evolution using asperities of infinite ductility (material model O-A-B-D in Figure 5.3). In (a) failure has occurred for all data points other than those denoted NF (No-Failure), whereas, in (b) no failure occurs for any of the models since the material is infinitely ductile.

Figure 5.11a shows the results based on the junction forces up to the point of first element failure, while Figure 5.11b applies to the complete evolution of infinitely ductile asperities. In each case, the overall behaviour, and the magnitudes of the predicted friction coefficients are similar between the cylindrical and spherical asperity junctions. Therefore, it might be expected that the underlying physics, determined above for the cylindrical asperities, would be generally applicable to spherical asperities.

5.3.2 Statistical analysis: Application to rough surfaces

The friction coefficients determined earlier apply to the interaction of two surfaces where the asperity heights and overlaps are constant for all asperity junctions. Real rough surfaces, however, exhibit a statistical distribution of asperity heights which leads to a variation in the possible asperity overlaps. Here, a ‘Greenwood-Williamson’ type method outlined by Franse *et al.* [133], and Faulkner and Arnell [78] is used to apply results from the cylindrical asperity case to the rough surface situation. This is done using the values of $\bar{Q}$ and $\bar{P}$ which were previously used to determine the friction results in Figure 5.10a. Analysis is carried out for four values of interface strength τ (0, $0.2\sigma_Y$, $0.4\sigma_Y$ and $1/\sqrt{3}\sigma_Y$). The procedure is outlined here for the $\tau = 0$ case. First, power-law curves were fitted to the average force versus overlap data:

$$\frac{\bar{Q}}{\bar{P}_c} = 15.154 \left(\frac{\varepsilon}{r} \right)^{1.347} \quad (5.6)$$

$$\frac{\bar{P}}{\bar{P}_c} = 16.085 \left(\frac{\varepsilon}{r} \right)^{0.553} \quad (5.7)$$

The approach, as first used by Greenwood and Williamson [64], involved a rigid flat surface pressed against a rough elastic surface with a statistical distribution of asperity

heights. The adaption of this technique to the modelling of the sliding interaction between asperities on both participant surfaces used by Franse *et al.* [133], and Faulkner and Arnell [78] requires a slightly different configuration. A schematic diagram of the modelling approach is shown in Figure 5.12.

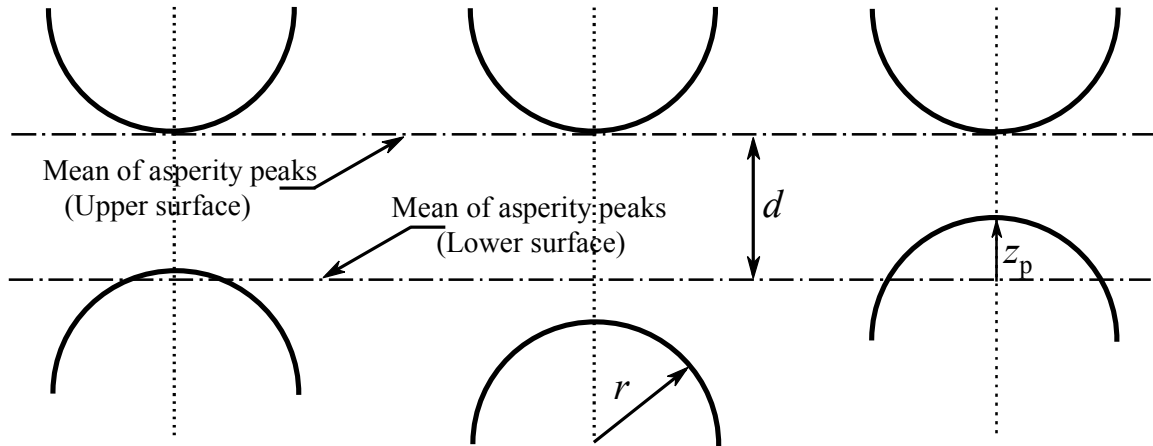


Figure 5.12: Schematic diagram of the rough surface contact model.

A rough surface with a stochastic distribution of asperities ϕ is brought into contact with a surface whose asperity heights are constant. The asperity peak heights of the stochastic surface, denoted by z_p , are measured from the mean plane of asperity heights, and the distance between the reference planes of the two surfaces is denoted by d . To allow the Greenwood-Williamson approach to apply to this setup, it is necessary to assume that each asperity on the stochastic surface can interact with a corresponding asperity on the ‘flat’ surface, so that when tangential movement occurs, each pair of asperities will undergo a sliding interaction where the average normal and tangential forces will depend on the overlap according to Equations (5.6 and 5.7). All asperities are assumed to have the same radius r , and each asperity pair acts independently of the other asperities. Assuming that there are N_a asperities on each surface, the expected total tangential and normal forces will be given by:

$$\frac{Q}{P_c} = N_a \int_d^\infty \bar{Q} \phi(z_p) dz_p \quad (5.8)$$

$$\frac{P}{P_c} = N_a \int_d^\infty \bar{P} \phi(z_p) dz_p \quad (5.9)$$

Given that $\varepsilon = z_p - d$, and substituting Equations (5.6 and 5.7) into Equations (5.8 and 5.9), we obtain:

$$\frac{Q}{P_c} = 15.154 N_a \left(\frac{1}{r}\right)^{1.347} \int_d^\infty (z_p - d)^{1.347} \phi(z_p) dz_p \quad (5.10)$$

$$\frac{P}{P_c} = 16.085 N_a \left(\frac{1}{r}\right)^{0.553} \int_d^\infty (z_p - d)^{0.553} \phi(z_p) dz_p \quad (5.11)$$

Now, assuming a standardised exponential distribution of asperity peak heights, and normalising the variables by the standard deviation of asperity peak heights σ_p (so that: $h = d/\sigma_p$, $s = z_p/\sigma_p$ and $r^* = r/\sigma_p$) gives:

$$\frac{Q}{P_c} = 15.154 N_a \left(\frac{1}{r^*}\right)^{1.347} \int_h^\infty (s - h)^{1.347} e^{-s} ds \quad (5.12)$$

$$\frac{P}{P_c} = 16.085 N_a \left(\frac{1}{r^*}\right)^{0.553} \int_h^\infty (s - h)^{0.553} e^{-s} ds \quad (5.13)$$

A Gaussian distribution of asperity heights is perhaps more common, however, the integrals that result are not tractable in closed form as for the exponential distribution. In any case, Greenwood and Williamson [64] suggest that the upper asperities of a Gaussian surface can be adequately described by an exponential distribution. Therefore, under light to moderate loads, this approximation is valid for Gaussian surfaces. The integrals in Equations (5.12 and 5.13) may then be evaluated as:

$$\int_h^\infty (s - h)^n e^{-s} ds = n! e^{-h} \quad (5.14)$$

Substituting the result of Equation (5.14) into Equations (5.12 and 5.13) gives:

$$\frac{Q}{P_c} = 15.154 N_a \left(\frac{\sigma_p}{r} \right)^{1.347} (1.347!) e^{-h} \quad (5.15)$$

$$\frac{P}{P_c} = 16.085 N_a \left(\frac{\sigma_p}{r} \right)^{0.553} (0.553!) e^{-h} \quad (5.16)$$

Finally, dividing Equation (5.15) by Equation (5.16) gives the overall coefficient of friction:

$$f = 1.27 \left(\frac{\sigma_p}{r} \right)^{0.79} \quad (5.17)$$

As a result of choosing an exponential distribution of asperity heights, there is no dependency of the friction coefficient f on the normal separation of the surfaces (and hence on normal load) in Equation (5.17) (which is consistent with the Amontons-Coulomb description of friction). The result in Equation (5.17) for the $\tau = 0$ case, together with the results for $\tau = 0.2\sigma_Y$, $\tau = 0.4\sigma_Y$, and $\tau = 1/\sqrt{3} \sigma_Y$ are shown in Figure 5.13. As expected, the predicted friction increases with interface shear strength τ . The analysis predicts a friction coefficient which is dependent on a ‘roughness’ parameter σ_p/r , and Figure 5.13 shows that f increases as σ_p/r increases. Both the standard deviation of asperity peak heights, and the mean asperity radius are difficult parameters to measure with commercially available profilometry equipment, but studies which are available in the literature [134-136] suggest that σ_p/r lies within the range shown in Figure 5.13 for most engineering surfaces. Finally, we note that the magnitudes of the friction coefficients predicted in Figure 5.13 are again within an appropriate range for sliding friction of metals in ambient conditions.

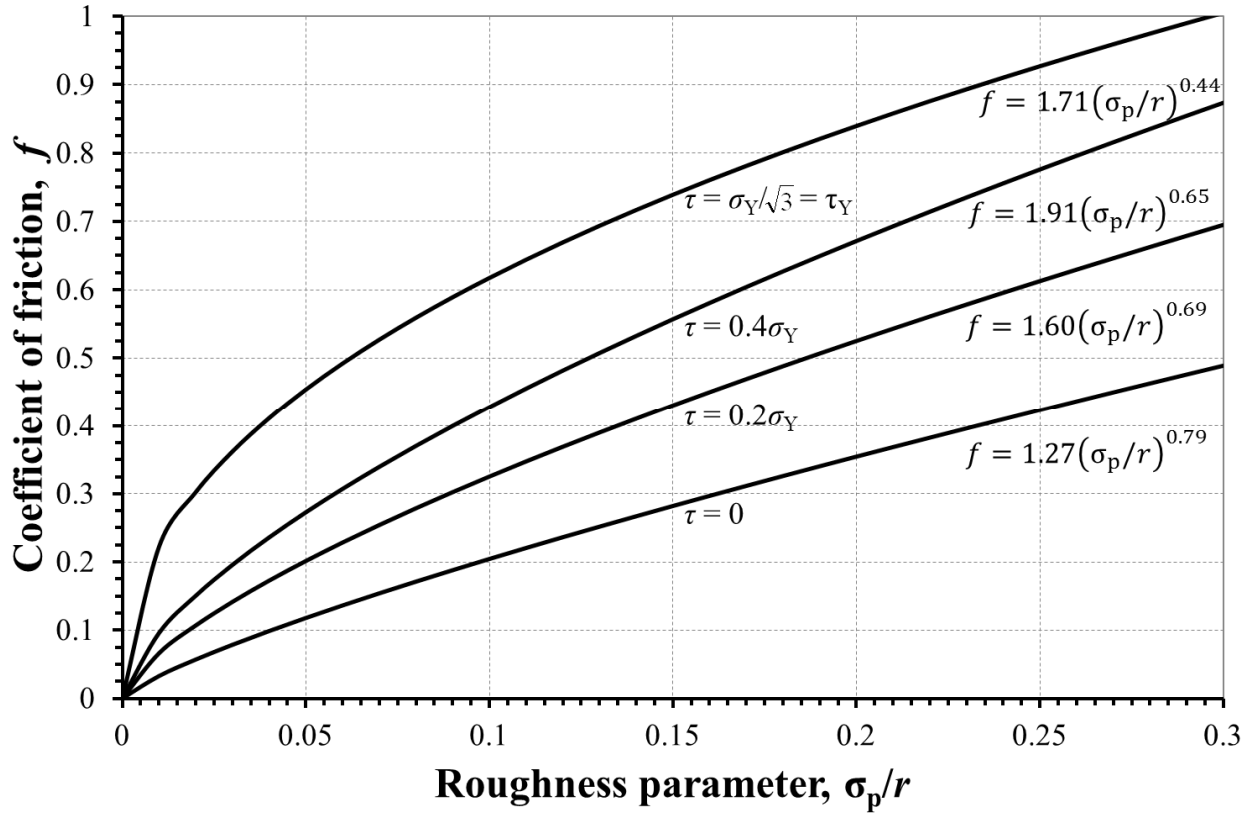


Figure 5.13: Predicted friction coefficients versus the surface topography parameter σ_p/r for the stochastic rough surface model applied to the average junction forces $\bar{Q}(\varepsilon)$ and $\bar{P}(\varepsilon)$ used in deriving the friction coefficients in Figure 5.10a.

5.3.3 Comparison with sliding friction experiments

Although the friction coefficients predicted above are clearly consistent with values typically measured for metals sliding in atmospheric conditions, it is useful to experimentally determine the friction coefficients at different pressures for Ti-6Al-4V on which the modelling was based. The in-line reciprocating sliding test apparatus, as outlined in Chapter 3, was used with some minor changes. Figure 3.10 in Chapter 3 showed a schematic diagram of the test setup, which involves the vertical oscillation of a specimen with flat parallel sides by the Dartec servo-hydraulic testing machine, while two flat-surfaced pads with rounded edges perpendicular to the sliding direction are pressed against either side by a normal load P supplied by hydraulic pistons.

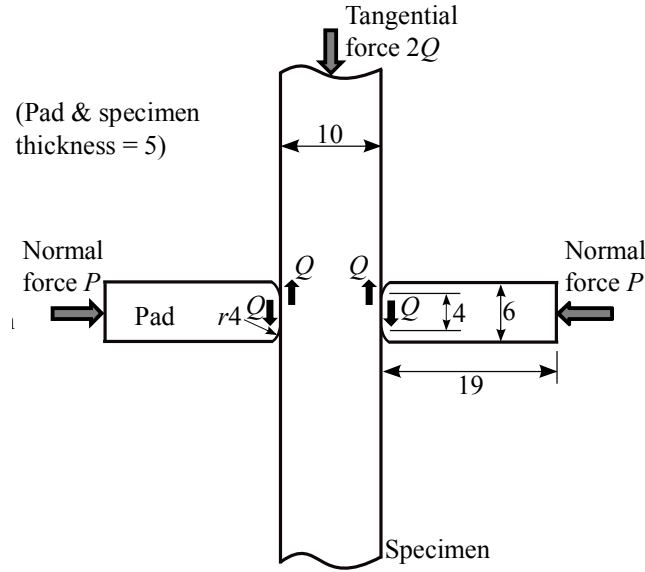


Figure 5.14: Reciprocating sliding friction test setup: pad and specimen geometry (dimensions in mm).

The pads and specimens were manufactured from Ti-6Al-4V, and the contacting surfaces were ground to a specified RMS areal roughness S_q of $1.5 \mu\text{m}$. The mechanical and chemical properties of the material used were given in Table 3.1. A schematic diagram of the geometry and loading conditions for the pads and specimen is shown in Figure 5.14. The pads used here differ slightly from those used in Chapters 3 and 4 as the nominal contact area is 20 mm^2 rather than 80 mm^2 . When sliding occurs, a friction force Q is developed at each contact, and a total tangential load of $2Q$ is measured by a 15 kN load cell mounted above the specimen. The friction coefficient for each contact pair is then calculated as Q/P . Specimen displacement is measured by a linear variable differential transformer (LVDT) as was shown in Figure 3.10. The analogue signals from the load cell and LVDT are processed as outlined in Section 3.3.3. To give more comprehensive results, four tests were carried out at average normal contact pressures p_m of 35, 70, 105 and 140 MPa. For these tests, the applied displacement amplitudes were ± 0.25 , ± 0.30 , ± 0.33 , and $\pm 0.37 \text{ mm}$, respectively. The displacement amplitudes were

slightly different for each test because higher normal pressures increase the level of displacement which occurs in the pre-sliding microslip part of the fretting cycle. Therefore, to give the same stroke length of gross sliding, slight adjustment of the applied displacement was required. Contact surfaces were cleaned beforehand with acetone, and tested in an unlubricated ‘dry’ environment with ambient conditions of temperature and humidity. The specimen oscillation frequency for each test was 0.5 Hz, and testing was carried out up to 5000 cycles. Figure 5.15 shows the resulting sliding friction coefficients during each test.

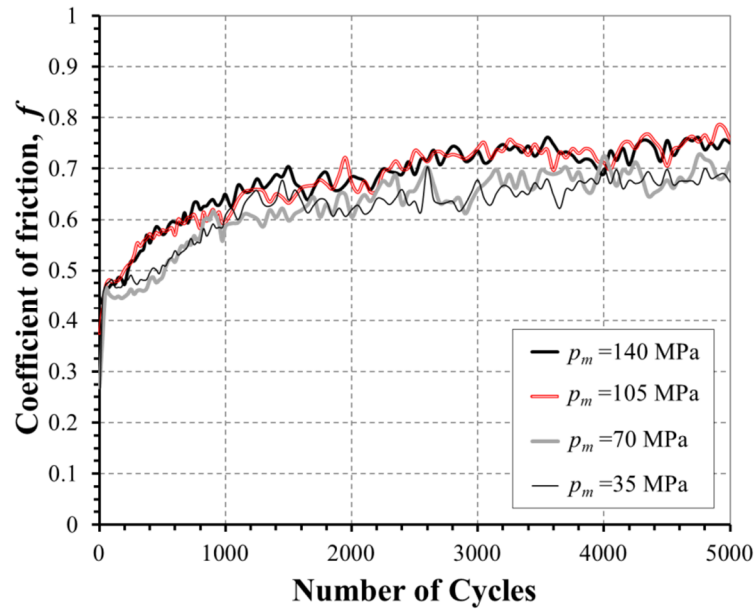


Figure 5.15: Average sliding friction results determined from reciprocating gross-slip tests with Ti-6Al-4V on Ti-6Al-4V. Results are shown up to 5000 test cycles for four mean contact pressures: 35, 70, 105, and 140 MPa.

It should be noted that the friction coefficient for each cycle was calculated as the average sliding friction over both the forward and backward strokes. In accordance with the Amontons-Coulomb description of friction, no significant dependence of the friction coefficient on the normal pressure is observed. Apart from the initial ‘run-in’ period, the sliding friction lies within the range 0.45–0.8 for all four tests. Comparison with Figure

5.10a and b reveals that this range lies within the central region of the results predicted by the friction modelling approach previously described.

The results show that models of the sliding interaction of elastic-plastic asperities incorporating ‘tangential adhesion’ at the interface can predict friction coefficients that are in general agreement with experimental measurements. It is clear from Figure 5.10a and b that, if the asperity interaction occurs with no ‘tangential adhesion’ ($\tau = 0$), then even the largest asperity overlap investigated, $\varepsilon = (30\%)r$, predicts a friction coefficient of just under 0.4. Given that the mean of the experimentally measured friction coefficients in Figure 5.15 takes on a steady value of about 0.7 in the second half of the test, it appears that some level of ‘tangential adhesion’ is required if elastic-plastic asperity models are to predict realistic friction coefficients. It is difficult to determine the required level of interface shear strength experimentally, but it is known to be highly dependent upon the condition of the surfaces. When tests are carried out *in vacuo*, and ‘clean’ metal-to-metal contact is allowed to occur, the measured friction coefficients are usually significantly higher than in ambient laboratory conditions [62]. However, metals tested in ambient conditions will inevitably contain an oxide layer, and an adsorbed gas layer containing water vapour and oxygen molecules [38]. It seems that the degree of ‘surface cover’ by such layers may influence the level of tangential adhesion. In Figure 5.15 the friction coefficient increases rapidly during the initial ‘run-in’ period, and then continues to increase until a steady value is reached. This is likely to be due to the breakdown of such layers and the consequent increase in adhesion between the surfaces. It seems that the modelling results presented in this work (Figure 5.10 and Figure 5.13) are more representative of metals sliding in air than of the higher friction often observed with tests carried out *in vacuo*, where metal-to-metal contact occurs. The key to modelling still higher friction coefficients than those obtained here may be the inclusion

of normal adhesion, as well as tangential adhesion, in the model. This is a subject for future work, but it is clear that the vast majority of engineering contacts occur under atmospheric conditions where the present friction predictions are in the same range as typical experimental values.

5.4 Conclusions

The finite element method has been used to develop a model of sliding friction based on the interaction of single elastic-plastic asperity junctions. The present work advances that of previous studies by catering for greater asperity overlaps, enforcing a ‘Bowden and Tabor’ type shear strength at the interface, and allowing for material failure within the asperity.

It was found that ‘implicit’ FE solution procedures encountered convergence difficulties when analysing the more demanding contact conditions and the ‘explicit’ solution procedure was therefore used. For the bulk of the modelling, cylindrical elastic-plastic asperities were used. The predicted friction coefficients were found to increase with asperity overlap and interface shear strength, and friction coefficients from the models incorporating a failure criterion were slightly higher than the corresponding model with infinite ductility (though both showed the same trend). The asperities were modelled as the titanium alloy Ti-6Al-4V, and unlike previous studies, the predicted friction coefficients lie within a range that would be representative of metals sliding in ambient conditions. A comparison of cylindrical and spherical asperities for one particular asperity overlap showed that each gave friction coefficients of similar magnitudes. Sliding friction experiments were carried out with Ti-6Al-4V, and the resulting friction coefficients were roughly in the mid-range of the predicted friction coefficients. Since

the predicted friction coefficients are a function of asperity overlap (and of interface shear strength), the results obtained were therefore used to develop a model of a rough surface interaction where a range of overlaps combine to give an overall friction coefficient. The friction coefficients obtained for the rough surface model were also within an appropriate range for sliding friction of metals in ambient conditions.

The predicted friction coefficients arise physically from both plastic dissipation of energy, and the dissipation of energy occurring at the interface in overcoming the adhesional shear strength. It seems clear from the present model that some level of tangential adhesion is required in order to give friction coefficients which are large enough to be representative of experimental measurements. In future work, it may be necessary to model still higher asperity overlaps, and also to incorporate the effect of normal adhesion. Developing a comprehensive statistical model of a rough surface interaction from this type of modelling would require experimental knowledge of the shear strength of real asperity junctions. Further, methods are required to allow for the interaction of two stochastic surfaces with the correct description of asperity heights and shape as determined from suitable metrology techniques. Another aspect that may require attention is the variation in mechanical properties across the surface grain structure of real metal interfaces. However, the main outcome of the present work is to show that realistic friction coefficients can be predicted from the sliding asperity interaction model based on the physical mechanisms of material plasticity and tangential interface adhesion.

Chapter 6

Contact stiffness of rough interfaces

6.1 Introduction

Up to now, the focus has mainly been on friction; this chapter looks at the other important parameter in characterising frictional joints: contact stiffness. Rough surface interfaces between contacting bodies inevitably introduce extra system compliance in both normal and tangential directions, giving rise to finite normal and tangential contact stiffness. Contact stiffness is the instantaneous rate of change of load (normal or tangential) on a contact with a corresponding change in relative normal or tangential position of the two surfaces in contact. It is the change in position of the two bodies due solely to the presence of the rough interface which results in the concept of a finite interfacial stiffness. Contact stiffness is particularly important in vibration analysis of jointed structures where the interface stiffness of the joints affect the vibration response of the overall structure. In order to improve the predictive techniques available to vibration engineers, the physics of contact stiffness first needs to be understood. The background and literature relating to experimental work on contact stiffness has already been outlined in Chapter 2, (Section 2.4 and 2.5.4). In Chapter 2, it became clear that remote

measurement techniques (e.g. an LVDT) cannot be used effectively to measure contact stiffness, thus methods which are more local to the interface are required. In this chapter, effort is focused on how best to measure and isolate both normal and tangential contact stiffness, as well as, on developing understanding of how contact stiffness is affected by various variables. The effect of variables such as: normal loading, tangential loading, and nominal contact area is investigated. Obtaining a ‘feel’ for the magnitudes of the interfacial stiffness values in comparison to the substrate material is also something which is undertaken.

The chapter falls broadly into two main sections: first, an investigation of tangential contact stiffness in gross-slip fretting is conducted where results from a finite element (FE) modelling approach are compared with experimental stiffness results obtained by Kartal *et al.* [94, 95] using digital image correlation (DIC), and second, contact stiffness measurements (normal and tangential) are carried out on non-fretting contacts subject to unidirectional loading. Two measurement techniques are used for this: digital image correlation and ultrasound. Both have been individually used before for measuring contact stiffness, but here, these vastly different techniques are compared for the first time using experiments designed to allow both measurements to be taken simultaneously in the same test. Since digital image correlation features throughout, it seems expeditious to outline the background and principles of this technique first.

6.2 Digital image correlation

6.2.1 Introduction to digital image correlation

A digital image of an object is a discrete intensity map of the light levels throughout that object – each separate intensity value being contained in an entity called a pixel. In a greyscale image, for example, each pixel can take on a discrete number of grey levels.

Digital image correlation is an optical non-contact technique that allows full-field displacements to be obtained by tracking the movement of features from an initial reference image through subsequent images after some loading event has occurred. Developed at the University of South Carolina in the 1980s [137-141], the technique is now widely used in experimental mechanics for measuring displacements on the planar surfaces of solid bodies. It has been used in fields such as plasticity [142], fracture mechanics [143], high-temperature deformation [144], determination of material properties [145], biomechanics [146] etc. More recently, the method has been employed to study smaller scale deformations using images derived from scanning electron microscopy (SEM) [147, 148] and atomic force microscopy (AFM) [149]. The technique is more robust and versatile than many other optical techniques such as holography interferometry, speckle interferometry and moiré interferometry since it requires only a standard digital camera, sufficient light and enough randomly distributed surface features on the specimen. Extra light can easily be provided by a light source, and for surfaces without the required amount of natural contrast, a speckle pattern can be sprayed on. The resolution is limited only by the digital camera resolution and the magnification used.

6.2.2 Principles of DIC

The post-processing is mostly carried out by commercial software, but the general procedure is as follows. The image is firstly divided into sub-regions since it is easier to track the grey levels of multiple pixels rather than that of a single pixel. Consider the square undeformed sub-region S in Figure 6.1 containing $(2M + 1)(2M + 1)$ pixels with its centre at $C_S(x_0, y_0)$ and an arbitrary pixel at $P_S(x_j, y_k)$ located at x and y distances of Δx and Δy respectively from the centre.

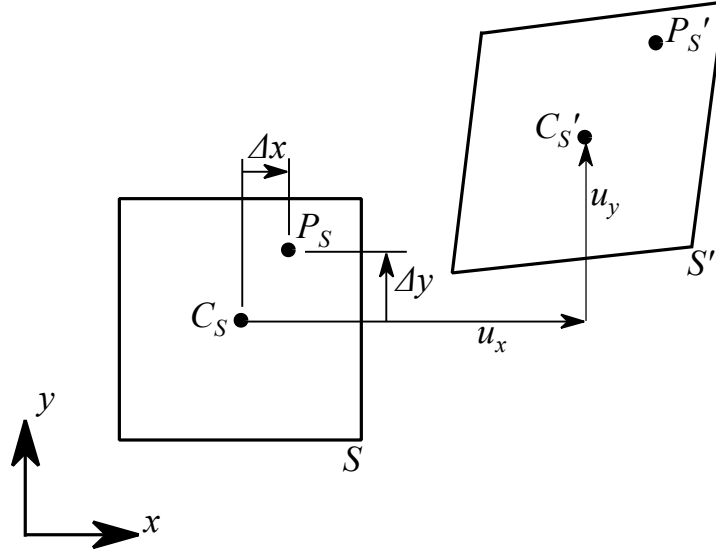


Figure 6.1: Schematic diagram showing a deformed and undeformed subregion.

After some deformation, the sub-region moves to S' and the two points move to $C_S'(x_0', y_0')$ and $P_S'(x_j', y_k')$ with the centre point C_S having displacements u_x and u_y . These displacements and their accompanying gradients with respect to x and y can be found by finding the position of the best match for the reference image grey level intensity pattern in the deformed image. This is usually achieved through some variation of either a cross-correlation or least-squares approach. The simplest of these are the basic cross-correlation function (CCF) and least-squares function (LSF):

$$\text{CCF}\left(u_x, u_y, \frac{\partial u_x}{\partial x}, \frac{\partial u_x}{\partial y}, \frac{\partial u_y}{\partial x}, \frac{\partial u_y}{\partial y}\right) = \sum_{j=-M}^{j=+M} \sum_{k=-M}^{k=+M} [I_u(x_j, y_k) I_d(x_j', y_k')], \quad (6.1)$$

and

$$\text{LSF}\left(u_x, u_y, \frac{\partial u_x}{\partial x}, \frac{\partial u_x}{\partial y}, \frac{\partial u_y}{\partial x}, \frac{\partial u_y}{\partial y}\right) = \sum_{j=-M}^{j=+M} \sum_{k=-M}^{k=+M} [I_u(x_j, y_k) - I_d(x_j', y_k')]^2, \quad (6.2)$$

where, I_u is the grey level intensity function for the reference image before deformation and I_d is the corresponding intensity in the deformed image. Assuming an affine mapping or shape function (i.e. uniform displacement gradients or strains are assumed within the sub-region), the coordinate of P_S after deformation will be related to the coordinate before deformation by:

$$x_j' = x_j + u_x + \frac{\partial u_x}{\partial x} \Delta x + \frac{\partial u_x}{\partial y} \Delta y, \quad (6.3)$$

and

$$y_k' = y_k + u_y + \frac{\partial u_y}{\partial x} \Delta x + \frac{\partial u_y}{\partial y} \Delta y. \quad (6.4)$$

Depending on which approach is used (i.e. cross-correlation or least-squares), and in conjunction with Equations (6.3) and (6.4), by finding the displacement components and gradients which either maximise the cross-correlation (Equation 6.1), or minimise the least squares sum (Equation 6.2), a good estimate for the actual displacement and deformation can be found for the sub-region. This is usually performed using some variation of Newton-Raphson non-linear optimisation. However, an initial guess is required for this technique; therefore, a two-step approach is usually used. Approximate displacements with only integer level pixel accuracy are determined in the first step by assuming zero displacement gradients and using a coarse searching method. In the second step, the initial guess for the displacements is input as the starting point to the Newton-Raphson method, however, since points may displace to sub-pixel locations, interpolation to produce a continuous intensity function of grey levels over region S' in the deformed image must be performed. Schemes such as bilinear interpolation or bicubic interpolation can be used here. However, a higher-order interpolation scheme

(e.g. bicubic spline or biquintic spline interpolation) is recommended (Schreier *et al.* [150] and Knasss *et al.* [151]) since they provide higher registration accuracy and better convergence of the algorithm than the simple interpolation schemes do. The result should be accurate displacement and gradient predictions for the sub-region, and by analysing many sub-regions, a full-field map of displacements and strains can be computed by the method.

6.2.3 Experimental considerations

Any DIC analysis is dependent on the quality of the captured images, and even though DIC has fewer experimental requirements than some of the other optical techniques, there are some issues which require consideration. The properties of the surface features are important: the pattern must contain sufficiently small features which are randomly distributed with good contrast and illumination. Stable and constant illumination during testing is also important as any change will alter the grey level intensity upon which the subsequently computed displacements will depend. In addition, the plane of the measurement surface and that of the camera sensor should remain parallel and at a constant distance apart during the experiment; any out of plane movement of specimen or camera can lead to the introduction of artificial displacements. Since DIC works with displacements in terms of pixels, calibration is required to determine the size of a pixel and the results will naturally depend upon this calibration. When the images contain a complete view of some part of the specimen geometry whose dimension is known, calibration is often easily performed by simply dividing this distance by the corresponding distance in pixels. Often though, the images come from an interior region

with no obvious dimension to compare to (as in the present work), and in this case, calibration can be carried out by performing a displacement of a known amount and dividing the distance moved by the equivalent number of pixels, or by independently measuring the distance between two distinct surface features using, for example, an optical microscope and dividing by the pixel equivalent. In the current work, both of these calibration methods were used and each gave a very similar result for the pixel size. Two recent reviews which provide a more detailed overview of the use of digital image correlation are given by Pan *et al.* [152] and Hild and Roux [153].

6.2.4 Obtaining contact stiffness using digital image correlation

To obtain measurements of contact stiffness using the DIC technique, local images with sufficient resolution containing the contact interface are required at each load increment. For ease of analysis, the contact interface should be either vertical or horizontal so that the x and y directions correspond to the normal and tangential directions, and the interface should be roughly in the centre of the image. Using suitable DIC software such as Imetrum Video Gauge [154] (used in the present work) or LaVision's DaVis StrainMaster software [155] (used in Kartal *et al.* [94, 95]), relative displacement between two measurement points (one on each side of the interface) can be determined. These measurement points will, of course, be the centre points of DIC sub-regions. Load can then be plotted against relative displacement, and the slope of the curve gives the tangent stiffness. A profile of stiffness versus distance between measurement points can then be obtained – though it is difficult to obtain data from measurement points spaced apart by less than about 100 μm . This is because the interface will have a finite thickness, and also, some distance into the bulk material is required to allow sufficiently

sized sub-regions for image correlation. Both normal and tangential contact stiffness can be obtained depending on which load (normal or tangential) was incremented during testing.

6.3 Tangential contact stiffness in gross-slip fretting

6.3.1 Introduction

Kartal *et al.* [94] were the first to measure tangential contact stiffness using digital image correlation. Their measurements were based on the correlation of digital images taken at a region local to the contact interface during gross-slip fretting tests on Ti-6Al-4V. These tests are similar to the in-line fretting tests discussed in Chapters 3 and 4 as flat pads with rounded edges perpendicular to the sliding direction were pressed against an oscillating flat-sided specimen making a nominal contact area of 80 mm². Figure 6.2 shows a schematic diagram of the contact geometry. This was followed in Kartal *et al.* [95] by measurements (using the same testing arrangement) where the nominal contact area was scaled to 20 mm². The results from [94] and [95] are profiles of tangential contact stiffness versus distance between DIC measurement points at various test cycle numbers. It is clear that tangential contact stiffness results obtained by DIC, such as these, may be affected to some extent (however small) by bulk material compliance between the two measurement points, as well as, by the interface compliance itself. However, it is not straightforward to determine the relative significance of these two factors from experimental DIC results. Neither is it easy to say how the measured values might differ if a different contact area were used. To explore these issues, a finite element (FE) model of the Ti-6Al-4V pad-specimen experiment described above was developed using ABAQUS software [128].

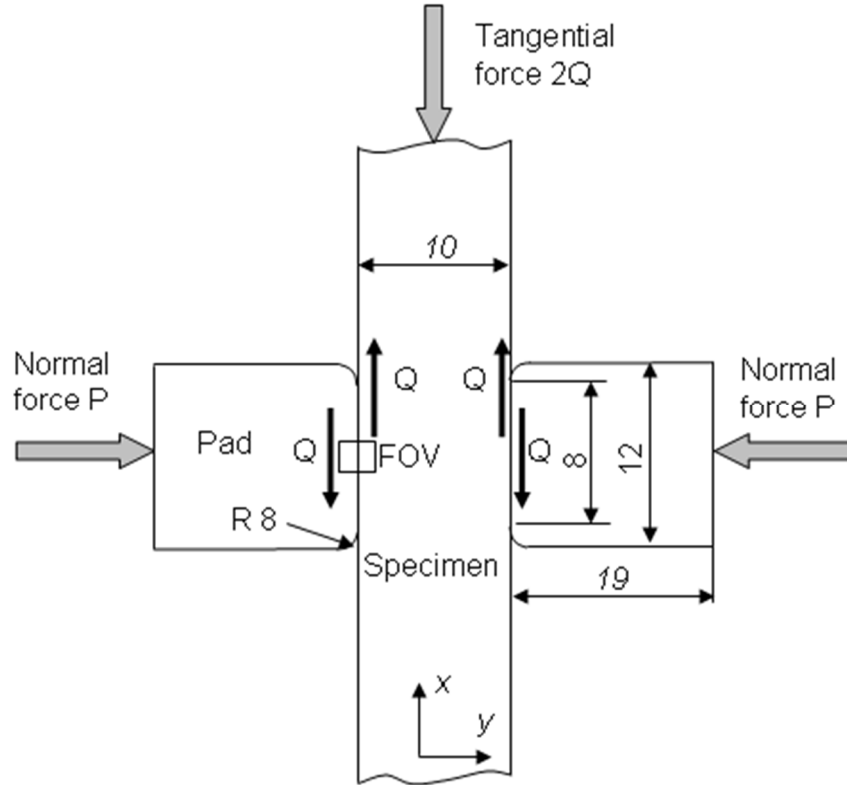


Figure 6.2: Contact geometry for the tangential contact stiffness tests (using DIC) carried out during gross-slip fretting in Kartal *et al.* [94]. The field of view (FOV) for obtaining digital images is shown.

6.3.2 Model setup

The FE model (Figure 6.3) is a smooth contact one and contains no interface compliance; thereby, allowing isolation of the tangential stiffness contribution due to bulk deformation between the DIC measurement points.

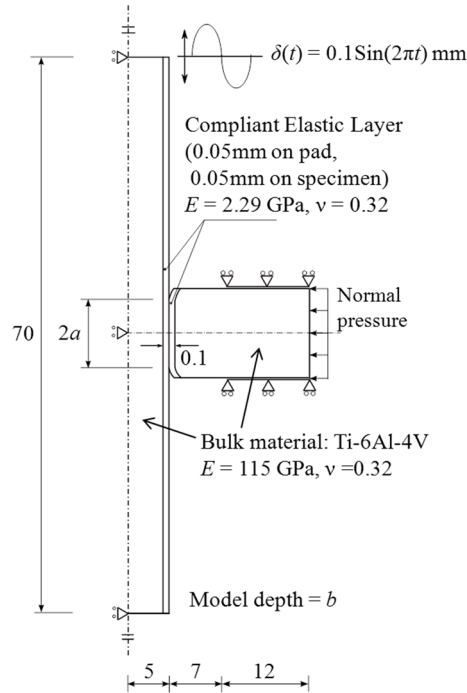


Figure 6.3: Finite element model of the Ti-6Al-4V pad-specimen experiments: schematic diagram showing the overall model parameters. Dimensions in mm.

As a result of symmetry in the test, only half of the arrangement was modelled and a linear elastic material behaviour was used. As in the actual test, the pad was held against the specimen with an average normal contact pressure of 70 MPa over a nominal contact area of 80 mm^2 while the specimen was given a quasi-static vertical oscillation. The Lagrange multiplier friction formulation was used with a friction coefficient of 0.6 chosen for consistency with the experimentally observed value (see Chapter 3). The tangential contact stiffness was determined when the contact was fully stuck after the point where the motion reverses (see Figure 1.2b). Results were derived using the relative tangential displacement of pairs of points/nodes situated at various distances (in the range 0-1 mm) from the interface: the points in each individual pair being equidistant from the interface (on opposite sides) and situated along the horizontal pad-specimen centreline. The mesh, shown leftmost in Figure 6.4, consisted of 2D plane strain linear quadrilateral elements (CPE4R), with a region of triangular elements (CPE3) situated

outside the measurement zone on the pad (three other models are also shown in Figure 6.4 and these are discussed later). Reduced integration was employed to avoid the quadrilateral elements inside the measurement zone being too stiff in bending. Figure 6.3 also shows the inclusion of a compliant interfacial elastic layer, which may be introduced to represent the compliance of the rough interface. The compliant layer was used to tune the FE model to match the experimental stiffness results by altering the Young's modulus of the layer. However, initially the model was analysed as homogeneous bulk Ti-6Al-4V in the absence of the compliant layer with $2a = 8$ mm, and $b = 10$ mm (see Figure 6.3).

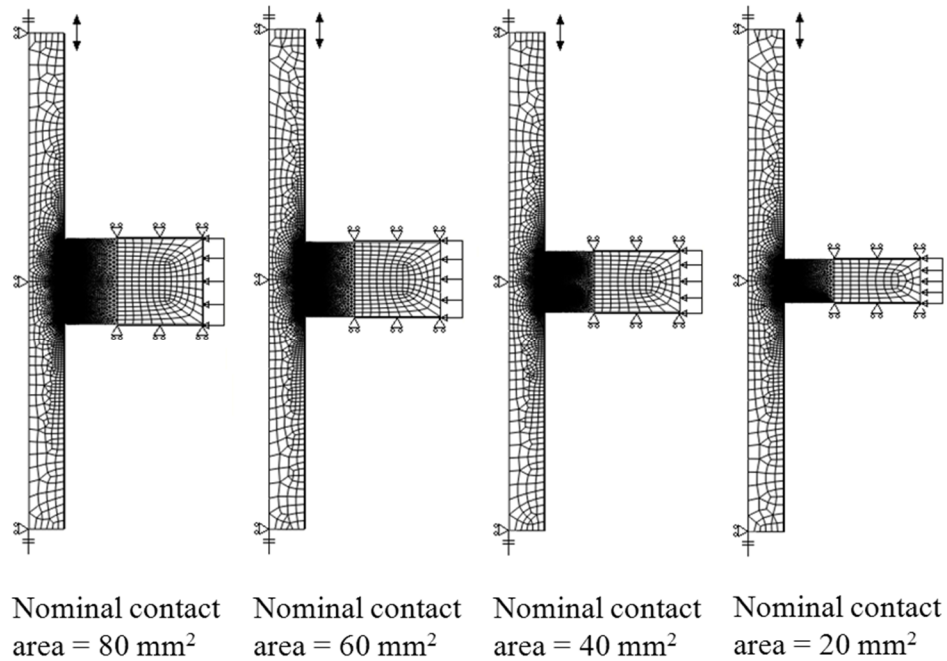


Figure 6.4: Individual FE models for the four different nominal contact areas.

6.3.3 Results and discussion

Before embarking on a discussion of results, it is important to point out that all results in this chapter apply to a single contact (i.e. they are based on the tangential load Q), whereas results in Kartal *et al.* [94, 95] apply to both contacts (i.e. they are based on a

tangential load of $2Q$ – the load measured at the load cell). The result of the FE analysis is shown in Figure 6.5 where it is compared with an experimentally determined DIC result from Kartal *et al.* [94] (taken at 500 cycles). As we would expect (since this particular FE model does not account for the rough interface), the FE stiffness is greater. However, the difference is quite significant, and while the FE result is very dependent on the distance between measurement points, the experimental result is not so sensitive to position. This suggests that the interface is much more compliant than the bulk material. We can attempt to isolate the actual ‘interface stiffness’ $K_{\text{Interface}}$ since the DIC stiffness K_{DIC} contains both interface stiffness and bulk stiffness, while the FE stiffness K_{FE} contains only bulk stiffness. Considering the experimental DIC stiffness as the series sum of the ‘interface stiffness’ plus the ‘bulk stiffness’, we obtain:

$$\frac{1}{K_{\text{DIC}}} = \frac{1}{K_{\text{FE}}} + \frac{1}{K_{\text{Interface}}}. \quad (6.5)$$

Using Equation (6.5), the isolated ‘interface stiffness’ was calculated from the DIC and FE data shown in Figure 6.5. The result is also plotted in the same figure, and it can be seen that it is very similar to the DIC data. This confirms that the interface is considerably more compliant than that of the bulk material located between the measurement points, and that the measured DIC stiffness is so little affected by bulk compliance in this case that it may be considered almost equivalent to the interface stiffness.

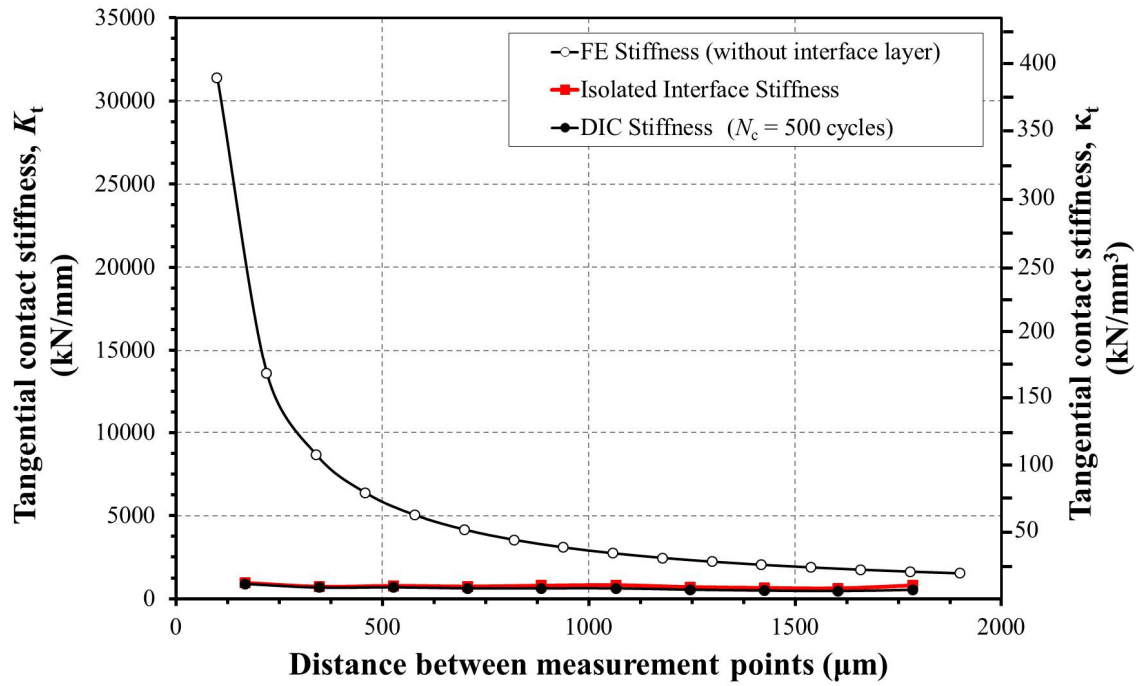


Figure 6.5: Comparison of tangential contact stiffness from the FE result (without an interface layer) to an experimental DIC result from the ‘smooth ground’ tests in Kartal *et al.* [94]. Also shown is the ‘isolated interface stiffness’ (Nominal contact pressure: 70 MPa, Nominal contact area: 80 mm^2).

If this compliant interface is incorporated into the FE model, some additional information can be obtained (e.g. about the relationship between tangential contact stiffness and nominal contact area). Therefore, a thin compliant linear elastic layer was incorporated on each of the bodies in the model (50 μm each on the pad and specimen) to give a total layer thickness of 100 μm (see Figure 6.3). Although the layer thickness is much greater than the initial surface roughness ($S_q \approx 1.5 \mu\text{m}$), the roughness values usually increase significantly during a fretting-wear test (see Chapters 3 and 4). Wear scars from Ti-6Al-4V pads and specimens tested under the same conditions are shown in Figure 3.14 in Chapter 3, and in these, the flatness of the worn surfaces exceed 100 μm after 5600 cycles of testing. The elastic modulus of the ‘interface layer’ was then adjusted until the FE stiffness results matched the experimental DIC results. It will be appreciated that the required modulus value will, of course, be a function of the chosen layer thickness –

though the precise layer thickness is not important as long as its properties have been chosen correctly so that the model correctly simulates the experimental contact stiffness versus distance profile. The result of this approach is shown in Figure 6.6 where the tuned FE model stiffness values are now seen to agree with the experimental DIC results.

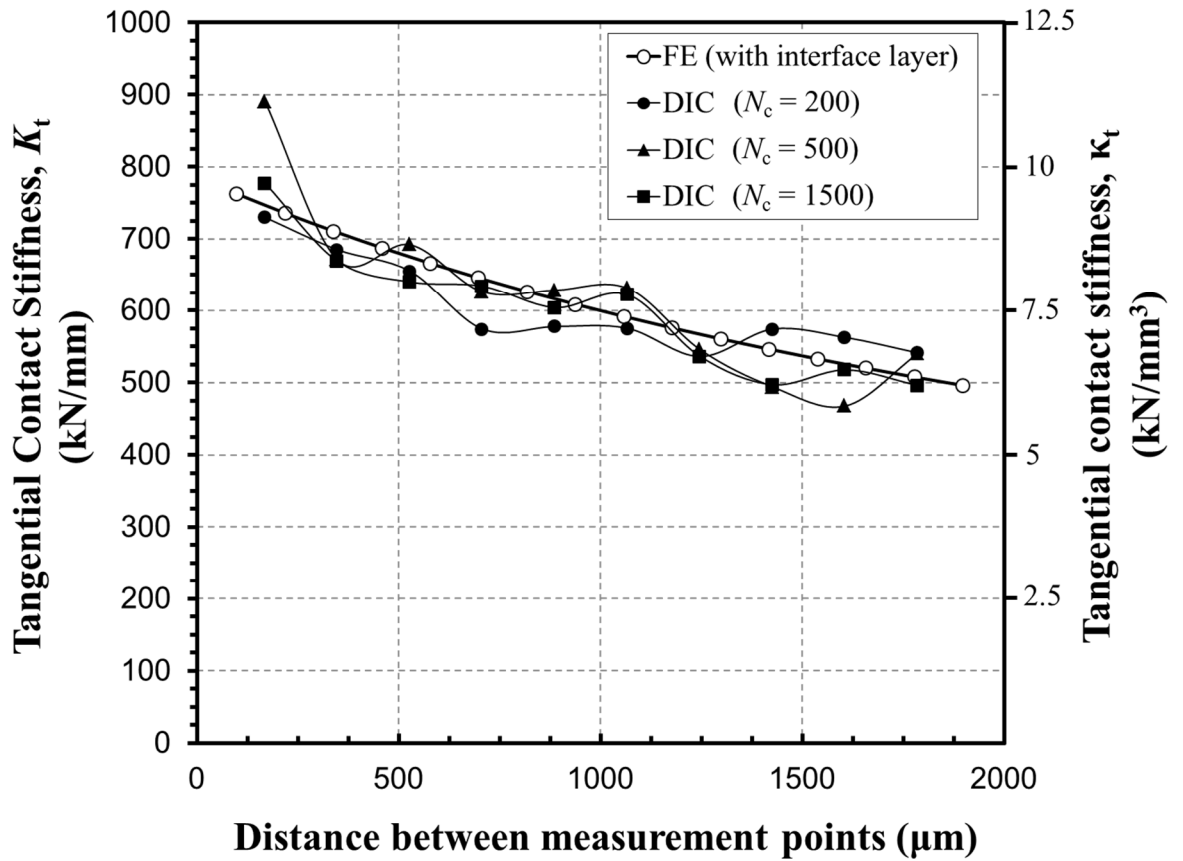


Figure 6.6: Comparison of tangential contact stiffness from the ‘interface layer’ FE result with experimental DIC results for three different test cycle numbers, N_c , for the Ti-6Al-4V ‘smooth ground’ case in Kartal *et al.* [94] (Nominal contact pressure was 70 MPa).

The simulated result also matches the gradual fall in stiffness as distance from the interface increases. For the layer thickness chosen here, the required layer modulus was 2.29 GPa. This suggests, in fact, that the interface behaves like a 100 μm layer of polystyrene on a Ti-6Al-4V substrate. Again, this further assists in explaining why the DIC measurement is dominated by interfacial compliance and is scarcely affected by bulk

compliance. This high interface compliance is probably attributable to factors such as: surface roughness; oxide layers; and probably most importantly, compacted wear debris. Sevostainov and Kachanov [156] envisaged the contact of rough surfaces as producing a mix of voids and contact regions where the voids as well as local slip at the micro-contacts would serve to reduce the interface shear stiffness. It must be remembered that the present modelling analysis is using results measured many cycles into fretting-wear tests where wear debris will play a major role. However, Berthoud and Baumberger [87] also reached the conclusion that interface compliance is dominant after performing static unidirectional tests on aluminium alloy (AG5) and poly-methyl-methacrylate (PMMA) contacts. Later in this chapter (Section 6.4), equivalent non-fretting Ti-6Al-4V contacts will be investigated during the first cycle of monotonic loading.

If the surfaces were ideally smooth, the contact stiffness would scale with area in a manner determined by the appropriate elasticity solution; however, a further consequence of a highly compliant interface on a stiff substrate is that tangential contact stiffness is likely to be proportional to nominal contact area. Therefore, three further FE models were developed (using the interface layer properties already determined) to give nominal contact areas of 80, 60, 40 and 20 mm². Figure 6.4 shows the geometry and mesh for each of the four different models. It should be noted that the aspect ratio of the contact area $2a/b = 0.8$ (see Figure 6.3) was maintained constant for all four cases (so as to match the ratio used in experiments). The vertical dimensions of the pad, as well as the edge radii, were also scaled to maintain geometric similarity. Both the horizontal length dimensions of the pad as well as the specimen dimensions were held fixed. Figure 6.7 shows the variation in tangential contact stiffness with nominal contact area obtained from the models. In deriving these FE stiffness values, relative displacement was obtained from measurement points with a spacing of 265 µm to allow comparison with

experimental results. Two experimental results for a contact area of 80 mm^2 (Kartal *et al.* [94]) and 20 mm^2 (Kartal *et al.* [95]) are added to the plot for comparison. Figure 6.7 shows that the FE model predicts a linear relationship between tangential contact stiffness and nominal contact area, while the change in stiffness between the two experimental data points agrees reasonably well with the FE prediction as area is increased from 20 to 80 mm^2 . Proportionality with contact area suggests that stiffness results can be normalised by contact area in order to compare results from tests with different contact areas. Again, it should be remembered that results in this section are based on analysis of measurements derived from fretting-wear tests [94, 95] and may not apply elsewhere. In fact, in Section 6.4.7, it will be shown that this dominance of the interface over the contact stiffness is dependent upon normal contact pressure and may not apply at higher pressures (even in the fretting wear case).

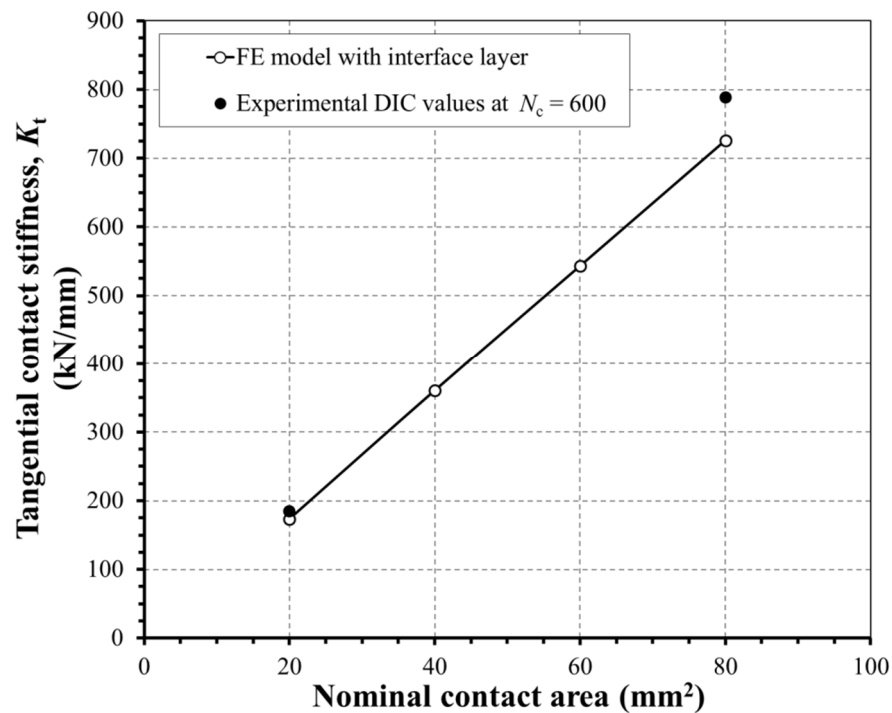


Figure 6.7: Variation of tangential contact stiffness with nominal contact area predicted by FE modelling including experimental data points at 80 mm^2 and 20 mm^2 taken from Kartal *et al.* [94] and [95], respectively (normal contact pressure was 70 MPa and distance between measurement points was $265 \text{ }\mu\text{m}$ for both FE and DIC).

6.4 Measuring contact stiffness: a comparison of digital image correlation and ultrasound techniques

6.4.1 Introduction

In the literature, the contact stiffness of rough interfaces has been separately measured using digital image correlation [94, 95, 97] and ultrasound [98, 100, 106, 108-114], however, these techniques have never been compared under identical conditions (or otherwise). Here, unidirectional (i.e. non-fretting) ‘first load-up’ tests are performed on contacts where both DIC and ultrasound are used simultaneously to measure contact stiffness. The comparison is undertaken for both normal and tangential contact stiffness. The two techniques are very different; one uses an ultrasound signal reflected from the interface itself, while the other is based on the correlation of side-on images of the interface. The displacements induced by the ultrasound technique are of the order of Ångstroms, whereas, those involved in a DIC measurement will be at the micron scale.

Therefore, by assessing the level of agreement in trends and magnitudes between the techniques, the experiments aim to learn more about precisely what each technique is measuring. The tests involve measuring contact stiffness while incrementing either the normal or the tangential load, and this also allows some other aspects of the physics of contact stiffness to be investigated. The digital image correlation technique has already been outlined; the theory of ultrasound measurement of rough interface contact stiffness is now outlined before the specific experiments are discussed.

6.4.2 Theory of contact stiffness measurement using ultrasound

Ultrasound is simply a cyclic displacement wave whose frequency is sufficiently high ($>20,000$ Hz) that its sound is inaudible to humans. It has long been used to detect the presence of internal flaws which reflect ultrasound and in medical imaging etc. In the 1970s and 80s, papers by Kendall and Tabor [98], Tattersall [99], and Baik and Thompson [103] began to outline how ultrasound might be used to measure the contact stiffness of rough interfaces. When an ultrasound wave is incident normal to a perfect boundary between two materials (1 and 2) the reflection coefficient R (proportion of the incident wave which is reflected) will depend on the acoustic impedances of the material on each side of the boundary according to:

$$R = \frac{Z_1 - Z_2}{Z_1 + Z_2}, \quad (6.6)$$

where, Z_1 and Z_2 are the acoustic impedances (the product of the material density and the speed of sound in the material) on either side of the boundary (i.e. of bodies 1 and 2). Since the speed of sound itself depends on stiffness and density, it is these two properties which determine the transmission/reflection characteristics. If both materials are identical, then all of the incident wave will be transmitted. On the other hand, if the acoustic impedance on one side of the interface is much lower, then most of the incident wave will be reflected. This is generally the case with metal-air interfaces (for example, at an aluminium-air boundary, the reflected wave has 99.99995 % the amplitude of the incident wave [106]) However, if the boundary is a rough contact interface (between two bodies) then this introduces an additional compliance which will give an increased reflection coefficient. A rough contact will contain regions of intimate contact where ultrasound is transmitted as well as air gaps where it will be reflected. It has been shown [98] that if the wavelength of the incident wave is large compared to the sizes of the air

gaps in the plane of the interface, then the proportions of transmitted and reflected ultrasound will be independent of the exact shapes and sizes of the air gaps and will depend only on the stiffness of the interface. Kendall and Tabor [98], Tattersall [99], and Baik and Thompson [103] all contributed to the development of a quasi-static spring model (Figure 6.8) to allow this interface stiffness to be isolated. The model treats the interface as a set of linear springs in parallel giving a total stiffness per unit area of κ ; it is presently outlined here as put forward by Tattersall [99].

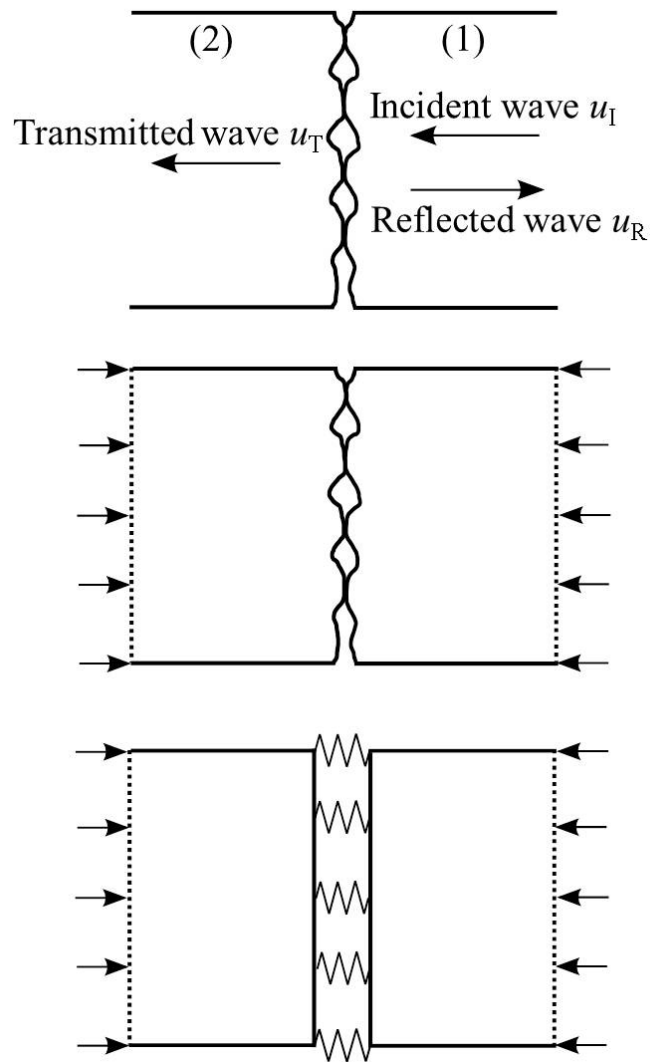


Figure 6.8: The spring model for the interaction of rough interfaces with ultrasound.

If we let the incident acoustic displacement wave travelling in the positive x -direction with angular frequency ω be:

$$u_I = \exp\{i\omega(t - x/c_1)\}, \quad (6.7)$$

and the reflected and transmitted wave displacements be:

$$u_R = A \exp\{i\omega(t + x/c_1)\}, \quad (6.8)$$

and

$$u_T = B \exp\{i\omega(t - x/c_2)\}, \quad (6.9)$$

where, A and B are complex coefficients giving the amplitude and phase of the reflected and transmitted waves respectively and c_1 and c_2 are the wave velocities, then the displacements in bodies 1 and 2 are given respectively by:

$$u_1 = \exp\{i\omega(t - x/c_1)\} + A \exp\{i\omega(t + x/c_1)\}, \quad (6.10)$$

and

$$u_2 = B \exp\{i\omega(t - x/c_2)\}. \quad (6.11)$$

The variation in the ‘gap’ between the two surfaces is then given by:

$$\Delta u = u_2(0, t) - u_1(0, t), \quad (6.12)$$

which is positive if the gap increases. The stress fields in bodies 1 and 2 are respectively:

$$\sigma_1 = E_1 \frac{\partial u_1}{\partial x} = -i\omega Z_1 \exp\{i\omega(t - x/c_1)\} + i\omega Z_1 A \exp\{i\omega(t + x/c_1)\}, \quad (6.13)$$

and

$$\sigma_2 = E_2 \frac{\partial u_2}{\partial x} = -i\omega Z_2 B \exp\{i\omega(t - x/c_2)\}. \quad (6.14)$$

And the stress in the boundary due to the spring force is given by:

$$\sigma_b = \kappa(u_2(0, t) - u_1(0, t)), \quad (6.15)$$

that is:

$$\sigma_b = \kappa(B - A - 1) \exp(i\omega t). \quad (6.16)$$

The boundary condition is that the three stresses σ_1 , σ_2 and σ_b are all equal at the boundary because the stress is transmitted instantaneously from body 1 to body 2 through the springs. On putting:

$$\sigma_1(0, t) = \sigma_2(0, t) = \sigma_b(t), \quad (6.17)$$

and cancelling the common exponential factor, we obtain:

$$i\omega Z_1(A - 1) = -i\omega Z_2 B = \kappa(B - A - 1), \quad (6.18)$$

which can be solved for A and B ; A is in fact the reflection coefficient R as the incident wave had unit amplitude. Thus:

$$R = A = \frac{Z_1 - Z_2 + i\omega(Z_1 Z_2 / \kappa)}{Z_1 + Z_2 + i\omega(Z_1 Z_2 / \kappa)}. \quad (6.19)$$

It can be seen that if the interface is infinitely rigid ($\kappa = \infty$) this expression reverts to Equation 6.6 which describes the reflection coefficient at a perfect interface. When $\kappa = 0$, there is no connection between the two parts giving $R = 1$. If, as in the case of the present work, the material on both sides of the interface is the same, then Equation 6.19 reduces to:

$$R = \frac{1}{\sqrt{1 + \left(\frac{2\kappa}{\omega Z}\right)^2}}. \quad (6.20)$$

Solving explicitly for contact stiffness gives:

$$\kappa = \frac{\omega Z}{2} \sqrt{\frac{1}{R^2} - 1}, \quad (6.21)$$

and substituting $\omega = 2\pi f_u$ and $Z = \rho c$ gives:

$$\kappa = \rho c f_u \pi \sqrt{\frac{1}{R^2} - 1}, \quad (6.22)$$

where, f_u is the ultrasound frequency, ρ is the material density, and c is the wave speed in the material. Equation 6.22 can be used for the calculation of both normal stiffness and shear stiffness by simply using the appropriate transducer type and wave speed c (i.e. longitudinal or shear). Therefore, from experiments in which the incident wave and the reflected wave from a rough surface are recorded, the reflection coefficient R can be found, and Equation 6.22 used to determine the contact stiffness.

6.4.3 Overview of the test setup

In order to compare ultrasonically measured contact stiffness with contact stiffness measured by DIC, a test was conceived whereby both measurements could be taken from the same contact and during the same test. The test bore many similarities with the previous in-line friction tests carried out in Chapters 3 and 4, and with the tests carried out to measure tangential contact stiffness during gross-slip fretting in Kartal *et al.* [94, 95] as the same servo-hydraulic tensile testing machine was used and flat-and-rounded pads in contact with a flat were again used. Figure 6.9 shows a schematic diagram giving details of the contact geometry and the components specially designed to accommodate the test. As before, the pad surfaces were flat, rectangular in shape, with two parallel rounded edges and two 90 degree edges, and made contact with a prismatic specimen

with flat sides. Essentially, capturing digital images of a field of view (FOV) local to the contact interface during loading allows contact stiffness to be measured by DIC, while, an ultrasonic transducer mounted to the back face of the pads allows concurrent contact stiffness measurement of the same interface by ultrasound. Also included in the design is the facility to control the tangential load Q and the normal load P on each of the two contact interfaces.

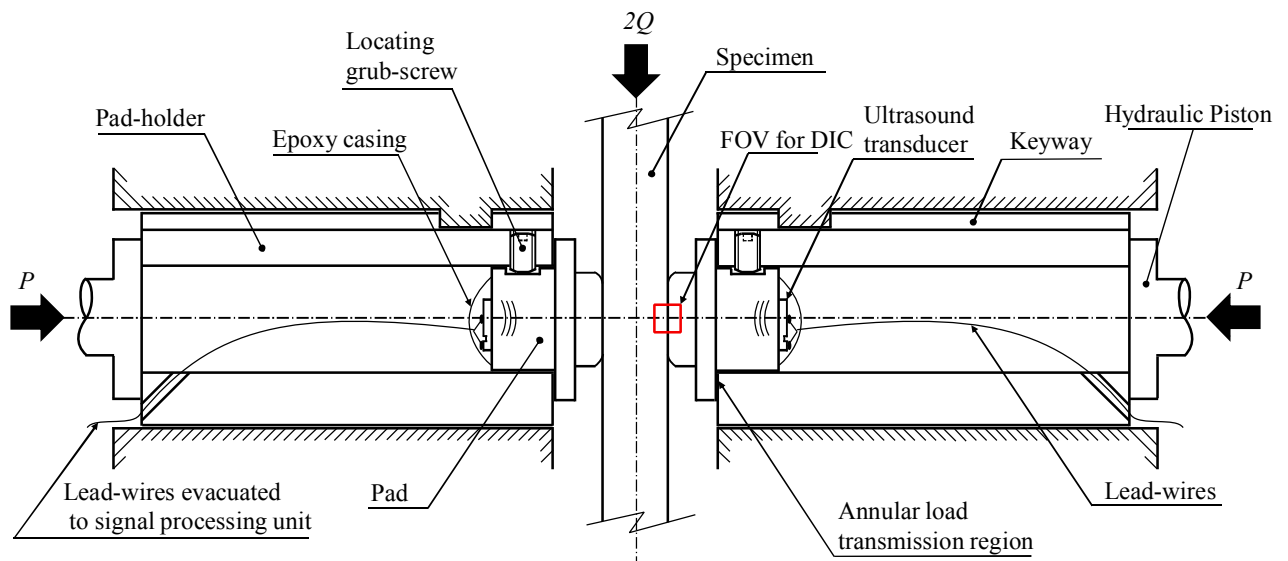


Figure 6.9: Schematic diagram of the specimen and pads shown with accompanying local test setup for DIC-ultrasound testing.

In previous tests of this type in Chapters 3 and 4, the normal load P was transferred to a pad via its back-face; however, in this case no loading of the back face could be permitted as the ultrasound transducer would be adversely affected. Thus, new types of pads and pad-holders were designed so that the loading could be transmitted to an annular region of the pad (Figure 6.9) by the front face of the pad-holders. The new pad-holders were also designed to evacuate the ultrasound lead wires from the ‘work-zone’ as shown in Figure 6.9. Figure 6.10 shows a stand-alone photograph of a specimen in contact with two pads. The specimen mounting thread, annular pad loading region, ultrasonic

transducer, and ultrasound lead wires can be seen. Pads and specimens were designed so that two sizes of contact area could be tested: 80 mm^2 and 50 mm^2 . For the 80 mm^2 area, the contact patch was a rectangle of length 8 mm and depth 10mm while for the 50 mm^2 interface, geometric similarity was maintained and the contact rectangle was 6.32 mm by 7.90 mm. The distance between the two contacts was always 10mm.

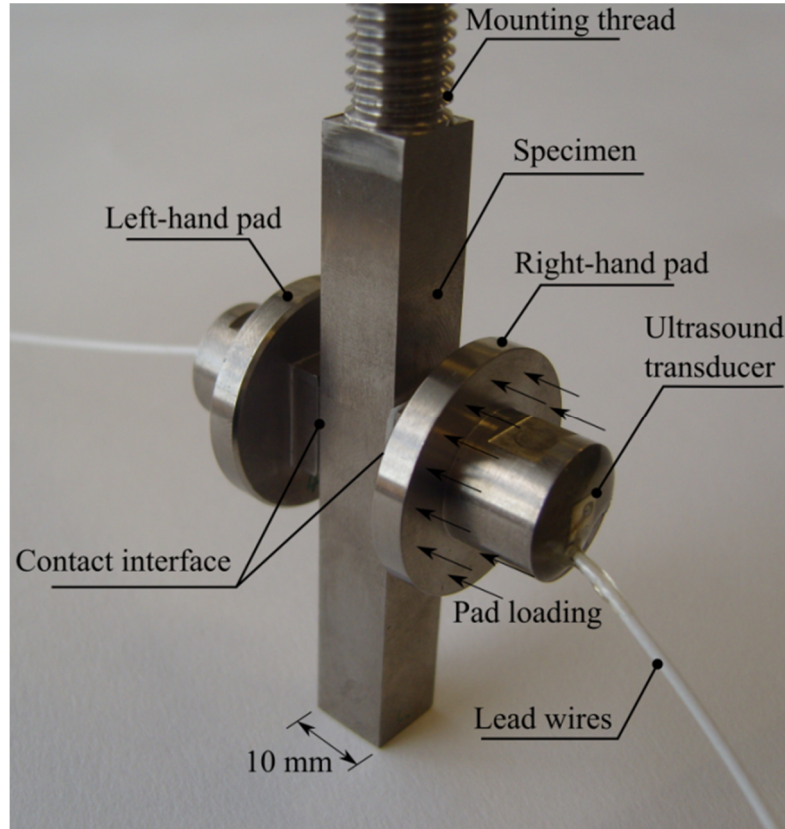


Figure 6.10: Photograph of a specimen in contact with two instrumented pads.

The pads and specimens were manufactured from the aerospace titanium alloy Ti-6Al-4V. Both pad and specimen surfaces were smooth ground to a nominal area based root mean square (RMS) roughness S_q of $2.3 \mu\text{m}$. The mechanical and chemical properties of the Ti-6Al-4V used here are the same as those given in Table 3.1 of Chapter 3. The test equipment used to provide the required constraints, to apply and measure the loads (P and Q), and to capture the digital images is shown in Figure 6.11. The details shown earlier

in Figure 6.9 are housed inside a work-holding block which allows the pad-holders to slide in the direction normal to the contact interface to accommodate the normal loading which is applied by the hydraulic pistons shown (the value is determined by a pressure gauge). Figure 6.11 also shows the hydraulic actuator and load cell used to apply and measure tangential load, respectively. The tangential force load cell was calibrated for a full-scale range of 0-15 kN with a resolution of 1N.



Figure 6.11: Photograph of the Dartec servo-hydraulic tensile testing machine showing various features of the test setup including the camera and Questar microscope used for obtaining digital images.

6.4.4 Experimental procedures for ultrasound measurements

Since the aim of the experiments is to measure both normal and tangential contact stiffness, both longitudinal and shear ultrasonic transducers were used. These transducers were commercial, broadband, piezo-ceramic transducers supplied by Tribosonics, UK

and were used as both transmitter and receiver (i.e. pulse-echo mode). The centre frequency for the longitudinal and shear transducers was approximately 9.6 and 2.3 MHz, respectively (except for the shear transducers used in tests 4, 5, 6, and 10 – see Section 6.4.6 for test numbering – where a 5.2 MHz approx. shear transducer was used). The bandwidth for the longitudinal and shear transducers (measured to a 6 dB reduction in amplitude) was approximately 4–6 MHz and 2–3 MHz, respectively. Transducers for the 80 mm² contact area pads were 4 mm × 4 mm in size and those for the 50 mm² contact were 3.16 mm × 3.16 mm; scaling the transducers in this way was carried out so that the signals would encounter the same fraction of the contact width in each case. An electrical connection to the top and bottom transducer faces is required to apply a voltage signal across the transducer which is then converted to a mechanical vibration by the piezoelectric property of the transducer material. The transducers were of the ‘wrap around electrode’ type so that both wires could be soldered directly to the top face even though one of these connections makes electrical contact with the bottom face only. The transducers were glued to the centre of the back face of the pads (which had been smooth ground in preparation) using Tribobond adhesive (a cyanoacrylate based glue). Two lead wires (Filotex 50 VMTX miniature PTFE coaxial cable) were then soldered to the transducer which was then encased in epoxy to protect the connections and the transducer.

Since the setup incorporates two contacts, a shear transducer was always placed on one pad and a longitudinal on the other. Only the right-hand contact was used for DIC, however, so whichever stiffness (normal or tangential) was being measured by DIC at the right-hand contact was also measured by the appropriate ultrasonic transducer (i.e. longitudinal or shear) on the right-hand pad while the left-hand pad contained a transducer of the other type. Two ultrasonic pulser-receivers (UPR) were used to

generate voltage pulses to actuate the transducers. Equation 6.22 requires that the density and wave speeds for the material be known in order to calculate contact stiffness values. A density of 4420 kg/m^3 supplied by the manufacturers TIMET [129] was used. The speed of both longitudinal and shear waves in the material were determined experimentally using the pads by sending a pulse to the surface and back (a known distance) and recording the time taken. The measured longitudinal and shear wave speeds in the material were 6148.28 m/s and 3092.8 m/s , respectively. Since the spring model (Equation 6.22) relates reflection coefficient to stiffness, the main experimental task is to determine the reflection coefficient when load is applied to the surface. The reflection coefficient is simply the reflected wave divided by the incident wave. The incident wave, however, can be obtained simply by recording the reflected wave from a metal-to-air interface since the acoustic impedance of air is very low in comparison to a metal such as Ti-6Al-4V, and hence, the reflected wave in this case is equivalent to the incident wave. This ‘reference signal’ was recorded before testing when the pads were not in contact with the specimen. Load was then applied to the interface and the reflected signal was recorded for each load increment. The incident and reflected signals (amplitude versus time) were then amplified by an ultrasound pulser-receiver (UPR), digitised by a digital oscilloscope, and passed to a computer for processing. A fast Fourier transform (FFT) was subsequently implemented using LabView software to obtain the frequency spectrum (amplitude versus frequency). The programme used the average of 50 successive pulses in obtaining the frequency spectrum. At each frequency, the amplitude of the reflected wave was then divided by the amplitude of the incident wave to obtain reflection coefficients which were then used with Equation 6.22 to determine contact stiffness. The average across the bandwidth of the transducer (upper 6 dB) was then used to calculate the contact stiffness for each load increment.

6.4.5 Experimental procedures for digital image correlation measurements

In order to measure contact stiffness using digital image correlation, images of the side-on face of the pad and specimen which include the vertical contact interface are required during loading. To obtain the digital images during the tests a PixelINK PL-B741U CMOS 1.3 mega-pixel monochrome digital camera [157] was used. This camera has a simple USB 2.0 interface and is based on the Cypress IBIS5 CMOS global shutter progressive scan sensor with a 2/3" optical format. This gave images of the field of view (FOV) with 1280×1024 pixels. In order to obtain high resolution, a Questar QM1 long distance microscope [158] with a variable zoom and a working distance of 150–355 mm was used as the lens to magnify a region local to the contact. The Questar (with the camera mounted) was fixed to a precision translation stage in order to allow focusing of the FOV (Figure 6.11). It was positioned as close as possible to the pad/specimen 'side-surface' to give maximum magnification. The resulting FOV (which was positioned at the centre of the right-hand interface) was approximately $1.3 \text{ mm} \times 1 \text{ mm}$ giving a pixel size of approximately $1.02 \text{ }\mu\text{m}$.

The imaged surfaces of the pad and specimens were not specially prepared as the original grinding operation gave sufficient features for image correlation. Illumination of the FOV was provided by a fibre-optic illuminator which gave co-axial illumination of the FOV. Images were recorded initially and also with each loading increment. The digital image correlation analysis of the images was then carried out using Imetrum Video Gauge software [154]. Crucially this software allows the placing of targets on either side of the interface and can immediately output the relative displacement which occurs

between a pair of targets. Figure 6.12 shows an image recorded from one of the tests showing all 20 targets used for analysis. The rectangular targets were 80×250 pixels in size while the four square targets shown were 80×80 pixels.



Figure 6.12: A sample digital image of the pad-specimen interface from one of the tests showing the targets (T1 – T20) used for DIC analysis with the Imetrum Video Gauge software.

The main relative displacement measurement used for most of the analysis which follows was derived as an average of the relative displacements associated with each of the five pairs of targets positioned closest to the interface (i.e. targets 1&2, 13&14, 15&16, 17&18 and 19&20). Five further target pairs (i.e. targets 3&4, 5&6, 7&8, 9&10, and 11&12) were also positioned on the images in order to study the relationship between contact stiffness and distance from the interface. Both normal and tangential contact stiffness can be calculated as the software allows either normal or tangential relative displacement to be evaluated depending on which of the forces (normal or tangential) were incrementally increased during a test. Plots of load (from a single contact) versus relative displacement were then produced and curves were fitted to the data (using

Matlab) so that the tangent stiffness values could be obtained as the slope of the curves at various points during loading. A rational polynomial function (5th degree/5th degree) was found to best fit the data. Stiffness values were subsequently normalised by the nominal contact area to give stiffness per unit area (i.e. the same units as the ultrasound stiffness values given by the spring model in Equation 6.22). The resolution of the software quoted by Imetrum is one thousandth of a pixel size. Therefore, given that the pixel size is 1.02 μm with the present setup, the quality of the results will be limited only by noise introduced to the images themselves by small vibrations/movements during testing.

6.4.6 Testing programme

A total of ten tests were carried out and details of these are given in Table 6.1. Tests were carried out for ‘first load-up’ of the contacts and not over many cycles as with the previous fretting tests outlined in Chapters 3 and 4. This is because the ultrasound result depends on a reference signal, and if surface wear is allowed to occur this would invalidate the initial reference signal. Tests 1–6 had an 80 mm² contact area, while tests 7–10 had a contact area of 50 mm². In tests 1, 2, 7 and 8, the normal load was held constant to maintain a mean contact pressure of 70 MPa while the tangential load was incrementally increased up to the point of slip. This allowed the tangential contact stiffness to be measured simultaneously at the right-hand contact by both ultrasound and DIC while normal stiffness was recorded by ultrasound (only) at the left-hand contact, thus permitting investigation into the effect of increasing tangential load upon normal and tangential contact stiffness. In tests 3, 4, 5, 6, 9, and 10, no bulk tangential load was applied and the normal load was increased allowing comparison of measurements of normal contact stiffness by both techniques at the right-hand contact (except for tests 5

and 6 where DIC was not used); tangential contact stiffness was measured by ultrasound only at the left-hand contact. This allowed enquiry into the effect of increasing normal pressure upon both contact stiffness components. Loading was increased manually (by the hydraulic actuator of the test machine in the case of tangential load and by a hydraulic pump in the case of normal loading). At each load increment, both a digital image and a sequence of reflected ultrasound pulses were recorded (the ultrasound pulses being recorded separately at different times from both the left and right hand contacts). Tests were carried out on dry, unlubricated contacts with ambient pressure and temperature conditions.

Table 6.1: Outline of the testing programme.

Test No.	Contact area (mm ²)	Loading sequence	Measurement technique and parameter measured
1 & 2	80	$p_m = \text{constant} = 70 \text{ MPa}$, Q then increased	DIC (for κ_t) + Ultrasound (for κ_t and κ_n)
3 ³ & 4	80	$Q = \text{constant} = 0$, p_m increased to 200 MPa	DIC (for κ_n) + Ultrasound (for κ_t and κ_n)
5	80	p_m increased to 70 MPa,	Ultrasound only (for κ_t and κ_n)
6	80	p_m increased to 200 MPa	Ultrasound only (for κ_t and κ_n)
7 & 8	50	$p_m = \text{constant} = 70 \text{ MPa}$, Q then increased	DIC (for κ_t) + Ultrasound (for κ_t and κ_n)
9 & 10	50	$Q = \text{constant} = 0$, p_m increased to 200 MPa	DIC (for κ_n) + Ultrasound (for κ_t and κ_n)

6.4.7 Results and discussion

Figure 6.13 shows an example of the type of displacement profile observed in the digital images at the (initially horizontal) pad-specimen centreline as tangential load was applied at constant normal pressure. Some deformation occurs in the pad and specimen, but the large discontinuity in displacement at the interface is due to the compliance caused by the

³ When p_m reached 200 MPa, tangential load Q was then applied and κ_t was measured by DIC.

rough surface interaction. It can be seen that relative displacement between points closest to the interface is dominated by this interface contribution.

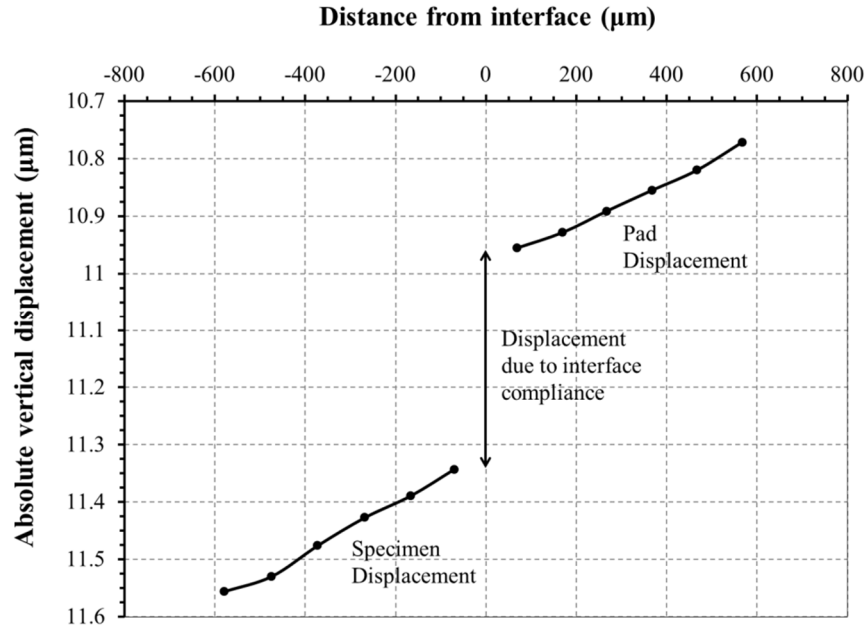


Figure 6.13: Absolute displacement profile of points initially on the horizontal pad-specimen centreline recorded in Test 1 when $Q/fP = 0.48$ as the tangential load is increased.

Figure 6.13 shows tangential displacement of the pad and specimen for just one particular value of tangential load (corresponding to $Q/fP = 0.48$) in Test 1. Figure 6.14, however, shows the full variation of relative tangential displacement (for the measurement points closest to the interface) with applied tangential load for each of the four tests where tangential load was increased monotonically towards slip (Tests: 1, 2, 7, & 8). The average coefficient of static friction in these four tests was 0.24 – less than the values determined for Ti-6Al-4V in Chapter 3 which reached a steady average of about 0.6 in gross-slip fretting tests. This is because, as we have seen previously, a ‘run-in’ period is required to remove initial surface layers, but in the present tests only the ‘first load-up’ is considered so that the friction remains low. The data in Figure 6.14 show a distinct trend even though the relative displacements are very small (increasing to a maximum of between 2–3 microns near to the point of slip). It can be seen also that the force-

displacement curves here are distinctly non-linear and suggest plastic deformation. Since all the results in this section are from the ‘first load-up’ of the contacts, this plastic flow is to be expected. Also shown (in red) are the rational curve fits which were used to determine tangent stiffness (slope of the curve fits) at each load increment.

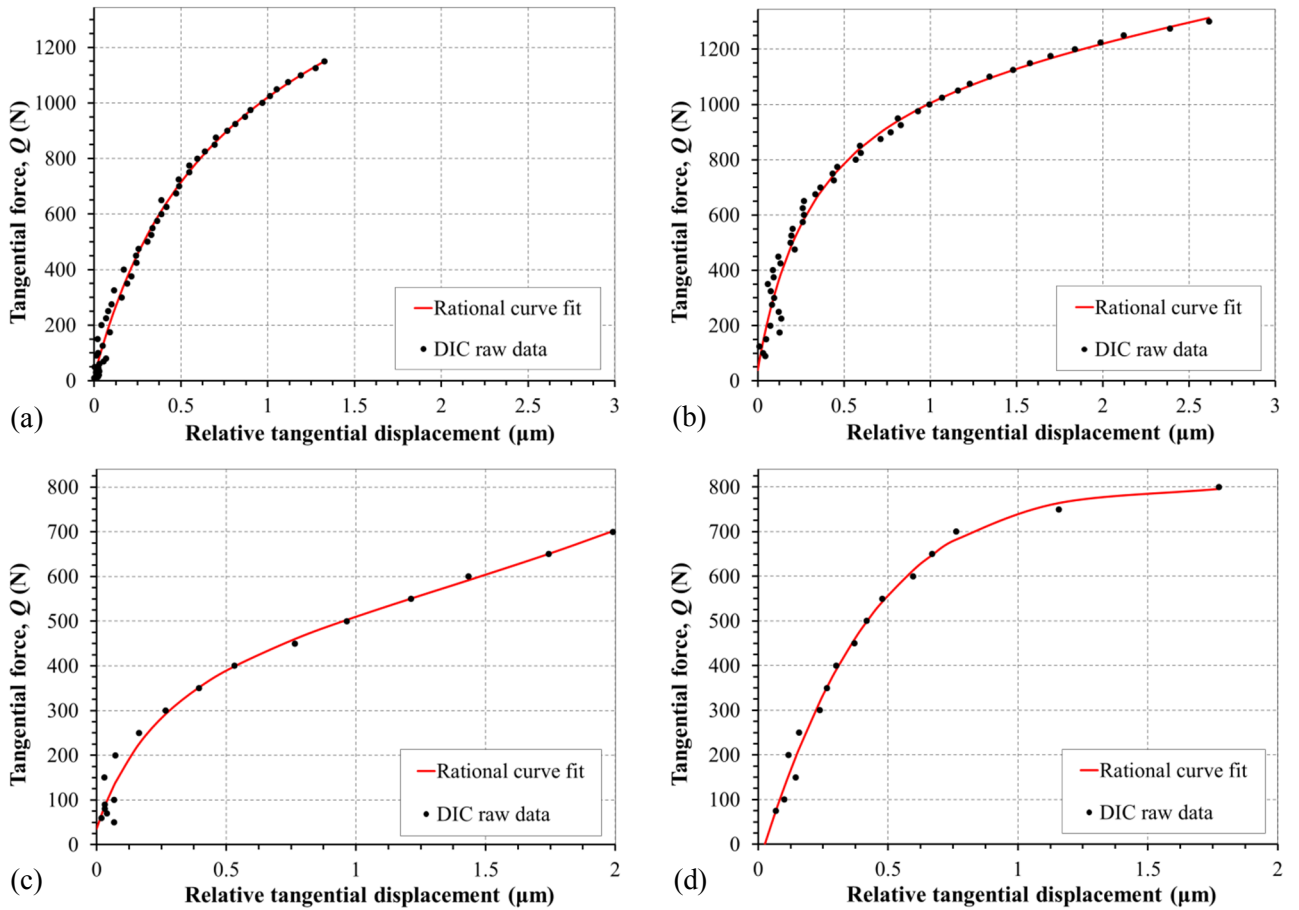


Figure 6.14: Plots of tangential force, Q , versus relative tangential displacement derived from DIC for: (a) Test 1, (b) Test 2, (c) Test 7, and (d) Test 8. Distances (in μm) between measurement points for calculation of relative displacement were: 137 for (a), 145 for (b), 102 for (c), and 142 for (d). Rational curve fits to the data are also shown. $p_m = 70$ MPa in each case.

The resulting variation in tangential contact stiffness (obtained by DIC) with tangential load for these four tests is shown in Figure 6.15 together with the corresponding ultrasound stiffness values derived simultaneously from the same contact. It can be seen that as tangential loading on the contact is increased, the contact stiffness as measured by DIC decreases. The ‘ultrasound stiffness’, on the other hand, does not decrease correspondingly, and instead either remains almost constant or increases somewhat.

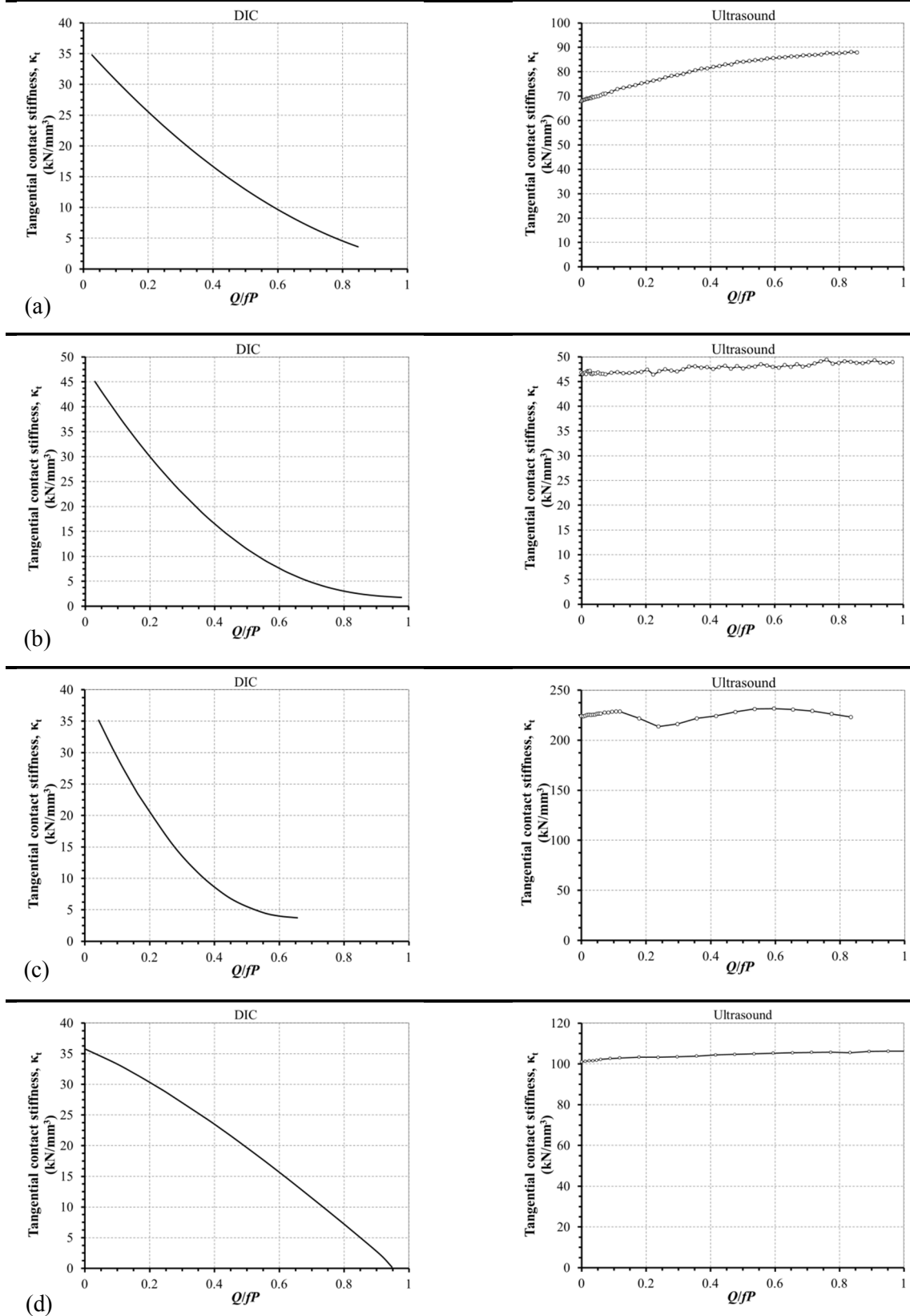


Figure 6.15: Comparison of DIC (left) with ultrasound (right) for the variation of tangential contact stiffness (per unit area) with tangential load for: (a) Test 1, (b) Test 2, (c) Test 7, and (d) Test 8. Distances (in μm) between measurement points for calculation of relative displacement for DIC were: 137 for (a), 145 for (b), 102 for (c), and 142 for (d). $p_m = 70\text{MPa}$.

A likely reason for this difference was first hinted at by Kim *et al.* [111]. They pointed out that since rough contacts quickly become elasto-plastic as loading is increased, and because ultrasound applies a local loading-unloading cycle centred on the static stress, the ‘ultrasonic interfacial stiffness’ is actually measuring the local unloading stiffness. This will be an elastic stiffness since the stress state is taken back inside the yield locus by the small scale perturbation (of the order of Ångstroms) supplied by the ultrasound. This local unloading stiffness does not ‘feel’ the plastic softening of the asperities, and therefore, even though the tangent stiffness of the loading curve decreases due to plastic yielding of asperities, the local unloading stiffness would not be expected to change as the tangential load is increased. This is explained schematically in Figure 6.16.

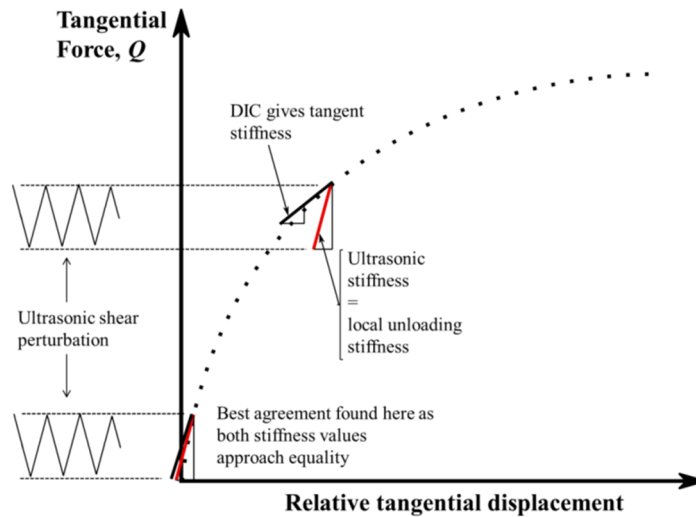


Figure 6.16: Schematic showing the difference between tangential contact stiffness derived from DIC (the tangent stiffness) with that derived from ultrasound (the local unloading stiffness).

Therefore, the closest agreement between ultrasound and DIC can be expected at the onset of micro-slip ($Q = 0$) where tangent stiffness and unloading stiffness are expected to be approximately equal. The effect discussed here, though proposed by Kim *et al.* [111] in a discussion of normal contact stiffness, had not previously been observed experimentally. This is because efforts have tended to focus, not on the variation of

tangential contact stiffness with tangential load, but instead on the variation with normal load where the effect could be expected to be masked by the significant increase in the real area of contact which occurs during normal loading. This would tend to cause both the ‘ultrasonic stiffness’ and the tangent stiffness of the loading curve to increase with load. In any case, to confirm the effect, an independent technique to measure the tangent stiffness of the loading curve concurrently with the ultrasound measurements is required. The use of DIC in conjunction with ultrasound in the present work has permitted this comparison to be made, thus allowing the difference to be observed. A moderate increase in ultrasonically measured shear stiffness is seen to occur in Figure 6.15a, b, and d. This may be due to a slight increase in contact area with tangential load owing to junction growth as first observed by McFarlane and Tabor [65]. This increase in contact area is small in comparison to that which would be expected in normal loading. The evolution of the normal contact stiffness with tangential load measured by ultrasound in Tests 1, 2, 7, and 8 is given in Figure 6.17. This stiffness is also relatively unaffected by tangential load and good repeatability can be seen across the four independent tests.

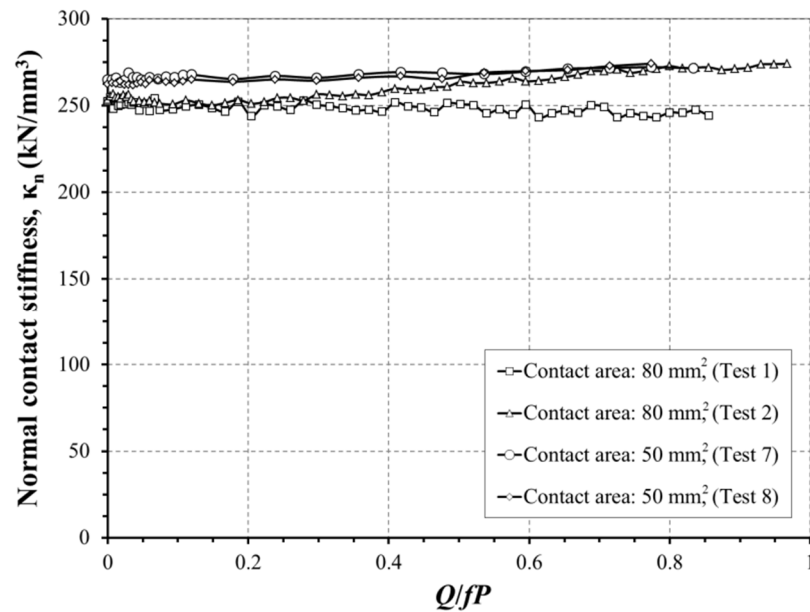


Figure 6.17: Variation of normal contact stiffness (per unit area) with tangential load measured by ultrasound in Tests 1, 2, 7, and 8. Mean normal contact pressure is 70 MPa in each test.

Figure 6.18 shows normal force versus relative normal displacement for the four tests where both DIC and ultrasound were used to measure normal stiffness (i.e. Tests: 3, 4, 9, and 10). Again, despite the small displacements involved, a definite trend is apparent in the raw data. The rational curve fits from which tangent stiffness values were obtained are also shown. Figure 6.18a shows that good agreement is observed when the ‘Imetrum DIC results’ are compared to those from a different software package (DaVis StainMaster by LaVision). These two packages use different algorithms for the calculations of displacements and their agreement in Figure 6.18a increases confidence in the results.

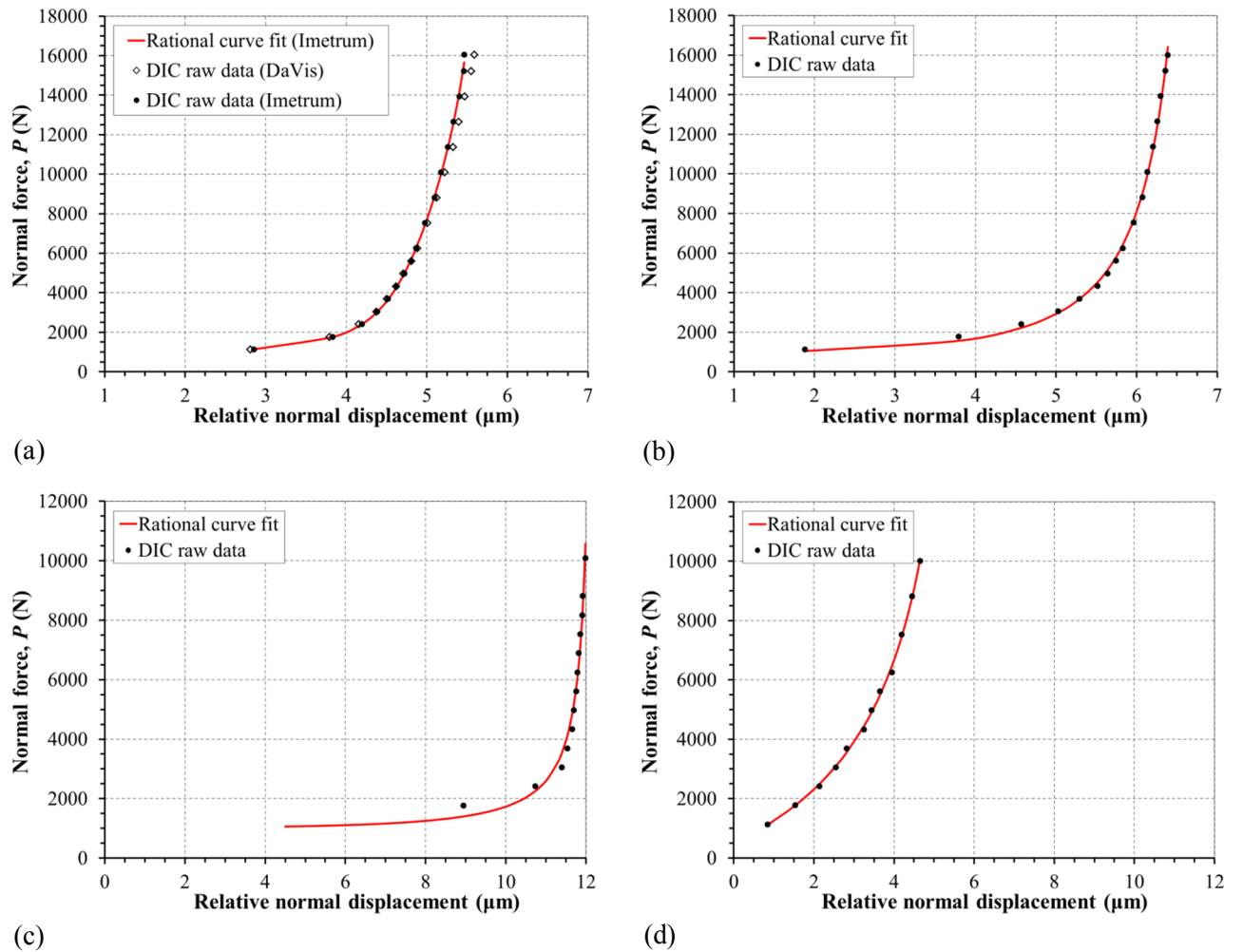
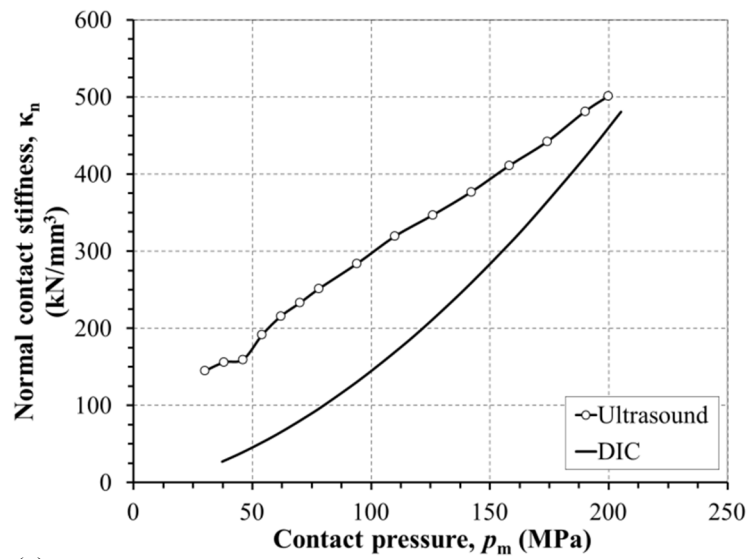
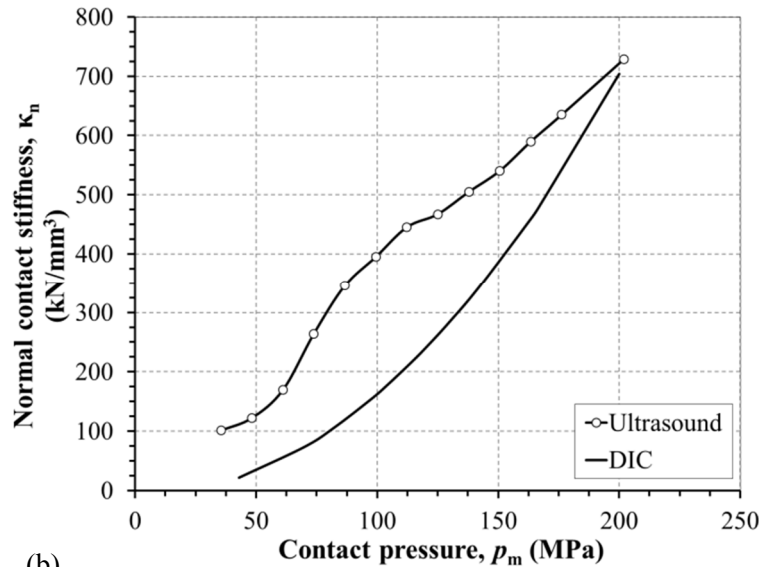


Figure 6.18: Plots of normal force, P , versus relative normal displacement derived from DIC for: (a) Test 3, (b) Test 4, (c) Test 9, and (d) Test 10. Initial distances (in μm at $P = 0$) between measurement points for calculation of relative displacement were: 126 for (a), 135 for (b), 111 for (c), and 150 for (d). Rational polynomial curve fits to the data are also shown.

Figure 6.19 compares normal contact stiffness derived from DIC analysis (slopes of the loading curves) with that derived from ultrasound for tests 4 (contact area: 80 mm^2) and 9 (contact area: 50 mm^2). As expected, both DIC and ultrasound stiffness increase in this case owing to the increase in real area of contact with normal loading. The DIC stiffness is again greater than the ultrasound stiffness but this may be due to the fact that the ultrasound technique does not ‘feel’ the plastic softening of the asperities as it measures local unloading stiffness rather than tangent stiffness as discussed above.



(a)



(b)

Figure 6.19: Comparison of DIC with ultrasound for the variation of normal contact stiffness (per unit area) with normal contact pressure: (a) Test 4, and (b) Test 9. Initial distance between measurement points for DIC (i.e. before loading) were $135 \text{ }\mu\text{m}$ for (a) and $111 \text{ }\mu\text{m}$ for (b).

In Figure 6.20, ultrasonically measured tangential contact stiffness is plotted against normal contact pressure for tests 3, 4, 6, and 9. Again, as expected, the stiffness values increase with normal load and reasonable repeatability can be observed between the four tests. The stiffness values in Figure 6.19 and Figure 6.20 appear to show fairly linear variations with contact pressure unlike the results from Gonzalez-Valadez *et al.* [113] shown in Chapter 2 (Figure 2.10) where the stiffness obeys a concave-down square root like variation with a gradually reducing gradient as the normal pressure is increased. This may be because the material being tested here has a high yield strength value (1000 MPa) and may not display the softening behaviour until pressures greater than 200 MPa are reached. Linearity is in fact predicted by the elastic ‘Nowell model’ of tangential contact stiffness (outlined in [95]) which is based on the application of Mindlin’s solution [115] for tangential loading of Hertz contacts to the statistical rough surface model of Greenwood and Williamson [64] using an exponential distribution of asperity heights.

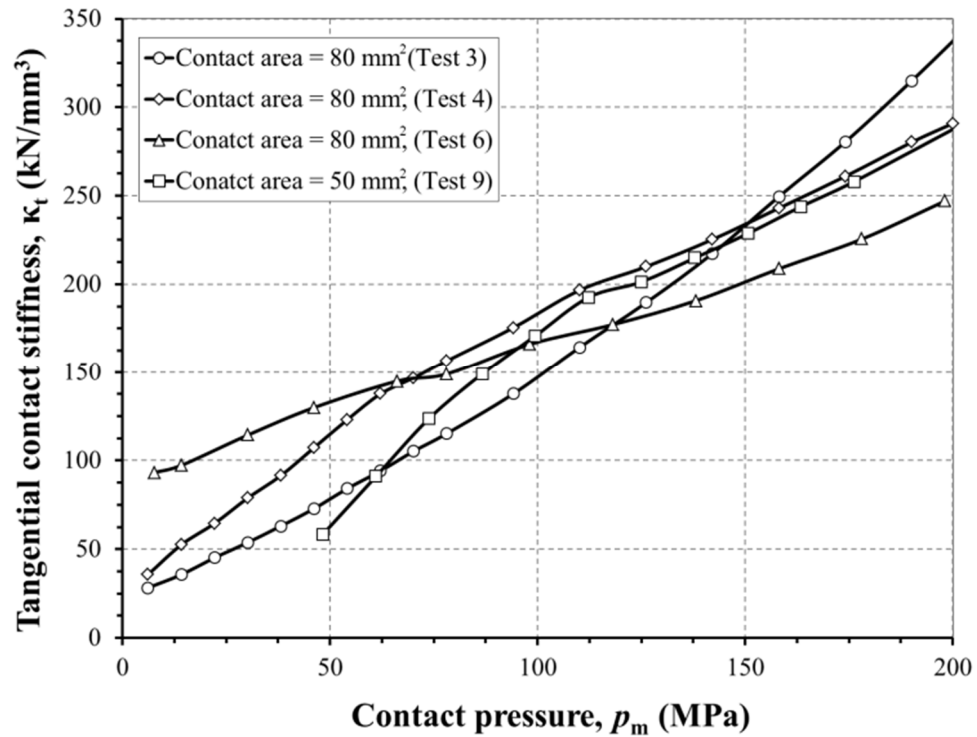


Figure 6.20: Variation of tangential contact stiffness per unit area (ultrasound only) with p_m for Tests 3, 4, 6, & 9.

For the ultrasound measurements in both Figure 6.17 and Figure 6.20 (and for the DIC results in Figure 6.15), there is no discernible difference between the stiffness per unit area results from the 80 and 50 mm² contact areas: this suggests that the contact stiffness [load/distance] may again be proportional to nominal contact area (as was the case for the fretting situation in Section 6.3). In Figure 6.21, contact stiffness measured by DIC is compared to that given by an elastic ‘smooth contact’ FE model of the pad-specimen experiment. Details of this model have already been outlined in Section 6.3 and the results shown in Figure 6.21 are from the FE model with 80 mm² contact area without any layer to represent the interface compliance. In Figure 6.21a and Figure 6.21b, tangential and normal contact stiffness are plotted against distance between measurement points, respectively.

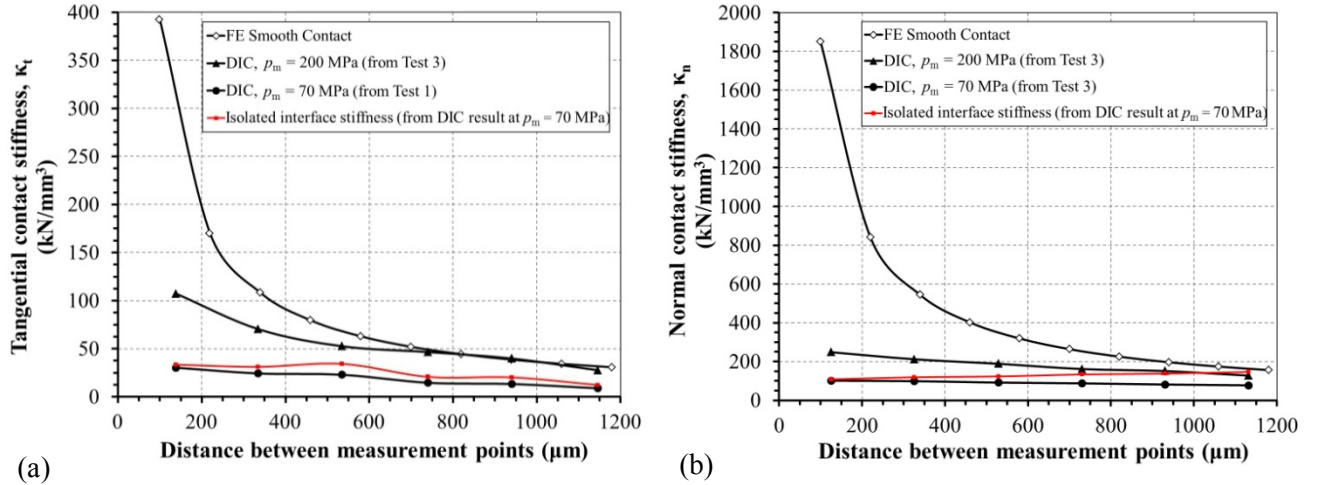


Figure 6.21: Comparison of contact stiffness (per unit area) derived from an elastic ‘smooth contact’ finite element model with that derived from DIC: (a) tangential contact stiffness (DIC tangent stiffness at $Q = 0$ from Test 1 and Test 3), and (b) normal contact stiffness (DIC tangent stiffness from Test 3). Isolated interface stiffness for the DIC results at the 70 MPa normal pressure is also given. Contact area: 80 mm² in both cases.

The DIC stiffness plots shown in Figure 6.21a are the initial tangential contact stiffness of the pre-sliding regime (i.e. slope of the loading curve at $Q = 0$) where tangent stiffness is

expected to be closest to the elastic ‘unload stiffness’ (see Figure 6.16). The previous comparison of the FE model to experimental results (Figure 6.5) was to the tangential contact stiffness results obtained during gross-slip fretting wear tests by Kartal *et al.* [94] (normal pressure: 70 MPa) where the experimental result was significantly less stiff than the elastic smooth contact FE prediction. Here also, the DIC stiffness values are considerably less than the FE result for the 70 MPa contact pressure, though they are greater (by about four times) than the corresponding DIC stiffness values quoted in Kartal *et al.* [94] (see Figure 6.5) – the reason for this is discussed later. The ‘isolated interface stiffness’ (see Section 6.3.3 and Equation 6.5) is also shown for the DIC result at 70 MPa normal contact pressure, and this is again similar to the DIC result. It can, therefore, be assumed that at a normal pressure of 70 MPa, the interface compliance is dominant and that proportionality of stiffness [force/distance] with nominal contact area will hold true. However, Figure 6.21a also shows a DIC result for a normal contact pressure of 200 MPa where the tangential contact stiffness is considerably closer to the predicted FE result.

In the case of normal stiffness (Figure 6.21b) the experimental result at $p_m = 70$ MPa is considerably less than the FE smooth contact stiffness, though again, as the normal pressure is increased from 70 to 200 MPa, the experimental stiffness approaches closer to the FE smooth contact elastic result. As with tangential contact stiffness, both DIC and FE stiffness values increase as the interface is approached by the measurement points – though the DIC stiffness increases by much less than the FE as it becomes dominated by the interface compliance as the interface is approached (this is also clear from the displacement profile which was shown in Figure 6.13).

Figure 6.19, Figure 6.20, and Figure 6.21, above, all show that both normal and tangential contact stiffness are in fact ‘variable stiffness springs’ whose stiffness values depend upon the contact pressure. Figure 6.21 helps to shed light on an important

anomaly in the literature where some authors (Johnson [84], and O'Connor and Johnson [86]) who observed the remote load-displacement response of real contacts found that the 'multi-asperity interface' had little impact on their results which agreed well with a smooth contact elastic analysis; whereas, other authors (Berthoud and Baumberger [87], and Shi and Polycarpou [116]) observed results where the rough interface reduced the stiffness of the contact arrangement to well below that of the elastic smooth contact predictions.

It is suggested here that it is the degree to which the contact area approximates the smooth contact case that determines whether the contact stiffness will agree with elastic predictions or be dominated by the rough interface. Essentially, as the real area of contact approaches the apparent area of contact, a response similar to the smooth contact prediction would be expected. Whether or not a contact approximates the smooth case probably depends on factors such as contact pressure, material hardness and surface roughness. The suggestion here is that at low normal pressures; high hardness; and high roughness, contact stiffness will be controlled by the multi-asperity contacts of the rough interface, but at high normal pressures; low hardness; and low surface roughness, these contacts will merge until the contact approaches the smooth contact case. When a contact is dominated by a multi-contact interface, both micro-slip on the scale of the asperities (Campaña *et al.* [159]) and extra compliance introduced by voids (Sevostianov and Kachanov [156]) tend to reduce the contact stiffness well below the smooth contact prediction (which is based purely on bulk deformation). The hypothesis just outlined is consistent with Figure 6.21 where it can be seen that at the lower normal pressure (70 MPa), the contact stiffness is lower and more dominated by the rough interface, whereas at the higher pressure (200 MPa), the contact stiffness appears to approach closer to the smooth contact FE result.

This analysis would also help explain the discrepancy arising from test results in the literature. If we examine the results in the literature in more detail, the discrepancy can more easily be understood. Of those authors who found agreement with the smooth contact prediction, Johnson [84] was using smooth ball bearing type surfaces while O'Connor and Johnson [86] used a very high mean contact pressure (472 MPa). Both these authors used the sphere-on-flat arrangement. On the other hand, the authors who found that the interface roughness dominated the measured contact stiffness were using very low contact pressures (Berthoud and Baumberger [87] used mean pressures up to 0.04 MPa for their flat-on-flat tests, and Shi and Polycarpou [116] used contact pressures up to 0.055 MPa for their sphere-on-flat experiments). It would seem intuitive (purely for geometric reasons) that an approximation to the smooth contact case is harder to achieve with nominally flat conformal contacts than with non-conformal contacts. This is because nominally flat surfaces can come into contact at only a few points even at high pressures; whereas, conformal contacts are forced into the desired contact by geometrical requirements even at low pressures. This may explain why the results given here still deviate significantly from the smooth contact elastic result even at a normal pressure of 70 MPa (Figure 6.21). Authors [84, 86, 87] measured the response to tangential loading while [116] measured the response to normal loading, and these observations, together with the results in Figure 6.21, suggest that the conclusions being made here apply equally to both normal and tangential contact stiffness. In summary, the effect which the rough surface interface has upon the contact stiffness probably depends mainly upon contact pressure, surface roughness and surface hardness.

Finally, some discussion on the comparison of stiffness magnitudes is warranted. First, it is useful to compare DIC results obtained here at $p_m = 70$ MPa with available results on the same material (Ti-6Al-4V) taken at the same normal contact pressure using similar

optical techniques in the literature. These are only available for tangential contact stiffness in gross-slip fretting tests. Kartal *et al.* [94, 95] (using DIC), obtained values between 8–12 kN/mm³ closest to the interface using contact areas of 80 and 20 mm², and Proponentner [160] (by tracking two laser points on either side of the interface) obtained values of 12–20 kN/mm³ on a much smaller rig setup with a 1 mm² contact area. In the present work, values of 35–45 kN/mm³ were obtained (see Table 6.2). There are some important differences which explain these higher values: first, the values quoted from the literature are from fretting wear tests where wear debris will serve to reduce interface stiffness, and also, the authors in [94, 95, 160] measured the initial secant stiffness at the beginning of micro-slip rather than the initial tangent stiffness at the beginning of micro-slip. Another key difference is that the stiffness results quoted in Table 6.2 are the initial tangent stiffness values at the very onset of the microslip regime during ‘first load-up’ of the contacts, whereas, those by Kartal *et al.* [94, 95] and Proponentner [160] are the initial ‘unloading stiffness’ at the point of motion reversal during reciprocating sliding. This ‘unloading stiffness’ is generally expected to be elastic; though, wear debris is a complicating factor.

Turning to the comparison of DIC and ultrasound stiffness magnitudes in the present work, both can be graphically compared for tangential contact stiffness in Figure 6.15 and for normal stiffness in Figure 6.19, and numerical values of measurements by the two methods are also given for tangential and normal contact stiffness respectively in Tables 6.2 and 6.3 for $p_m = 70$ MPa, and $Q = 0$ (i.e. at the onset of tangential loading in the case of tangential stiffness).

Table 6.2: Comparison of the DIC and ultrasound measurement techniques for tangential contact stiffness measured at $p_m = 70$ MPa and $Q = 0$. Distances (in μm) between measurement points for calculation of DIC relative displacements were: 137, 145, 102, and 142 for tests 1, 2, 7, and 8 respectively.

Test No.	Contact area (mm^2)	Tangential contact stiffness, κ_t (kN/mm^3)	
		DIC	Ultrasound
1	80	37	68
2	80	45	47
3	80	-	105
4	80	-	147
5	80	-	98
6	80	-	81
7	50	35	224
8	50	37	101
9	50	-	114
10	50	-	42

Table 6.3: Comparison of the DIC and ultrasound measurement techniques for normal contact stiffness measured at $p_m = 70$ MPa and $Q = 0$. Initial distances (in μm before loading) between measurement points for calculation of DIC relative displacements were: 126, 135, 111, and 150 for Tests 3, 4, 9, and 10, respectively.

Test No.	Contact area (mm^2)	Normal contact stiffness, κ_n (kN/mm^3)	
		DIC	Ultrasound
1	80	-	252
2	80	-	253
3	80	101	448
4	80	80	233
5	80	-	220
6	80	-	366
7	50	-	265
8	50	-	262
9	50	73	236
10	50	56	140

Figure 6.15 and Table 6.2 both show that the DIC tangential contact stiffness is reasonably repeatable in the four tests where it was measured. There is some degree of variability in the ultrasound measurements of tangential contact stiffness and this may possibly be explained by noting that ultrasound is very much a local measurement derived from a sub-region at the centre of the interface (which will be similar in size and shape to the emitting transducer) and will depend intimately on the contact which occurs in this region. Therefore, it can be sensitive to small changes in local roughness and waviness etc. Since the DIC measurement is taken from points located some 50 microns or so from the interface, the DIC measurement is less likely to be as variable since it depends more on the behaviour of the whole contact region. On average (based on all data in Tables 6.2 and 6.3), the ultrasound is approximately 2.7 times stiffer than the DIC result for tangential stiffness and 3.5 times stiffer for normal contact stiffness. However, given that the two techniques are widely different in their method of measurement, the fact that the agreement in magnitude is this close is noteworthy. Particular good agreement between DIC and ultrasound can be seen graphically for normal stiffness in Figure 6.19. An analysis of available results in the literature was outlined in Chapter 2 where tangential contact stiffness data from separate ultrasound and DIC studies were compared. Although those tests did not have the same materials; contact geometry; surface topography; or testing regime, their finding that the ultrasound results were stiffer is consistent with the conclusions being made here.

In attempting to understand why ultrasound gives a stiffer result, the following points should be noted. As we have seen, ultrasound gives the local unloading stiffness, whereas DIC gives the tangent stiffness. This argument can be used for normal stiffness; however, for tangential stiffness, as long as both stiffness values are determined at the very onset of microslip ($Q = 0$), the effect should not be very important. In addition, the

DIC result is derived from points located some 50 microns or so into the bulk, whereas the ultrasound result is derived directly from the interface itself where stiffness is expected to be higher (Figure 6.21 illustrates the predicted rise in stiffness as the interface is approached) – this argument applies to both normal and tangential stiffness. Also, in the case of tangential loading, it is difficult to obtain an accurate stiffness by DIC at $Q = 0$ since the relative displacement data can be difficult to interpret until Q is increased sufficiently (see Figure 6.14c).

6.5 Conclusions

This chapter has investigated aspects of the contact stiffness of rough interfaces using an experimental and numerical approach. It was shown that a smooth contact FE model can be used to distinguish the displacement contribution of the ‘bulk material’ from that caused purely by the interface in DIC measurements – thereby isolating the interface stiffness. This FE approach, used in conjunction with experiments from Kartal *et al.* [94, 95], was used to show that the interface which develops during fretting wear is one that is quite compliant (in the tangential direction) in comparison to the substrate bulk material (at least for the 70 MPa normal pressure investigated). For example, if the interface is modelled as a 100 μm thick elastic layer in this case, then a Young’s modulus as low as 2.29 GPa (i.e. approximately the stiffness of polystyrene) is required to simulate the experimental tangential contact stiffness results. One consequence of this is that the DIC measurement is almost equivalent to the ‘actual interface stiffness’ and is unaffected by bulk compliance (at least when measured reasonably close to the interface). Another ramification of a compliant interface on a stiff substrate was shown to be the resultant proportionality of tangential contact stiffness with nominal contact area which permits

normalisation of stiffness results by nominal contact area as a means of comparing results from contacts with different areas. This result was specific to a particular contact pressure of 70 MPa and may not apply to higher pressures.

In the second part of the chapter, a comparison of concurrent DIC and ultrasound measurements was undertaken in a series of unidirectional ‘first load-up’ tests where either the normal or tangential load was incrementally increased. Despite the small displacements involved ($< 2\text{--}3\ \mu\text{m}$ up to slip), the DIC technique was seen to produce distinct load versus relative displacement curves for both normal and tangential loading directions. The load-displacement curves were non-linear indicating plastic flow: as would be expected from ‘first load-up’ of a rough contact. For the first time, a fundamental difference (proposed by [111]) between the tangent stiffness taken from the loading curve and the ‘ultrasonic stiffness’ was observed. Namely, that ultrasound always measures the local unloading stiffness which is elastic and unaffected by the plastic softening of asperities tending to reduce the tangent stiffness. This effect was observed experimentally in the present work in the tangential loading case where DIC stiffness was seen to reduce towards zero as loading proceeded; whereas, the ultrasound saw no reduction in tangential contact stiffness. For normal contact stiffness, both DIC and ultrasound values increased with normal load due to the increase in real area of contact which masks this effect (though normal stiffness measured by DIC was less stiff than ultrasound at equivalent normal loads – probably owing to plastic softening lowering the DIC stiffness but not the ultrasound stiffness). Confirming the work of previous researchers, tangential, as well as, normal contact stiffness was found to increase with normal loading.

When compared to the DIC results derived by [94] during fretting tests, the tangential contact stiffness values obtained by DIC in the unidirectional tests were about four times

stiffer – this may be due, among other things, to the tendency of wear debris in the fretting tests to reduce contact stiffness. The DIC results for contact stiffness arising from these unidirectional tests were also compared to stiffness values derived from the smooth contact FE model. For the lower contact pressure (70 MPa), the contact stiffness was dominated by the compliance of the rough interface. However, when the contact pressure was increased to 200 MPa, the contact stiffness approached closer to the smooth contact elastic FE result. This suggests that at low pressures the multi-asperity contacts dominate the contact stiffness, but as the pressure is increased, these contacts merge to form a contact which begins to approximate the smooth contact case. Therefore, at lower normal contact pressures, the result that contact stiffness is proportional to nominal contact area can be true even for the non-fretting case; in fact, the results show that no discernible difference in ultrasound and DIC contact stiffness (per unit area) between the 80 and 50 mm² contacts can be observed at 70 MPa. It was hypothesised (based on results here and in the literature) that the degree to which contact stiffness depends upon bulk deformation (when measured remotely) depends on how well the contact approximates the smooth contact equivalent. Following from this analysis, it seems reasonable to suggest that the extent to which the contact stiffness depends upon the interface will depend on three main factors: contact pressure, surface roughness, and surface hardness.

Turning finally to comparison of magnitudes, given the vast difference between the two methods, the fact that corresponding stiffness values are of the same order of magnitude is firstly of interest. Ultrasound was always stiffer than DIC: on average, for normal stiffness, the ultrasound result at $p_m = 70$ MPa was about 3.5 times stiffer than DIC, and for tangential stiffness, the ultrasound result at $p_m = 70$ MPa was 2.7 times stiffer. This disagreement is sufficiently small so that we cannot raise a question about the validity of

either method especially given that ultrasound measures local unloading stiffness whereas DIC measures tangent stiffness. However, it does seem that there is quite a degree of variability in the ultrasound measurements which are probably more sensitive to local contact and roughness conditions than DIC. DIC may be more robust in certain circumstances since its measurement depends more on the entire contact (as the measurement points are somewhat removed from the surface). There are disadvantages to the DIC technique also, such as the fact that a reliable tangential contact stiffness measurement at the very onset of micro-slip is difficult to determine due to the small displacements and the 'slack' involved before the load is sufficiently increased. Choosing which technique to use in a given case will depend on which technique can be physically accommodated. Also if tangent stiffness is required, DIC is the obvious choice, whereas ultrasound is suited to measuring the elastic local unloading stiffness. In addition, DIC is more suited to measuring contact stiffness in fretting tests as the ultrasound reference signal is invalidated by surface wear in fretting.

Chapter 7

Conclusions and future work

In this chapter conclusions are drawn from the research discussed in this thesis. The conclusions are divided into three sections covering the following topics: gross-slip fretting, friction modelling, and contact stiffness. Suggestions for possible future work which could be carried out in order to make further advancement are then outlined.

7.1 Conclusions

7.1.1 Gross-slip fretting

On the topic of fretting wear, it was shown that the full range of 3D roughness parameters (both amplitude and spatial), together with new 3D profilometry techniques can be of considerable benefit in the analysis of surface evolution. In particular, auto-correlation and cross-correlation parameters proved useful in estimating the lateral size of wear scar surface features. A minimum size of scan area was shown to be required to give reliable surface topography parameters and both interferometry and focus variation microscopy based 3D surface profilometry techniques were shown to report similar results.

Friction and wear behaviour during gross-slip fretting were investigated using a series of in-line and torsional tests on two different materials: the titanium alloy Ti-6Al-4V and the

nickel alloy Udimet 720. Effect of initial roughness was also investigated. It was found that sliding friction rises sharply due to the erasing of initial surface layers in the first few cycles of fretting (from values as low as 0.15), before varying more gradually within a range of approximately 0.35–0.75 for Ti-6Al-4V and 0.50–0.88 for Udimet 720 for the remainder of the tests. Friction was slightly higher for the Udimet 720 than for the Ti-6Al-4V.

Initial surface roughness was found to have little effect – probably because the initial topography is quickly transformed by the wear action, and also because initial differences in surface roughness may not have been large enough. Examination of the post-test wear scars (after 5000–6000 cycles of fretting), showed significant increases in surface roughness with both the in-line and torsional tests (peak-to-peak amplitude was 1 mm – for the torsional tests this was the amplitude at the ring median circle). Significant wear was visible from the 3D surface profiles and optical micrographs showed evidence of significant adhesive wear, material transfer, and creation of third body debris. With exactly the same test conditions, Ti-6Al-4V was found to give a distinctly different wear scar shape than Udimet 720. Ti-6Al-4V gave a W-shaped wear scar, while Udimet 720 gave a more random distribution of a few major pits with corresponding peaks on the counter-surface. This indicates that the chemical and tribological properties of a material are critical factors in determining wear scar shape.

A commonly observed frictional phenomenon in fretting where the friction force varies (usually increases) with sliding during the gross-slip phase of individual cycles was also investigated. This was found to be caused by wear-scar interaction. However, the effect occurred to much the same extent whether or not wear scar ends were present, thereby suggesting the presence of local interacting wear scar features distributed throughout the nominal contact area. Indeed, both qualitative and quantitative analysis of the post-test

wear scars from both in-line and torsional fretting tests revealed the presence of a distribution of local peaks and troughs. Auto-correlation and cross-correlation analysis of the 3D wear scars (for both in-line and torsional tests) showed that these features were similar in size to the imposed fretting displacement amplitude (peak-to-peak) for both the in-line and torsional tests. These features essentially butt against each other as the maximum displacement is approached. A simple model was put forward to show how the reaction forces at a local peak-trough interface contribute to producing the friction behaviour which is commonly observed. The model suggests that the shape of the friction variation is related to the wear scar shape.

It should be emphasised that interaction at the wear-scar ends almost certainly takes place, but in the present work, interaction of local features distributed over the contact region seem to have been responsible for most of the observed friction variation. Finally, it is important that researchers performing fretting tests in order to validate velocity dependent friction models should appreciate that the observed friction variation effects in the hysteresis loops are probably due to wear-scar interaction rather than velocity variations in most cases. This is important as some researchers in the dynamics community have already pointed to observed friction variation behaviour in fretting tests as validation of their models without considering wear effects.

7.1.2 Friction modelling

A model of kinetic friction was put forward based on the sliding interaction of cylindrical and spherical elastic-plastic asperities using finite element analysis. A number of advances were made over previous attempts at this type of modelling: greater asperity overlaps were facilitated, a ‘Bowden and Tabor’ type adhesional shear strength was enforced at the asperity junction interface, and asperity damage was incorporated in the

model. Friction coefficients were predicted using a single junction model whose results could be assumed to apply to two surfaces where all asperities have the same height and interact independently with the same overlap. The predicted friction coefficients were found to increase with asperity overlap and interface shear strength, and friction coefficients from the models incorporating a failure criterion were slightly higher than the corresponding models with infinite ductility (though both showed the same trend).

The friction predictions from the single asperity junction were then incorporated into a statistical model (similar to the Greenwood-Williamson model) allowing for a surface with a stochastic distribution of asperity heights to interact with a surface with a single constant asperity height. Using this approach, plots of predicted friction coefficient versus a characteristic surface topography parameter were obtained. The models were based on the titanium alloy Ti-6Al-4V, and so sliding friction experiments were also carried out on this material to compare with the models. It was found that the experimental friction coefficients lay in the mid-range of model predictions for the case where both surfaces had a single asperity height and for the case of the stochastic surface. Previous modelling attempts had under-predicted the friction coefficient and given results more appropriate to lubricated contact. However, since the approach outlined here has given realistic predictions of the friction coefficient, it can be concluded that the combination of friction generation by dissipation of energy due to plastic deformation, and due to the overcoming of an interface tangential shear strength, (in a suitable asperity model) is a plausible physical mechanism for friction.

7.1.3 Contact stiffness of rough interfaces

Contact stiffness was investigated in both fretting and non-fretting situations. In the fretting case, an elastic finite element model was developed for comparison with

tangential contact stiffness results in the literature measured by digital image correlation (DIC). For the particular contact pressure involved (70 MPa), the experimental stiffness values were considerably less than the finite element predictions. An FE model of the type used gives stiffness due only to bulk elastic deformation between measurement points, while experimental results contain a rough interface contribution in addition to the bulk contribution. Therefore, a method was developed to use the FE model to isolate the true interface stiffness by assuming that the interface stiffness and the bulk stiffness act in series to give the stiffness which would be experimentally given by DIC. This isolated, or true interface stiffness, was not very different to the experimental one, indicating that the bulk stiffness had little effect.

In addition, a second FE model was developed which included a compliant elastic interfacial layer, and this model was tuned (via the layer properties) to match the experimental results. The fact that, with a 100 μm thick layer, a Young's modulus similar in magnitude to polystyrene was required indicates that the interface was behaving like a compliant layer on a stiff substrate. The correctly tuned model was then used to investigate the effect of nominal contact area on the tangential contact stiffness, and a linearly proportional relationship was found. This is due to the high interface compliance in comparison to the bulk material. A main contributor to interface compliance in fretting is probably wear debris.

In the non-fretting case (i.e. monotonic 'first load-up' of the contacts) at the same contact pressure (70 MPa), the isolated tangential contact stiffness values were somewhat higher, but still largely uninfluenced by bulk elastic deformation. However, when the contact pressure was increased to 200 MPa, the isolated interface stiffness approached closer to the FE elastic prediction (this was found for both normal and tangential contact stiffness). It was concluded that the extent to which the interface dominates the stiffness is

dependent on factors which influence to the degree to which the interface behaves like an ideal smooth contact. Therefore, parameters such as contact pressure, surface roughness, and surface hardness are all likely to influence the effect the interface will have on stiffness which is derived from displacement measurements made in the bulk.

In the non-fretting case, an experimental programme was undertaken to compare contact stiffness results (normal and tangential) measured using two markedly different techniques: DIC and ultrasound. Tests were carried out where DIC and ultrasound measurements were made simultaneously in the same experiment. The normal and tangential contact stiffness values were each measured during monotonic increases in normal load, and (separately) tangential load (up to the point of slip) in ‘first load-up’ of a rough contact. Although DIC had been used to take measurements of tangential contact stiffness during fretting, as far as the author is aware, this is the first occasion where it has been used to determine normal contact stiffness, and also the first time it has been used to measure load-deflection data at a contact for the ‘first load-up’ case. It has been found to be very suitable to each of these tasks (and also to the task of measuring tangential load-deflection data) since distinct load-deflection curves of the type that would be expected were found. This work also appears to be the first use of ultrasound for the measurement of contact stiffness in pre-sliding.

An important difference between stiffness values obtained by ultrasound and the load-deflection methods (i.e. DIC) was shown experimentally for the first time. It had been quite recently hypothesised that ultrasound would give an ‘elastic unloading stiffness’ since it applies a cyclic displacement disturbance. The stiffness values obtained from DIC, on the other hand, give the stiffness of the ‘load-up’ curve. This difference was indeed observed in the experiments of the present work (for the tangential loading case) as the ‘DIC stiffness’ decreased due to plastic softening of asperities when the tangential

load was increased, but the ultrasound stiffness did not, since the elastic unloading stiffness would not be expected to vary with tangential load. In fact the ultrasound stiffness increased slightly – probably due to junction growth.

This difference between the techniques (i.e. that ultrasound actually measures an ‘unloading stiffness’ while load-deflection methods measure a ‘loading stiffness’) had not been experimentally observed before for two reasons. Firstly, a load-deflection technique and an ultrasound technique would have to be used simultaneously in order to observe the difference; secondly, a majority of the work in the literature had been devoted to studying the effect of increasing the normal load. Results presented here, show that when normal load is increased, the normal stiffness measured by DIC and ultrasound both increase (in an approximately linear fashion for the load range covered here). This is because the increase in stiffness caused by an increase in real contact area masks the difference between the techniques in the normal load case. To conclude: ultrasound measures ‘local unloading stiffness’, while stiffness obtained from DIC loading curves give the local ‘load-up stiffness’. If everything is elastic there will be no difference between these, but if any irreversibility/plasticity is present, then the two will give a somewhat different result. This effect is detectable in tangential ‘first load-up’ of a contact, but not for the normal load case due to the growth in real contact area.

Turning to comparison of magnitudes, the ‘first load-up’ of a rough contact is expected to be plastic and irreversible, and the results given in this thesis indicate this. Therefore, given the fundamental difference in the techniques discussed above, one might expect ultrasound to give a stiffer result than DIC as ultrasound is not affected by the plastic softening of the ‘load-up’ curve. This is indeed what was observed in the case of normal loading where the ultrasound was, on average, 3.5 times stiffer than the DIC result. However, in the case of first tangential ‘load-up’, results were taken at the very initiation

of the partial-slip region where both the loading and unloading stiffness values could be expected to approach each other. Despite this, the ultrasound results were still stiffer than the DIC results (by 2.5 times on average) for the case of tangential contact stiffness. This might be a result of an additional difference between the two methods. DIC takes its displacement measurements at least some 50 μm , or so, from the contact interface; whereas, the ultrasound result is derived directly from the surface, and a simple elastic analysis (such as the FE models) indicate that tangential contact stiffness increases as the measurement points (between which relative displacements are being measured) approach the interface.

However, taking into account the wide difference between the two techniques (one measures on the Ångstrom scale, the other on the micron scale), the disagreement in magnitudes is quite small and probably explicable by physical reasons such as those outlined above. This level of agreement is, in a way, a validation of both techniques. It would seem that both provide an effective means of measuring contact stiffness.

The variation in ultrasound results between different experiments was more significant than for DIC, and this is likely due to the fact that ultrasound results are highly dependent upon the intimate contact which occurs at the part of the interface which is receiving the ultrasound waves. It is, thus, more sensitive to surface topography and alignment issues; whereas, DIC measurements are taken a little into the bulk material where more of an overall average effect of the interface can be expected to be present in the measured displacements.

Finally, which of the two techniques one chooses will depend on the particular task at hand. The fact that ultrasound gives an ‘unloading stiffness’ surely limits its use in situations where there is loading irreversibility, and where the ‘loading stiffness’ is

required. Ultrasound is also dependent on an initial reference measurement so that it is unsuited to measuring contact stiffness when wear is occurring; however, this is not a problem for DIC. Although the resolution of the DIC method is more limited, and it is difficult to measure stiffness at very low normal or tangential loads with DIC, it does appear to be more robust. The above developments represent significant advancement in the understanding of frictional behaviour, but there is some further work which could be done to refine these developments and this is addressed next.

7.2 Suggestions for future work

7.2.1 Gross-slip fretting

- Additional tests could be performed where the initial difference in surface roughness of the samples is greater. For example, a polished surface with an S_a value less than about 0.1 μm could be compared to rough milled or ground samples with an S_a value greater than about 4 μm . Although the effect of the roughness may get ‘erased’ due to fretting wear, it is likely that differences in behaviour would be apparent in the early test cycles.
- The distinct difference in wear-scar geometry between the two different materials used in this thesis, even though they were tested under identical conditions, needs further investigation. Further materials may need to be tested, and a dedicated study aimed at determining the underlying chemical or tribological reasons why different materials produce different wear scars in fretting may be required.
- The friction variation during the gross-slip phase of a fretting cycle has been identified with wear-scar interaction. In the present study, this occurred mainly at local features distributed over the nominal contact area whose sizes were similar to the applied fretting stroke. However, there is an obvious need to perform tests

where the applied test amplitude is varied in order to investigate its effect on the feature sizes. The effect of normal pressure on feature size may also warrant study, since the higher pressure tests for the torsional experiments in the present work appeared to produce larger feature sizes. Also, only two materials were tested in this work, and there is probably a need to check the validity of the conclusions with other materials having different tribological behaviour.

- In the current work, it appeared to be the case that most of the contribution to the friction variation arose from local wear-scar interactions rather than from interaction at the wear-scar ends. Studies to assess the relative importance of these two types of interaction may be useful. In this regard, and also to widen the conclusions of the present work, it may be helpful to perform similar investigations using different contact arrangements (i.e. cylinder-on-flat, sphere-on-flat etc.).
- Finally, the friction variation during sliding may have the effect of increasing the area of the hysteresis loop (i.e. the energy dissipated in damping and invested in frictional work). Clearly, this would have implications for both damping and fretting wear issues, and may need to be accounted for in predictive models of fretting joints by both the tribology and vibrations communities.

7.2.2 Friction modelling

- The bulk of the modelling in the present work was carried out with cylindrical asperities, and even though a selection of results from spherical asperities showed similar results, a complete set of results for spherical asperities would be useful for completeness. In addition, still great asperity overlaps could be modelled.

- In this work, a range of interface shear strength values up to the yield stress in shear of the material comprising the asperities were used and friction predictions were plotted for each one; however, there is a need for more experimental investigation to establish the nature of the interface shear strength. Namely, whether it varies with any other interface parameters and what its magnitude should be for a given material pair.
- The friction coefficients predicted in the present work were indicative of metals sliding in atmospheric conditions; however, it is known that higher friction coefficients are obtained in a vacuum where true metal-to-metal contact occurs. It is thought that these higher friction coefficients result from normal adhesion. Therefore, there is a need to take account of normal adhesion in the models if friction coefficients for true metal-to-metal contact are to be predicted.
- In the present work, a slightly modified Greenwood-Williamson statistical model of multi-asperity contact was used; however, attempts should be made to apply the single asperity junction results to more sophisticated models of surface contact which simulate more accurately the contact of two real rough surfaces (i.e. allowing for oblique contact of asperities etc.).
- Asperities are often less than the size of a grain in the microstructure of the parent material; hence, account needs to be taken of the fact that asperities in different grains and with different crystallographic orientations will behave differently.
- The friction tests which were used for comparison with the modelling in the present work were from standard ground engineering surfaces where the surface topography does not agree with that of the more idealistic type surface used by the modelling. In addition, the topography parameters required for comparison to the modelling are difficult to measure for standard ground surfaces. There is need,

therefore, to attempt to perform tests on a surface which has a prescribed distribution of asperities whose shape and size are known. For example: a surface with an array of spherical asperities all having the same radius and height. One promising technique which would allow this to be achieved is the surface patterning technique which uses the principle of laser interference. This would facilitate a better opportunity for comparing the friction modelling results with experiments.

- Ultimately, a fully predictive model of friction is required based on input data such as surface topography, material and chemical properties, mechanical properties, loading conditions, and geometry etc.

7.2.3 Contact stiffness of rough interfaces

- In the present work, contact stiffness was somewhat lower in the fretting wear case than in the non-fretting case. In the fretting situation, it was found that the interface behaved like a compliant layer on a stiff substrate. The extra compliance was attributed to the presence of wear debris. Proportionality of tangential contact stiffness with nominal area followed from this compliance. However, the fretting results applied to a fixed contact pressure of 70 MPa. Therefore, there is a need to see whether the interface compliance still dominates at higher pressures during fretting wear.
- To improve the comparison between ultrasound and digital image correlation measurements of contact stiffness, more focus should be given to making the comparison during unloading rather than loading. This is because (as was demonstrated in this work) ultrasound gives a local ‘unloading stiffness’ due to its cyclic nature.

- In the present work, both normal and tangential contact stiffness appeared to increase approximately linearly with normal pressure. However, it would be useful to take both DIC and ultrasound measurements at higher pressures to see whether the contact stiffness approaches a plateau (i.e. a limit indicative of the smooth contact case). This softening of the contact stiffness versus contact pressure curve may not have been observed for the pressures used in the present work due to the high strength of the alloy under investigation (Ti-6Al-4V).
- Any moves to increase the resolution of the DIC technique by obtaining images of still higher resolution are also welcome, especially when attempting to measure the initial portion of the load-deflection curve.
- A comprehensive study of how factors such as contact pressure, surface topography, surface hardness, and surface chemistry affect the extent to which remote contact stiffness (i.e. taken from the relative displacement of points in the bulk material) departs from theoretical smooth contact predictions would also be useful. For example, the present work suggests that, the higher the contact pressure, the more the contact stiffness will approximate the smooth contact case. Likewise, smoother surfaces should obviously give results which are more similar to the ideal smooth contact case than rougher surfaces.
- There is also some interest in what will happen to the ultrasonically measured tangential contact stiffness during sliding. Theoretically it should be zero, but this may not be so in practice.

7.2.4 Concluding remarks

The ultimate aim is to develop physical understanding to such a level that predictive models of friction and contact stiffness based on various material and surface properties can be realised. This is, of course, an enormously challenging task as there are so many factors which have a bearing on friction and contact stiffness. The complexity of frictional interfaces in terms of topography, chemistry, and mechanics means that it is always difficult for researches in one particular field to address the full depth of the problem. Advancement towards better predictive models will likely come about through piecewise advances in various aspects of the problem.

This thesis has provided advances in some of the mechanical engineering aspects of the problem. It has made important progress in: furthering the physical understanding of friction and contact stiffness, in understanding how friction and contact stiffness are affected by various other parameters, and in assessing techniques for accurate measurement and characterisation. Further developments will need to be achieved in such areas as comprehending the chemistry of the adhesion bonds, and the mathematics describing rough surface interaction etc. In particular, new experimental and modelling techniques to probe behaviour at the nano-scale may reveal further layers of understanding in the complex world of frictional interface behaviour. All of the developments required in the various aspects of the problem will ultimately have to be incorporated in inclusive predictive models of interface interaction which are more descriptive of real frictional behaviour.

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