SANDIA REPORT

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Report on the SNL/AWE/NSF International Workshop on Joint Mechanics, Dartington, United Kingdom, 27–29 April 2009

Daniel J. Segalman, Lawrence A. Bergman, David J. Ewins

Prepared by
Sandia National Laboratories
Albuquerque, New Mexico 87185 and Livermore, California 94550

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Daniel J. Segalman,
Sandia National Laboratories
P.O. Box 5800, Org 01220
Albuquerque, NM 87185-1393
djsegal@sandia.gov

Lawrence A. Bergman, University of Illinois, Urbana, Il
and David J. Ewins, FRS, University of Bristol, UK

Abstract

The SNL/AWE joint mechanics workshop, held in Dartington Hall, Totnes, Devon, UK 26-29 April 2009 was a follow up to another international joints workshop held in Arlington, Virginia, in October 2006.

The preceding workshop focused on identifying what length scales and interactions would be necessary to provide a scientific basis for analyzing and understanding joint mechanics from the atomistic scale on upward. In contrast, the workshop discussed in this report, focused much more on identification and development of methods at longer length scales that can have a nearer term impact on engineering analysis, design, and prediction of the dynamics of jointed structures.

Also, the 2009 meeting employed less technical presentation and more break out sessions for developing focused strategies than was the case with the early workshop. Several "challenges" were identified and assignments were made to teams to develop approaches to address those challenges.

Acknowledgment

The authors thank all of the participants for their contributions to the workshop.

Special thanks are due to Drs. Adnan Akay and Ken Chong of the National Science Foundation who provided funding to assist the US workshop participants with travel expenses. Dr. Akay also participated in the workshop.

Thanks are also due to Professor Tod Laursen, a workshop participant, and his students who helped with reformatting and sorting of the workshop presentations so that they could be included in this document.

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Chapter 1

Introduction

The issue of predictive structural dynamics is of fundamental importance in multiple sectors of our economy, including manufacturing, transportation, and defense. Applications are so broad as to include optimal design of jet engine components and the specification of tolerances for nuclear weapon components. It has been recognized since the 1960's that the fundamental barrier to predictive structural dynamic simulation resides in the nonlinearity and variability of the mechanical interfaces of practical structures. Historically, this limitation has been obviated by approximating the structure as a linear system and tuning the linear model for that system to match its measured properties.

Given the tremendous advances in computer resources - particularly massively parallel computers - and advances in experimental techniques, it is appropriate to reexamine the problem to assess the possibility of actually predicting structural dynamic response even before a prototype is constructed. For this purpose the Sandia National Laboratories and the National Science Foundation have sponsored a workshop in Arlington, Virginia, 16-18 October, 2006 [1]. A follow up workshop in Dartington Hall, Totnes, Devon, UK 26-29 April 2009 was sponsored by the British Atomic Weapons Establishment (AWE) and Sandia National Laboratories (SNL). It is this second workshop that is reported here.

While the first workshop focused on identifying what length scales and interactions would be necessary to provide a scientific basis for analyzing and understanding joint mechanics. The workshop discussed here focused much more on identification and development of methods at longer length scales that can have a nearer term impact on engineering analysis, design, and prediction of the dynamics of jointed structures.

Participants in that workshop included distinguished investigators from the United States and Europe representing expertise in the various sciences relevant to this problem. These include vibrations, tribology, elasticity, and computational mechanics. There was an effective balance of capability in theoretical mechanics, computing, and experiment.

Chapter 2

Focus and Goals

Addressing the Problems as Tractable Parts

The previous workshop attempted to identify each of the component technologies that touch on dynamics of jointed structures and then to draw the relevant connections among those technologies.

Relevant phenomena span scales from nanometers to the lengths of structures. A significant conclusion at the end of that workshop was that the range of length scales and the physics and chemistry that dominate at each scale made it unlikely that those scales could be coupled in a rigorous manner in the near future. This conclusion was first prompted by the observation that participants could address modeling issues either at the extremely small scales (atomistic-nano) or scales moderately large (asperity and up), but there were no clear strategies to bridge those scales. The difficulty seems to be that analysis at each scale requires consideration of physics and chemistry that dominate at that scale and that there are many discretely defined scales between the small and large scales that that remain to be addressed.

Consequently the 2009 workshop emphasized topics more on the macro- and meso-scales, where coupling through the relevant length scales becomes more tractable. The issue of bridging the many length scales from nano to macro will be the focus of future workshops, particularly as new experimental methods, theoretical insights, and computational capabilities become available.

Focus on these larger length scales resulted in identification of several core issues and the development of strategies to address them.

Themes

Consistent with the above, invited talks (Appendix D) and breakout sessions (Appendix E) focused on the following themes:

Theme A concerns the engineering end-user needs for better predictive models of joints.

It concerns the need for better understanding and theoretical modelling of the various friction phenomena which ultimately determine the mechanical characteristics of so many joints in engineering structures of all types. It also concerns the potential future benefits that might result from an improved prediction cap ability, such as the design of much more effective joints, and much more repeatable ones.

Theme B represents a critical and comprehensive assessment of our current capabilities in this general area. Theoretical, numerical and experimental techniques allow us a degree of control over the joints that we design and use today, but they are still far from ideal, o r from the levels of predictability that apply to the components which are assembled by using the joints of current interest.

Theme C includes the various recent developments and ideas for future methods which will eventually allow us to deliver the predictive modelling and design capabilities to make the joints of the future exactly what we want them to be: repeatable, efficient, with stiffness and damping performance characteristics as desired to optimize the dynamics of the stricture of which they form part.

Chapter 3

Results

The three day workshop brought together a diversity of participants from academia, government, and industry. Further the foci of those people ranged in length scale from asperity size up to the dynamics of large structures. Interests included mathematical elasticity, tribology, laboratory measurements of joint properties, finite element studies at various scales, and structural dynamics.

This combination of expertise and interest resulted in very effective interactions, rapid homing in on technical issues, and systematic development of paths forward.

Workshop Wind-Up

During the last session of the Workshop, there was a discussion seeking to distill the essential issues and topics which would emerge as the main items defining the new Road Map for the subject. The resulting list divides into 2 sets – **Actions** and **Challenges**. The Actions are tasks that are essentially short-term goals deemed to be necessary in order to consolidate the foundations of the subject to provide a sound basis for further research. The Challenges are much more substantial tasks, each requiring several man-years of research effort, whose objectives are to move the whole subject on to a new level of technical competence, heading to the ultimate goals of the ability to model, and to predict the dynamics of mechanical joints and thereby to design structures with optimum dynamic properties – including those whose dynamics are actively controlled by the joints themselves.

In many cases, offers of leadership of the tasks were recorded as were expressions of interest in participation.

Actions

1. Terminology & Vocabulary (***Segalman****; Bergman)*
2. Develop Hills Chart (***Dini****; Berger)*
3. Classification of Standard Joint Types *(Hills; Vakakis;* ***Starr***)

4. Classification/Cataloging of
   (a) Non Linearity ID Methods (***Vakakis***)
   (b) Modelling approaches (***Polycarpou****; Quinn)*
   (c) Measurement methods (***Nowell****; Bergman; Akay)*
5. Benchmark current computation multi-scale methods against analytic solutions (***Masud****; Laursen; Quinn)*
6. Create a formal Joints Modelling Network (or Community) with more frequent and regular contacts (this was expressed emphatically); meetings at relevant conferences; workshop series;... Wiki..Joints Chat room

   (***Ewins***;***Segalman****; Nowell;* ***Bergman*** *Gaul; Green;Surampudi; Dini;Quinn)*
7. Form Specialist sub-groups of Community to collaborate on specific Actions/Challenges, e.g.
8. University group on basic joints contact
   (a) mechanics science
   (b) others, to be identified
9. University/industry group on measurements of friction properties required for industrial applications
10. Industry-led group(s) to ensure liaison between end-user requirements and academic research activities (e.g. balancing accuracy requirements for application against accuracy of predictions)

Challenges

1. Round Robin/Benchmark Exercise for Hysteresis Measurements (***Ewins****: Nowell: Gola; Polycarpou; + possibly Epsion(Technion))*
2. Round Robin/Benchmark for Measurement/Prediction of Dissipation in Standard Joints (***Leming****; Goyder; Gaul; Ind; Vakakis)*
3. Repeatability (measurement-to-measurement) and Variability (unit-to- unit) Issue: need to be able to distinguish between, and to greatly improve performance in both aspects (i.e. design of better, more repeatable joints) *(Leming;* ***Goyder****; Gaul; Ind; Polycarpou; Farris; Mignolet)*
4. Framework for Multi-Scale Modelling *(Masud; Dini;* ***Nowell***)
5. Strategy for Uncertainty and Nonlinearity

6. Methodology to quantify cost benefits of improved joints designs
7. Universally-accepted Physical Theory of Friction (which explains, inter alia, where the energy goes)
8. Complex Loading Strategies
9. Measurement of Spatial Distribution of Key Physical Parameters
10. How to include surface chemistry?
11. Eventual implementation of prediction methods in commercial numerical codes

Followup

Since the conclusion of the workshop, the designated team leaders have been compiling and consolidating the responses for actions and challenges. A follow-up meeting to coordinate these actions occurred at the American Society of Mechanical Engineers (ASME) Design Engineering Technical Conference in August 2009.

One particular Action (number 6 has resulted in a well received proposal to create an Research Group within the ASME Board on Research and Technology Development(BRTD). The purposes of this group will be

1. to facilitate inter-institutional and international collaboration in joints research efforts.
2. to promote standardization of nomenclature and procedures where it could be found helpful.
3. to author and edit publications that advance the understanding of joints mechanics issues.
4. to advance research into joint mechanics and dynamics of jointed structures by development of a joints handbook. This handbook would be built through the selection of authors and editors to contribute to the revision of the Sandia Joints Handbook to be something of continuing and broad utility.

The ASME BRTD has approved such an ad hoc committee for the above purposes and will consider a sustaining Joints Research Committee when it meets in June 2010.

Chapter 4

Conclusions

The 2009 International Workshop on Joint Mechanics brought together experts of diverse backgrounds and interests, but those experts were able to focus very quickly on the key technical questions. Further they were able to identify and develop a core path forward to advancing research in this key technical topic.

The creation of a formal organization to serve as a platform from which to coordinate and facilitate further work should cement progress through the authoring and editing of handbooks, standardization and propagation of new methods, and coordination of other meetings.

References

[1] Daniel J. Segalman, Lawrence A. Bergman, and David J. Ewins. Report on the SNL/NSF International Workshop on Joint Mechanics Arlington Virginia, 16-18 October 2006. Technical Report SAND2007-7761, Sandia National Laboratories, Albuquerque, New Mexico 87185, January 2008.

Appendix A

Group Photographs

Figure A.1. Group photo taken at reception on eve of workshop.

Front Row, Left to Right Arif Masud, Yvgeny Petrov, Ionel Nistor, David Nowell.
Second Row Andreas Polycarpou
Third Row Dan Segalman, Randy Mayes, David Hills, Ken Johnson, Dorothy Johnson, Jeff Green, Christoph Schwingshacki
Rear row Yin Segalman, David Ewins, Adnan Akay, Dane Quinn, Charles Bodel, Alex Vakakis, Tod Laursen, Melih Eriten, and John Schofield

Figure A.2. Workshop participants ate together in the great hall.

On left, front to back: Lothar Gaul, Ken Johnson, Dorothy Johnson, David Nowell, (face not shown), (face not shown).
On right, front to back: Alex Vakakis, Norbert Hoffmann, Ionel Nistor, Charles Bodel, (face not shown), John Mottershead.

Appendix B

Workshop Agenda

Session I: Monday morning

0900 Welcome and Intro to the Workshop

0915 Introductions of all participants

1000 Overview of the Workshop:

1030 Coffee

1045 Recap of Prior Art and Efforts

1145 Input from the Sponsors and other End Users

1230 Summing up

1245 Lunch

Session II: Monday afternoon

Theme: What does the eng. community need now/soon in terms of joint modeling, and what will it do when it has it?

1400 Short presentations

1500 Break

1515 Breakout into 3 groups

1615 Collection of group feedback and compilation of prioritized list

1715 Break

1830 Short Talks

1945- Dinner

Session III: Tuesday morning

Theme: What can the community do today – analytical, computational, experimental?

0845 Review day 1 activities and confirm day 2 agenda

0900 Short presentations

1000 Break

1015 Breakout into 3 groups

1130 Collection of group feedback and compilation of prioritized list

1230 Lunch

Session IV: Tuesday afternoon

Theme: Ideas for new developments to take current capabilities closer to deliver the community's demands

1400 Short presentations

1500 Break

1515 Breakout into 3 groups

1615 Collection of group feedback and compilation of prioritized list

1715 Break

1830 Short Talks

1945- Dinner

Session V: Wednesday morning (details to be confirmed)

0845 Review of Day 2; Plan for Day 3

0900 Short Talks

1000 Break

1015 Develop Plan of Action

1230- Lunch and Departure

Appendix C

Participants

Joints Workshop List of Attendees

Name	Institution
Akay, Adnan	University of Bilkent
Berger, Ed	University of Virginia
Bodel, Charles	EDF
Brown, Dan	AWE
Ciavarella, Mike	Politecnico di Bari
Daborn, Phil	AWE
Daniel Mulvihill	University of Oxford
Ding, Dr Jian	University of Nottingham
Dini, Daniele	Imperial College
Eriten, Melih	University of Illinois
Ewins, David	Imperial College/University of Bristol
Farris,Tom	Purdue University
Gaul, Lothar	University of Stuttgart
Gola, Muzio	Politecnico di Torino
Goyder, Hugh	Cranfield University
Green, Jeff	Rolls-Royce plc
Hillls, David	University of Oxford
Hoffman, Norbert	Hamburg University of Technology
Ind, Phil	AWE
Johnson, Ken	University of Cambridge
Laursen, Tod	Duke University
Leming, Sarah	Sandia National Laboratories
Ma, Dr Xianghong	Aston University
Masud, Arif	University of Illinois
Mignolet, Marc	Arizona State University
Mulvihill, Daniel	University of Oxford
Nistor, Ionel	EDF
Nowell, David	University of Oxford
Petrov, Yvgeny	Imperial College
Polycarpou, Andreas	University of Illinois
Quinn, Dane	University of Akron
Randy Mayes	Sandia National Laboratories
Schofield, John	Rolls-Royce plc
Schwingshackl, Christoph	Imperial College
Segalman, Dan	Sandia National Laboratories
Starr, Mike	Sandia National Laboratories
Surampudi, Surya	European Office of Aerospace Research & Development
Vakakis, Alex	University of Illinois

Appendix D

Slide Sets Presented at the SNL/AWE/NSF Joints Workshop

The scope of the workshop is illustrated by the following slide sets of presentations made there.

Imperial College
London

Introduction to the 2nd Workshop on Joints Modelling, 2009

NSF-Sandia-AWE Joints Modelling Workshop, Dartington, Devon, UK April 2009 D J Ewins

D.1 Presentations of April 27, 2009 End User Needs, Requirements and Opportunities

D.1.1 Slide Presentation of David Ewins, Imperial College, London, England: *Introduction to the 2nd Workshop on Joints Modeling, 2009*

Imperial College London

Background

Joints have long been a problem for the structural dynamicist and, increasingly, the joints are becoming the weakest link in many design analyses.

This has been recognised often and there have been many previous attempts to improve the situation. This workshop is the latest in one series of such efforts that can be traced back at least 10 years…

Imperial College London

Previous Activities

SD2000: Forum for Future Directions in Structural Dynamics
1999, Sponsored by LANL

Workshop on Predictive Models for Joints and Interfaces
2000, Sponsored by SNL

Workshop on Modelling, Analysis and Measurement for Friction Constraints in Gas Turbine Components
2001, Sponsored USAF, AFRL, AFOSR

Workshop on Benchmarks in Contact Mechanics and Friction Damping ***2002, Sponsored by USAF, AFRL, AFOSR***

Workshop on Joint Mechanics
2006, Sponsored by NSF, SNL

Imperial College London

Previous Workshop

Brought together wide range of engineers from different groups, and covered a much broader range of disciplines than had been present in the earlier workshops. That meeting a Road Map as a central feature around which to structure discussions from the macro scale down to the nano scale. There, the objective was to construct a comprehensive map of all the features that might be important in the construction of a truly predictive model for friction contact phenomena.

Imperial College London

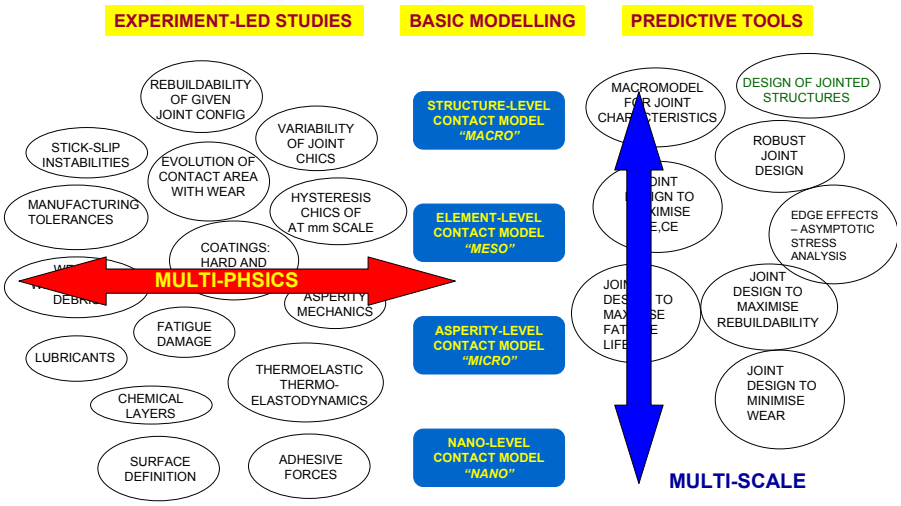
Previous Workshop

We started with…..

NSF-Sandia-AWE Joints Modelling Workshop, Dartington, Devon, UK April 2009 D J Ewins

Imperial College London

RESEARCH ROADMAP FOR FRICTION CONTACT AND WEAR IN STRUCTURES



Friction CONTACT ROADMAP v 6.3

NSF-Sandia-AWE Joints Modelling Workshop, Dartington, Devon, UK April 2009 D J Ewins

Imperial College London

Previous Workshop

and ended up with…..

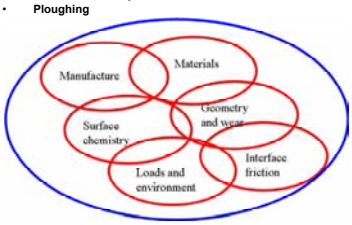
NSF-Sandia-AWE Joints Modelling Workshop, Dartington, Devon, UK April 2009 D J Ewins

Imperial College London

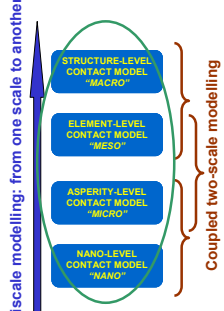
RESEARCH ROADMAP FOR FRICTION CONTACT AND WEAR IN STRUCTURES

PHYSICAL PHENOMENA

- Friction
- Materials properties
  - Elastic, plastic, visco, creep, microstructure, thermal exp
- Lubrication
- Fatigue, Fracture
- Wear
- Debris
- Thermal (heat gen?)
- Environment
- Contamination
- Surface Chemistry
- Ploughing
- Loads
- Load history
- Manufacturing
- Tolerance
- Residual stress
- Oxidation
- Corrosion
- Roughness
- Surface registration
- Adhesion (stick/slip?)
- Adhesives
- Dynamics



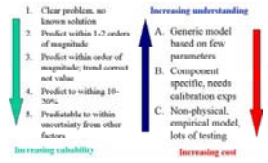
MODELS



A very difficult challenge: coupled multiscale modelling

PREDICTIVE TOOLS

- Vibration damping 3B
- Self-excitation
- Accuracy of positioning (in robots and manipulators)
- Stiffness 4B
- Hysteresis loop
- Fatigue life 2B
- Wear life 2C
- Impact strength 3C
- Temperature
  - Macro 4B
  - Micro 2C
- Heat transfer
  - Macro 4B
  - Micro 2C
- Electrical resistance ?
- Shock load transmission 3B
- Acoustic transmission ?
- Frictional limit (onset of slip) 3B
- Deformed shape (when slipped) 3B
- Surface roughness evolution
- Concept evaluation tool 1C



NSF-Sandia-AWE Joints Modelling Workshop, Dartington, Devon, UK April 2009 D J Ewins

Friction CONTACT ROADMAP v 2.

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Previous Workshop

One of the specific outcomes from the previous workshop was the formulation of three 'mini challenges':

Challenge 1: Experimental Measurements of Joint Properties

Challenge 2: Interface Physics

Challenge 3: Multi-scale Modelling

These were intended to focus attention for future research, and we shall hear shortly what has happened in the 2+ years since Washington

Imperial College London

This Workshop

We need to re-group and move ahead…..

Imperial College London

GOAL, OBJECTIVE, TASKS

GOAL

To be able to optimise design of structures with joints and interfaces from structural dynamics and integrity considerations

OBJECTIVE

To be able to construct mathematical models of joints and interfaces from conventional input data

TASKS

(a) To review the specific requirements for modelling joints in critical engineering structures and to identify future trends in joint design which will become possible with better models

(b) To review recent developments and the current state of the art of joint modelling

(c) To explore ideas for future developments in modelling methods to provide the predictive capabilities required by (a)

Imperial College London

Structure for this Workshop

Focus on 3 aspects of the subject:

A End User Needs, Requirements and Opportunities

B Current State of the Art in Joint Modelling

C New Ideas for Future Development of Joint Models

These correspond to -

A, Where do we want to be?

B, Where are we now?

C How might we get from B to A?

Imperial College London

Session I: Monday morning

Introduction

0900 Welcome and Intro to the Workshop - ***(Ewins)***
0915 Introductions of all participants
1000 Objectives of the Workshop - ***(Ewins)***
1015 Outcomes from 1st Workshop (2006) ***(Nowell, Polycarpou)***

1045 Coffee

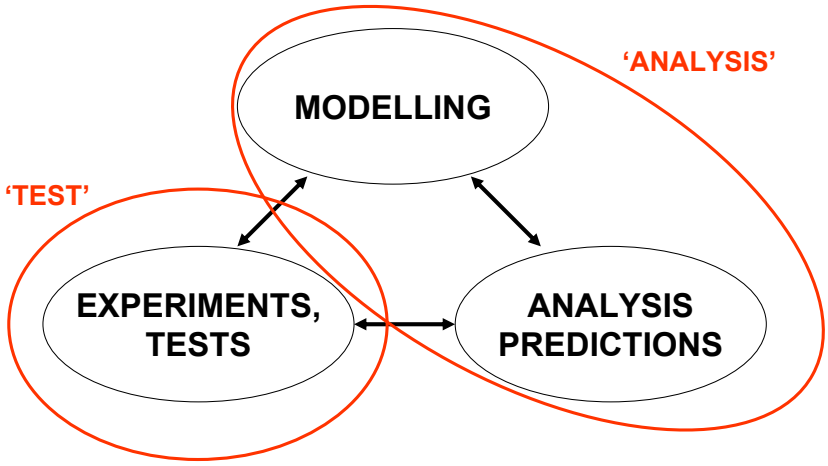
Stakeholder, Sponsor and End User Perspectives

1100 Overview of Previous Studies ***(Akay)***
1120 Industrial Perspectives from the Gas Turbine Industry (***Green/Schofield)***
1140 Sponsor Perspectives from Sandia and AWE ***(Segalman, Ind)***

1245 Lunch

Imperial College London

THE STRUCTURAL DYNAMICIST'S TOOLKIT



Imperial College

Structure for this Workshop

The structure of the Workshop is built around 3 Breakout Sessions – one for each theme - with the participants split into 3 parallel groups all addressing the same issues.

Each Session will be 'primed' by some short talks which are intended to stimulate ideas which can be debated in the ensuing small group discussions. The outcome of each Breakout Session needs to be an agreed and comprehensive statement of the issues covered by the title.

There will also be some other short talks, and posters, for the dissemination of recent work.

The Final Session will seek to reconcile the anticipated needs, current capabilities and future aspirations of the community with a view to identifying common or collaborative research activities, including benchmarking, all of which can strengthen individual bids for future funding.

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Session II: Monday afternoon

Theme A: What does the eng. community need now/soon in terms of joint modeling, and what will it do when it has it?

1400 Short presentations
- Structural assemblies ***(Vakakis for Bergman)***
- Gas turbines ***(Petrov)***
- Model Uncertainty ***(Mignolet)***
- Issues on nonlinear system identification" ***(Vakakis)***

1500 Break

1515 Breakout into 3 groups (Chairs: ***Nowell; Schofield/Green; Starr)***

1615 Collection of group feedback and compilation of prioritized list
1715 Break
1830 Short Talks – 1
Johnson; Mottershead; Farris
1945- Dinner

Session III: Tuesday morning

Theme B: What can the community do today – analytical, computational, experimental?

0900 Review day 1 activities and confirm day 2 agenda
0915 Short presentations
- Analytical Issues ***(Hills)***
- Computational Issues ***(Laursen)***
- Experimental ***(Gola)***
- Experiments towards joint modeling ***(Gaul)***

1015 Break

1045 Breakout into 3 groups (Chairs: ***Gaul; Hills; Laursen***)
1145 Collection of group feedback and compilation of prioritized list

1245 Lunch

Session IV: Tuesday afternoon

Theme C: Ideas for new developments to take current capabilities closer to deliver the community's demands

1400 Short presentations
- Experiments and modeling at microscale ***(Polycarpou)***
- Experiments and modeling at mesoscale ***(Leming)***
- Experiments and modeling at macroscale ***(Mayes)***
- Multiscale modeling of interfaces ***(Masud)***

1500 Break

1515 Breakout into 3 groups (Chairs: ***Berger; Ciavarella; Farris***)
1615 Collection of group feedback and compilation of prioritized list
1715 Break

1830 Short Talks – 2
Ciavarella; Ding; Starr

1945- Dinner

Session V: Wednesday morning (details to be confirmed)

900 Review of Day 2; Plan for Day 3

915 Short Talks – 3
Quinn; Ma; Dini

1015 Break

1045 Develop Plan of Action

1230 Lunch and Departure

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The Influence of Joints on the Dynamics of Gas Turbine Structures

David Ewins

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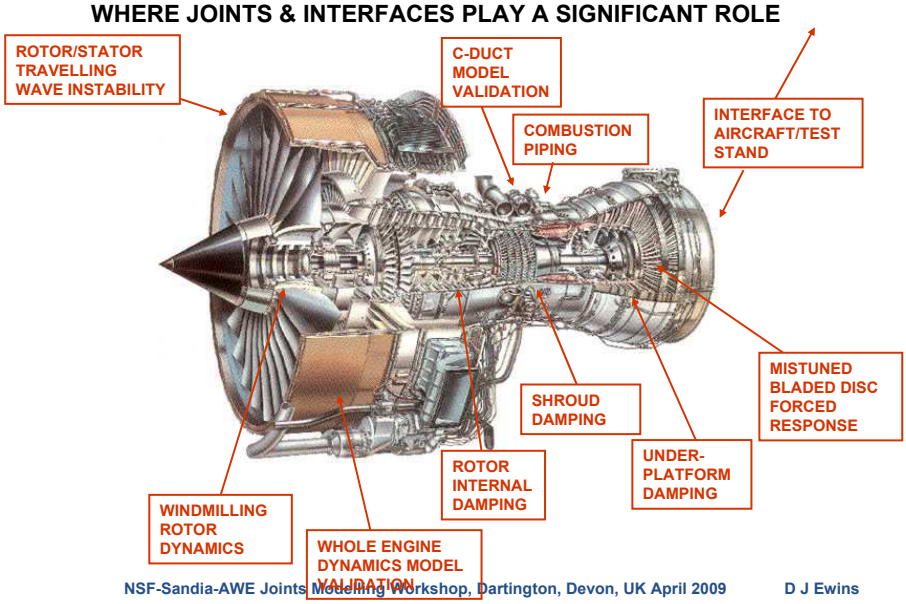
TYPICAL VIBRATION PROBLEM AREAS IN JET ENGINES WHERE JOINTS & INTERFACES PLAY A SIGNIFICANT ROLE



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The Critical Influences of Joints on the Dynamics of Gas Turbine Structures

- 'Joints' exert a non-negligible effect on the stiffness (and thus natural frequencies) and damping of all structural assemblies
- Current structural dynamic modelling capabilities are very much less advanced in respect of joints and interfaces than for any of the components that they connect
- Such models as do exist are heavily dependent on the availability of associated experimental measurements, many of which are difficult and expensive to acquire
- Consequently, the optimal design of many critical structures in gas turbines is significantly restricted by the lack of reliable predictive models of joints

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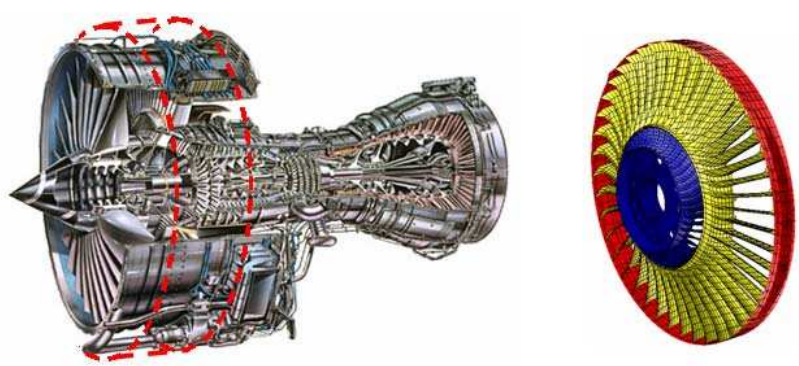

(a) (b)

Figure 4.1. Front Structure. (a) Location in the aeroengine and (b) design model.

From: Garcia, J. 'Development of Valid Models for Structural Dynamic Analysis'; PhD Thesis, December 2008

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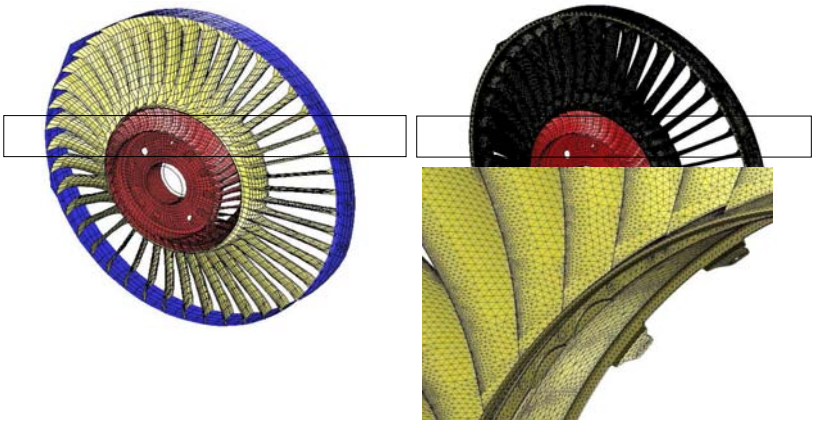
Component Models

Design Model *Supermodel*



Courtesy: Rolls-Royce

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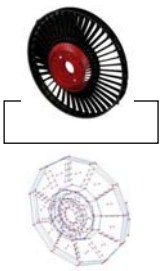
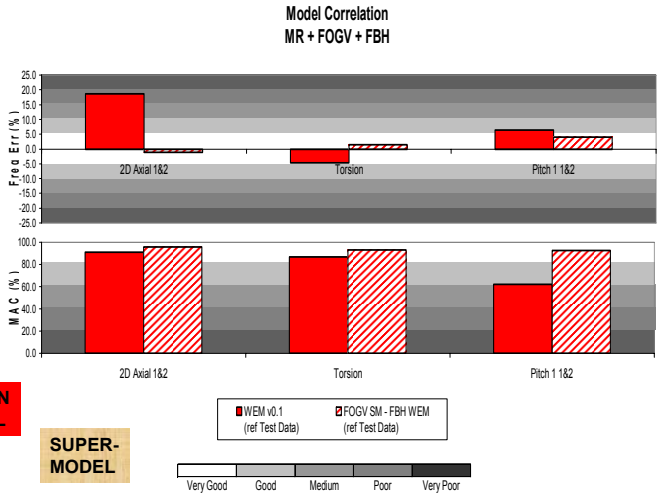
Component Models





DESIGN MODEL

SUPER-MODEL

Courtesy: Rolls-Royce

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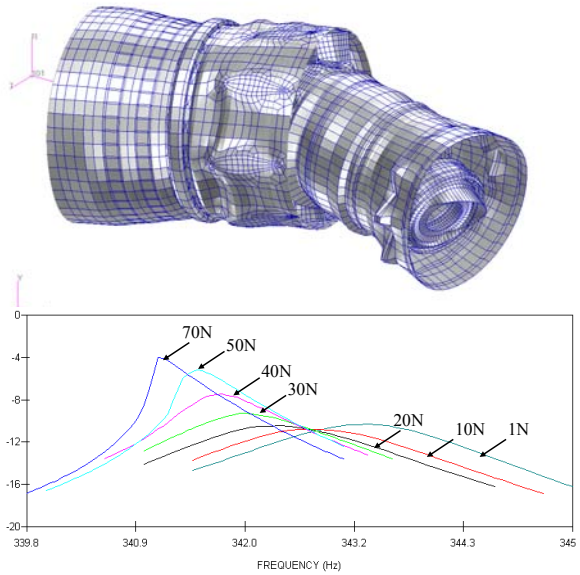
Effect of Nonlinear Joint Dynamics on Dynamic Behaviour of Engine Structures



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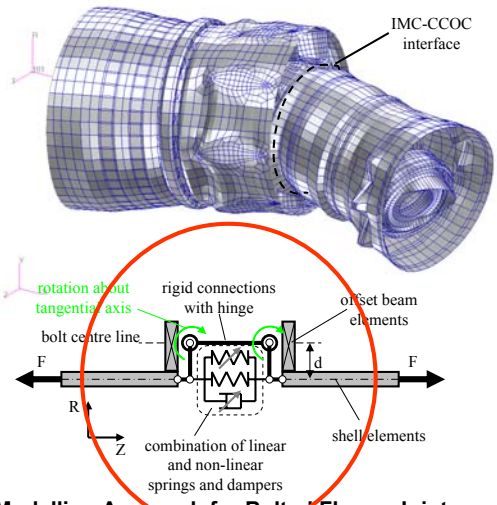
Incorporating Nonlinear Joint Behaviour into FE Models



Modelling Approach for Bolted Flange Joints

Courtesy: University of Kassel

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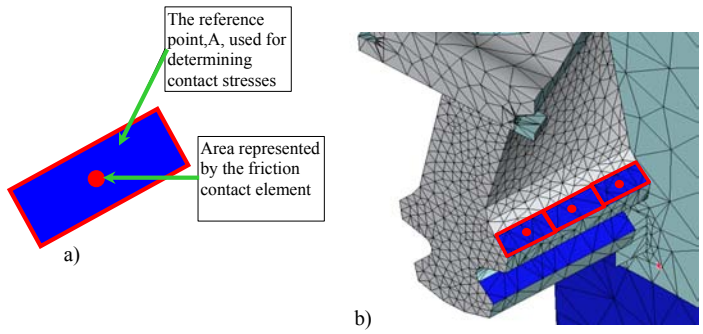
Modelling of Interaction at Contact Surfaces: Area Contact Elements



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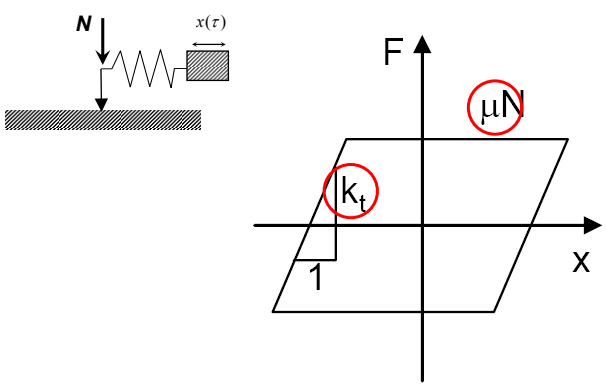
Friction Model Element Input Parameters



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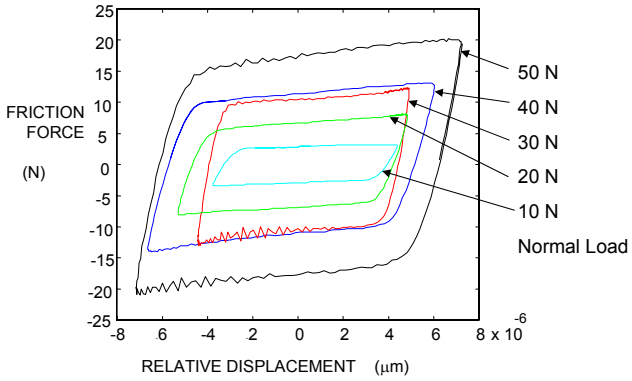
...and the role of experimental technologies therein?

A set of hysteresis loops, measured at different applied normal loads.



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The Structural Dynamics & Integrity Needs for Much Better Modelling of the Joints in Gas Turbines – 1/2

- **Current methods to account for the effects of joints and interfaces on the dynamics and integrity of gas turbine structures are basic, expensive and 'post'dictive, rather than predictive (sometimes referred to as 'retropredictive'**
- **They do not provide a full understanding of the controlling physics and, as a result, a model constructed for one particular joint cannot readily be extrapolated to another joint**
- **Today's joint models are much less advanced than those of the components which they connect**
- **The essential need for measured data inhibits attempts to use today's models to design joints so that they exhibit specific properties**

……

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The Structural Dynamics & Integrity Needs for Much Better Modelling of the Joints in Gas Turbines – 2/2

……

- **Truly predictive models for joints and interfaces are now urgently required:**

  (i) to restore a balance between the models of all the individual components in a complex structural assembly, and

  (ii) to pave the way to proactive design of joints to provide required properties (rather than simply representing characteristics that have been observed by measurement) and thereby to better optimise the design of these complex structures

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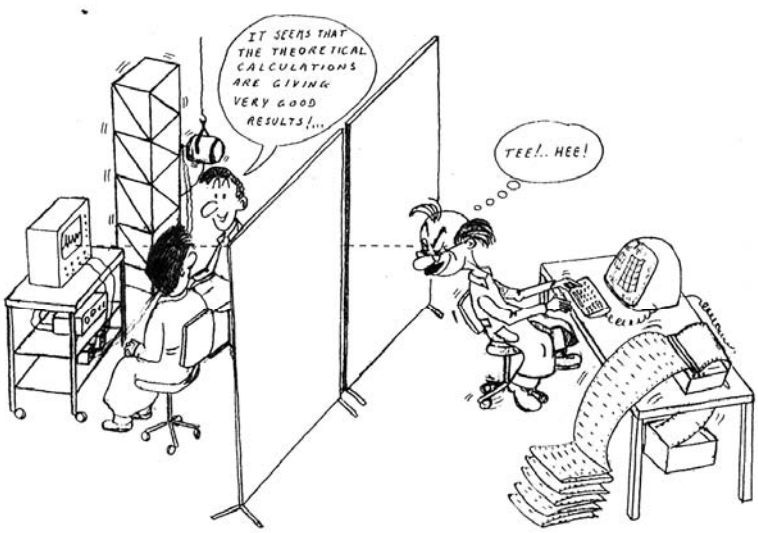

RETRO - PREDICTION

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Short Presentations

Ciavarella: **"Greenwood-Williamson roughness models with interaction" or Shakedown at frictional contacts" (B)**

Ding: **"Quantification of fretting damage via a contact-evolution based modelling approach" (B)**

Dini: **"New ideas and developments for improved modelling methods" (C)**

Farris: **"Recent Developments in Conformal Contacts" (B)**

Ma: **"The dynamics of microscale plates submerged in fluid" (C)**

Mottershead: "Nonlinear bolted-joint identification by force-state mapping" (B)

Quinn: **"Series-series Iwan models for two-sided interfaces" (C)**

Starr: **"Modeling Interfaces in a Structural Dynamics Analysis: Enriching our Joints Models and Capabilities" (A/B)**

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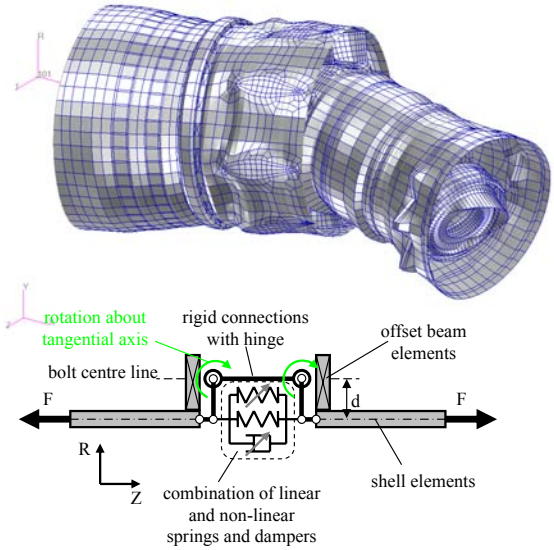
Incorporating Nonlinear Joint Dynamics Behaviour of the Structure into FE Models



Modelling Approach for Bolted Flange Joints

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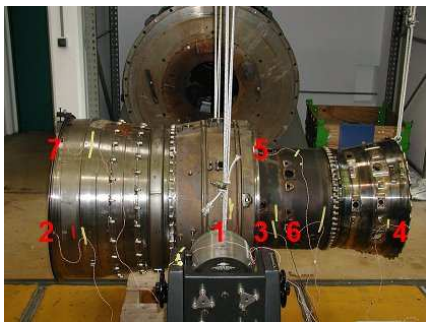
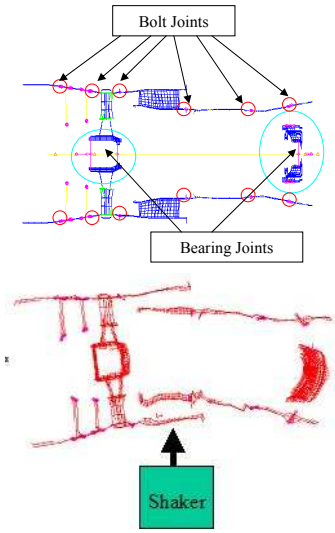
Aero-engine Casing Test Configuration





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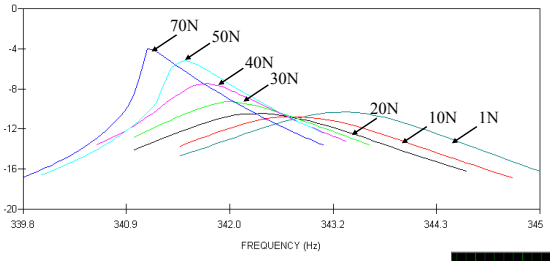
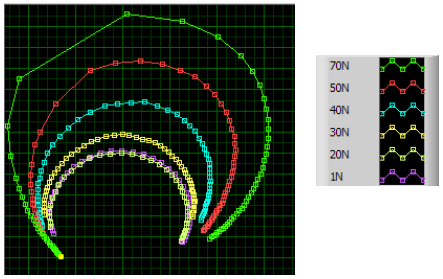
Test Data Obtained Using Force-Control Test



The first-order FRFs in Nyquist format are used to select the frequency range and frequency interval of measurement for CLV test



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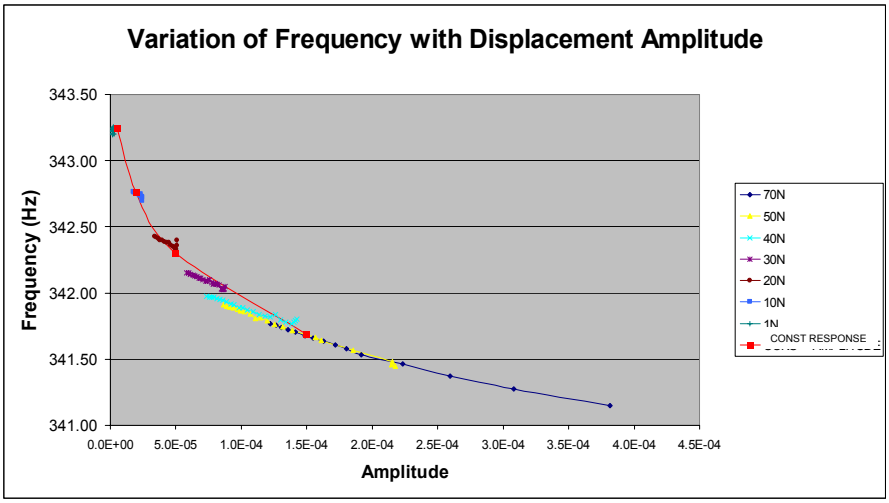

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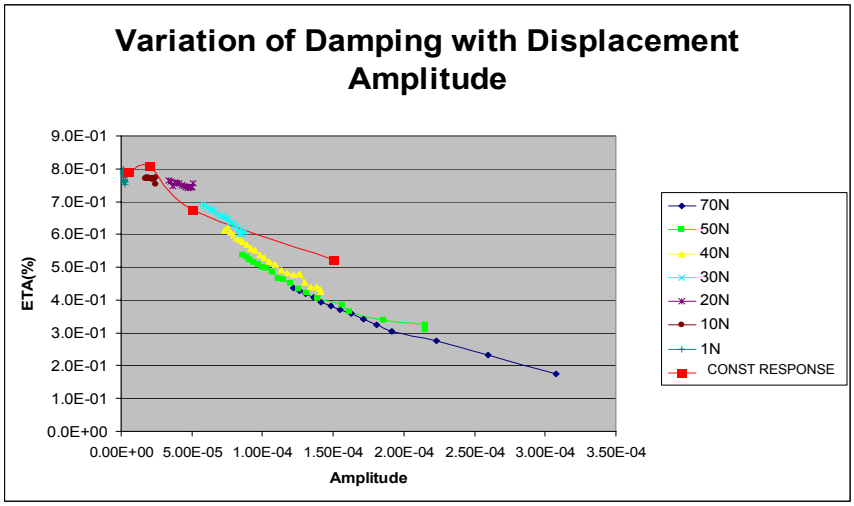

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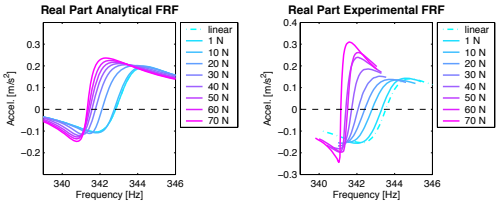
Comparison of Analytical and Experimental Non-linear FRF

ANALYSIS **EXPERIMENT**



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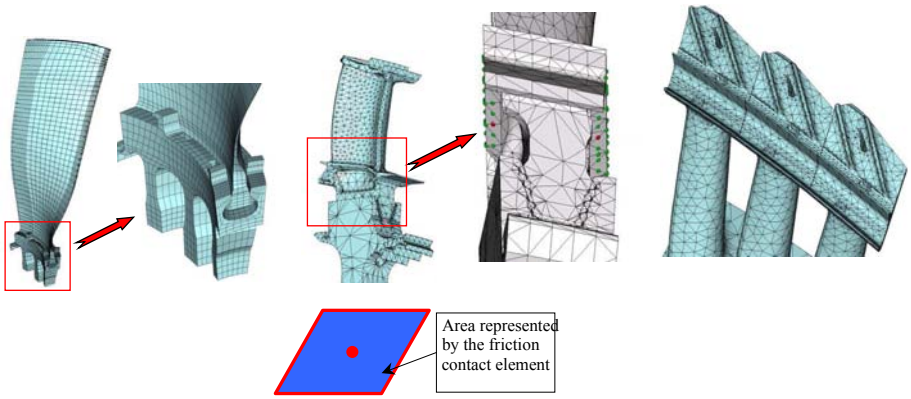
Examples of dynamic contact phenomena in bladed discs

Root damping and variable contact

Underplatform dampers

Contact of shrouds



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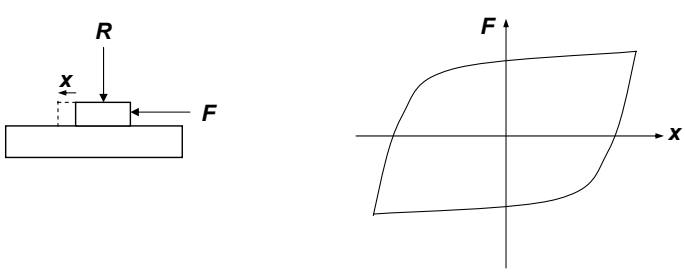
Characterization of Non-Linear Structural Elements

Non-linear, inertia-free structural components are generally characterized by a restoring force surface $F = f(x, \dot{x})$

For a friction contact it is reasonable to assume that $F = f(x, sign(\dot{x}))$ and a Force/Relative Displacement hysteresis loop is used.



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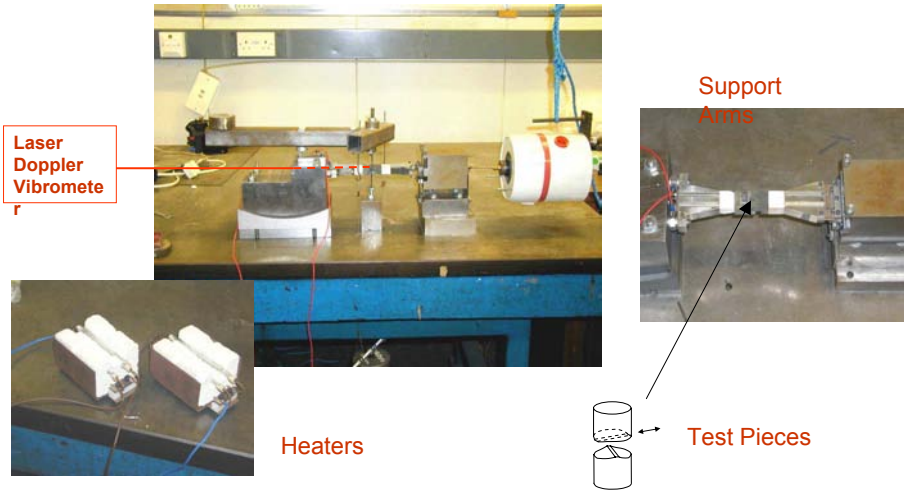
FRICTION HYSTERESIS LOOP TEST RIG.



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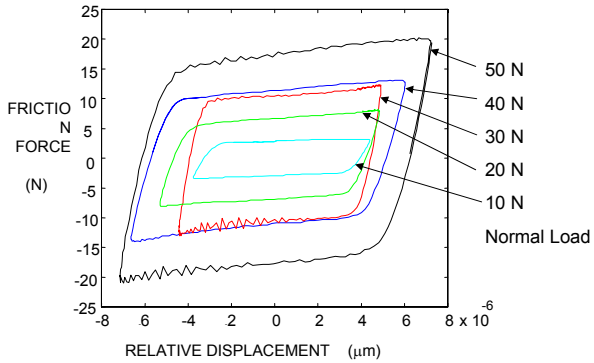
A set of hysteresis loops, measured at different applied normal loads.



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AN APPROACH TO THE TASK

Using the RoadMap as a guide,

(i) compile a list of all individual phenomena which need to be taken into account in modelling joint dynamics behaviour

(ii) Define the status of current modelling capability for each phenomenon

(iii) Develop the interdependencies between these various phenomena, and assess the status of their development

(iv) Chart possible scenarios for developing a uniform-level and consistent capability embracing all the critical phenomena, in graded stages – basic, design, advanced,…

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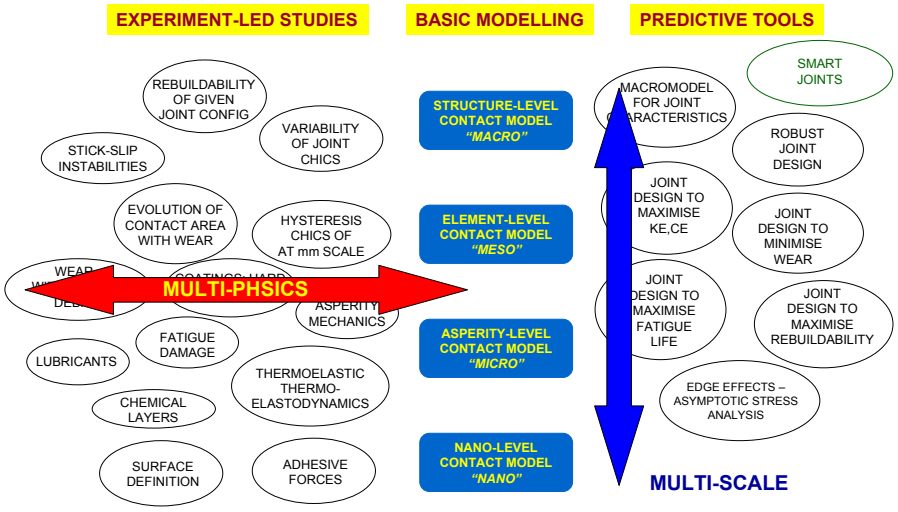
RESEARCH ROADMAP FOR FRICTION CONTACT AND WEAR IN STRUCTURES



Friction CONTACT ROADMAP v 5.1

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Plan for Day 1 - Morning

0830-0900 Review and plan for the day's breakouts

0900-1015 1st Breakout session

1015-1030 Coffee

1030-1045 Brief review of progress

1045-1145 2nd Breakout session

1145-1315 Lunch

1315-1400 Group Session: report back from breakouts.
Plan for further session

1400-1500 Breakout 3

1500-1515 Coffee

1515-(1600) Report back from 3rd Breakout

(1600)- 1700 Funding Group to meet; Group to discuss results of the day's sessions. Discuss outstanding actions. Agree plan for Day 3

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Plan for Day 2 - Morning

Task: To define the territory of the Contact Mechanics Roadmap

0830-0900 Briefing, plan for day

0900-1000 Breakout session 1:

- **Review list of Topics on Roadmap**
- **Produce definitive and comprehensive list of Research Themes (necessary to cover all the phenomena that will/may be necessary to include in a universal contact/joints/interface mechanics models)**
- **Assess the current status of development of each theme (re the availability of the basis of a mathematical model of that phenomenon)**

1000-1015 COFFEE

1015-1130 Breakout session 2
Define the interdependencies of each of these themes, showing sequencies as appropriate

1130-1145 End of morning briefing

1145-1315 LUNCH

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Plan for Day 2 - Afternoon

1315-1415 Group Discussion
Report back from Breakouts 1 and 2. To compile first version of New RoadMap

1415-1515 Breakout session 3
To chart possible routes through the map which emerges from 1 & 2
To indicate priorities, and perhaps develop ideas for phases of development

1515-1530 COFFEE

1530 – 1600 Group Discussion
To put together the three parts into the first draft of the overall RoadMap

1600 – 1700 Breakout session 4
Funding agencies group to consider the result and to develop comments, questions, suggestions for additinal information. What do the agencies look for from a workshop like this?

Rest of group . Discuss scale interface issues; to discuss the whole plan, and to compile a list of known research groups active in each of the research theme areas Also, to discuss procedures for day 3

1700-1715 Group Discussion (main group rejoined by Funding Agencies group)
Summarise Day 2 & Agree plan for day 3

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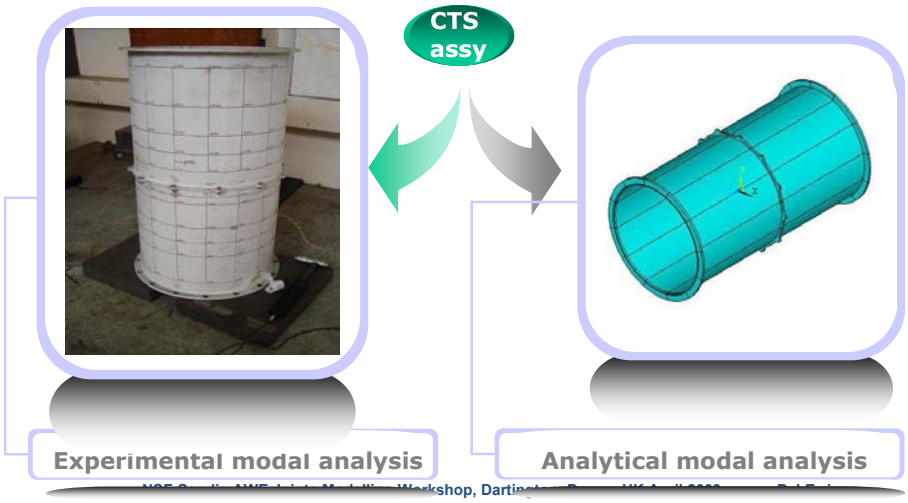
CTS Component I



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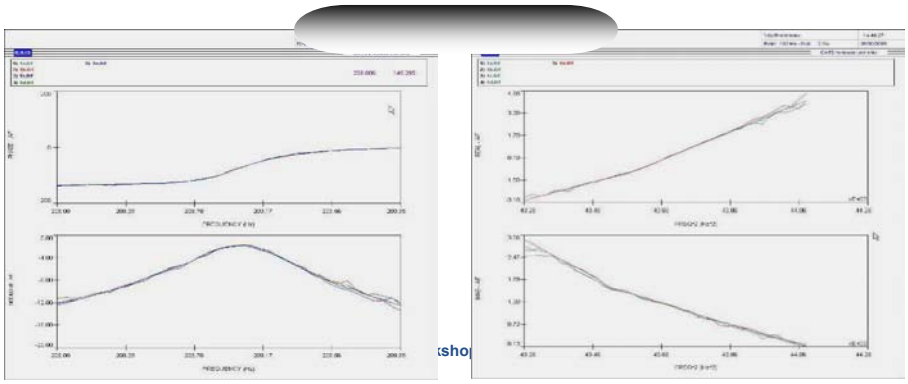
Repeatability

5 runs

- **Very little deviation**
- **Very good repeatability**
- **No change in parameters**



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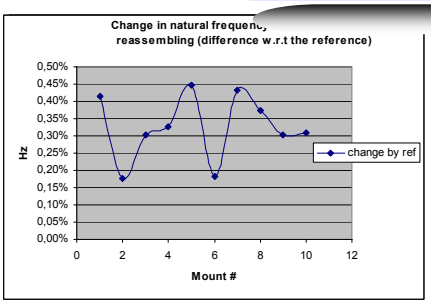
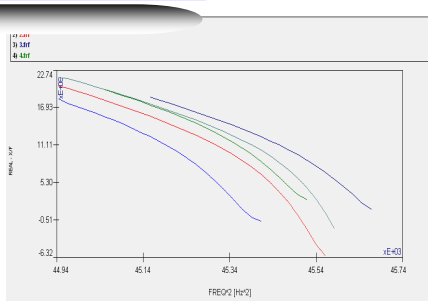
Assembling – Disassembling the structure

5 runs
Significant deviation
- **Important change in parameters**





- **No global parameter changing** : Tightening Torque constant, same relative positions
- **Consequence:** change in the joint parameters

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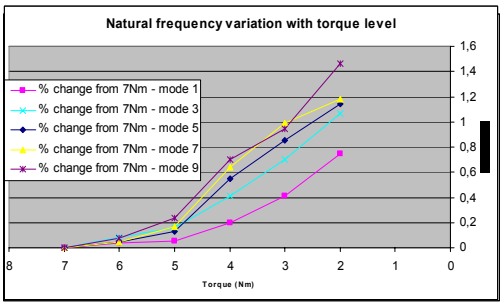
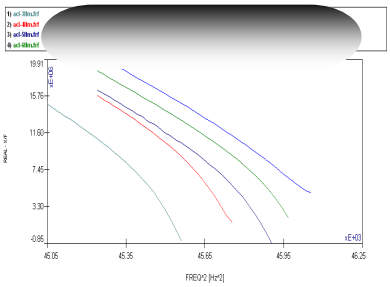
Influence of Tightening Bolt Torque

- **Shift** of the natural frequencies toward **lower frequencies**
- **Lower amplitudes** with lower tightening torque (more energy dissipated in friction)

5 runs

- Significant deviation
- Change in parameters
- Tightening tension is key to control parameter variability





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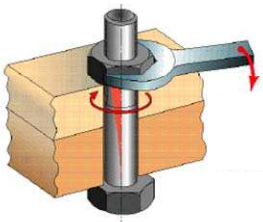
Influence of Tightening Bolt Torque



5 runs

- Very little deviation
- Very good repeatability
- No change in parameters

- **But the control in the tightening isn't really possible** : 20% change
- **Consequence:** no actual control of the joint parameters

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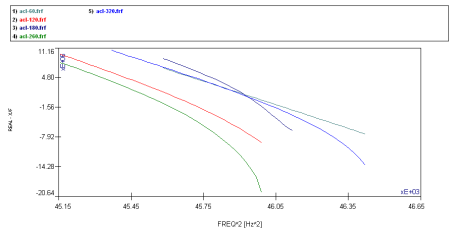
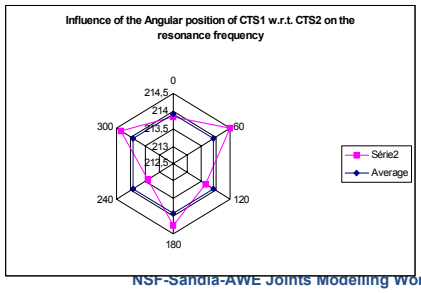
Influence of the Angular Position

5 runs

- Very little deviation
- Very good repeatability
- No change in parameters





- **No global parameter changing** : Tightening Torque constant, relative positions changing
- **Consequence:** change in the joint parameters

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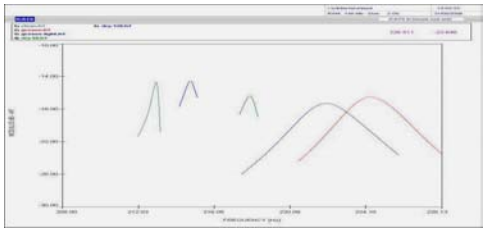
Influence of Interface Conditions



5 different conditions

- Significant deviation
- Strong influence on the damping
- Significant influence on the parameters

- **Nature of the interface changing** : Tightening Torque constant, relative position changing
- **Consequence:** change in damping

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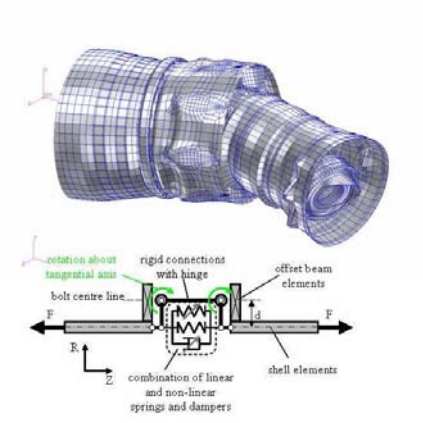
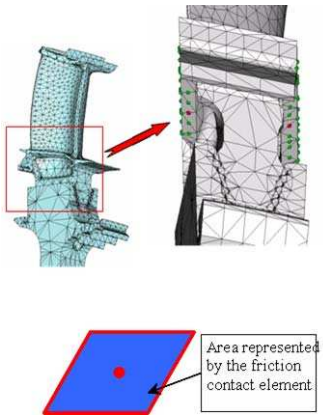
Two Areas of Particular Interest & Concern: Whole-engine Casings & Bladed Assemblies





D.1.2 Slide Presentation of David Nowell, University of Oxford: *Progress Since Washington Meeting*

Context

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- Previous Joints Workshops
  - New Orleans, 2001
  - West Palm Beach, 2002
  - Washington (Arlington, Virginia), 2006
- Report from Arlington meeting contained 3 'challenges'
  1. Experimental Measurement of Joint Properties
  2. Interface Physics
  3. Multi-scale modelling
- Formal progress on these challenges at a national/international level has been limited
  - But individual projects are taking place

Challenge 1 – Experimental Measurement of Joint Properties

Imperial College London

- Standardisation of experimental techniques
  - Round-robin exercise to measure frictional hysteresis loops for a well-characterised material pair
- 'Top-down' Modelling
  - Draw on results of above to produce a 'top-down' model of the contact, based on an interface constitutive law
  - Use this to predict hysteresis loop in a different configuration (e.g. different roughness/different material pair/different geometry)

The PAMFJP project

Imperial College London

- Predictive Approach to Modelling Frictional Joint Performance
- UK project, Funded by EPSRC
  - Collaborators: Imperial College London and University of Oxford
  - 4 years: October 2007 – Oct 2011
  - Research Assistant and Research Student at each institution.
  - Total Funding £0.75 million
- Industrial collaborators – Rolls-Royce plc and AWE

EPSRC UNIVERSITY OF OXFORD Imperial College London

Staff

Imperial College London

- Academic Staff
  - Prof David Ewins, Dr Evgeny Petrov, Vibrations – Imperial College
  - Dr Andy Olver, Dr Daniele Dini, Tribology – Imperial College
  - Prof David Nowell, Prof David Hills, - Solid Mechanics, Oxford
- Research staff and students
  - Simon Medina, Daniel Propentner, Christoph Scwingshackl, - Imperial
  - Mehmet Kartal, Daniel Mulvihill, - Oxford

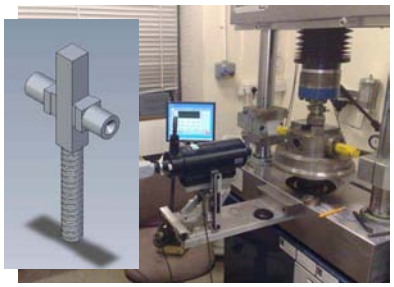
Main work tasks

Imperial College London

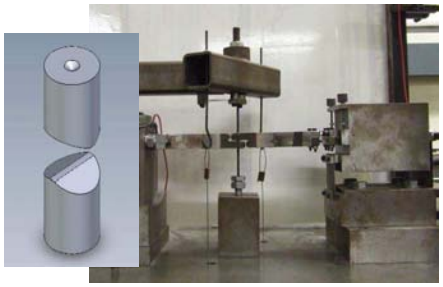
- Carry out independent hysteresis loop measurements on IC and Oxford equipment
  - Correlate results for friction coefficient and contact stiffness
- Development of physical understanding
  - Including measurements using SLIM apparatus
- Numerical modelling
  - Asperity level
  - Multi-asperity rough contact
- Validation
  - Prediction of response in different configuration

Oxford and Imperial rigs

Imperial College London



- 80 mm^2 flat and rounded contact
- 1Hz Frequency
- 0.6mm sliding distance
- Displacement measurement by remote LVDT or digital image correlation



- 1 mm^2 flat on flat contact
- ~100Hz Frequency
- 30μm sliding distance
- Displacement measurement integration of LDV measurements

Materials

Imperial College London

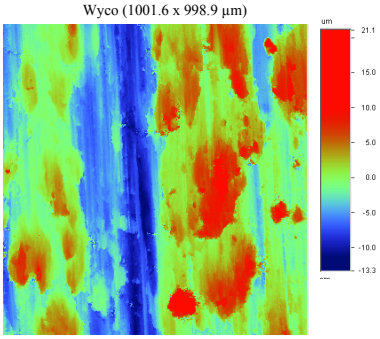
- Three material pairs chosen:
  - Ti6/4 ‘smooth’ ground
  - Ti6/4 ‘rough’ ground
  - Udimet 720 ‘smooth ground
- Specimens manufactured at Oxford to give ‘same’ surface finish for both specimen geometries
- Roughness of untested specimens measured at Imperial (Wyco) and Oxford (Alicona)

Roughness measurements Alicona-Wyco Comparison

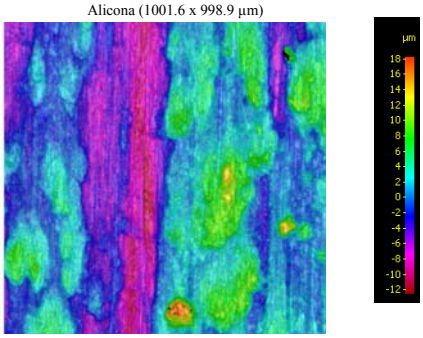
Imperial College London

Titanium (TB58)	Wyco	Alicona
S_a	3.60 µm	3.56 µm
S_q	4.51 µm	4.47 µm
S_z	29.55 µm	30.76 µm
S_{ku}	2.91 µm	2.9 µm
S_{sk}	0.10 µm	0.073 µm

Wyco (1001.6 x 998.9 µm)



Alicona (1001.6 x 998.9 µm)



Comparison of Mean Roughness Values

Imperial College London

	S_a	S_q	S_P	S_v	S_Z	S_{10Z}	S_{sk}	S_{ku}	S_k	S_{pk}	S_{vk}
Titanium (Smooth) ALICONA	1.19	1.49	7.80	7.06	14.87	13.06	-0.012	3.04	3.76	1.34	1.43
Titanium (Smooth) WYCO	1.04	1.27			14.39						
Nickel ALICONA	1.54	1.88	8.92	7.88	16.79	15.46	0.038	3.07	4.68	1.72	1.76
Nickel WYCO	1.36	1.72			26.27						
Titanium (Rough) ALICONA	2.13	2.74	12.74	10.58	22.79	20.42	0.54	3.49	6.15	3.61	2.21
Titanium (Rough) WYCO	2.53	3.11			35.89						

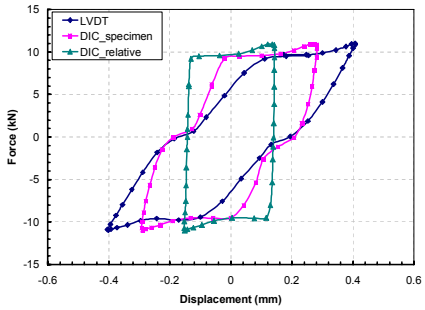
•Wyco measurements were taken as average values of two perpendicular thin strips 574 x 6999 µm

*All values in micrometers (µm)

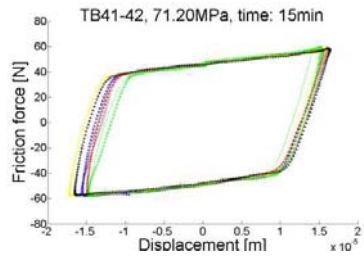
Hysteresis loop measurements

Imperial College London

Oxford



Imperial



- Both rigs show some change with time
  - Significant wear
- Similar features observed
  - E.g. rise in force during sliding phase
- Results obtained allow comparison of friction coefficient and stiffness values
  - Some issues still to be addressed (definition of μ, time, normalisation of stiffness

Comparison of results (provisional)

Imperial College London

- Contact stiffness (N/m/mm of contact area)

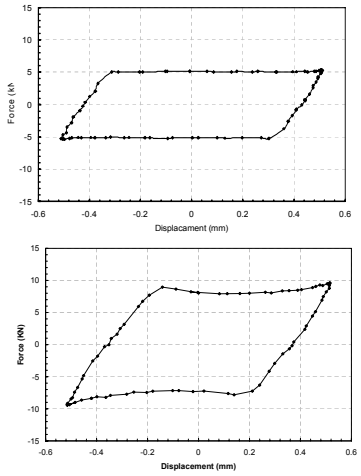
	Ti Smooth	Ti Rough	Nickel
Imperial	1.8×10^7	2.6×10^7	4.8×10^7
Oxford	Not yet measured with DIC	3.4×10^7	2.0×10^7

- Friction coefficient (after approx 3m sliding distance)

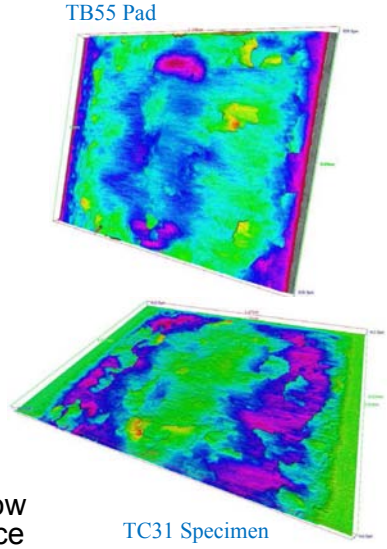
	Ti Smooth	Ti Rough	Nickel
Imperial	0.67	0.67	0.67
Oxford	0.61	0.71	0.69

Variation of friction during test

Imperial College London



TB55 Pad



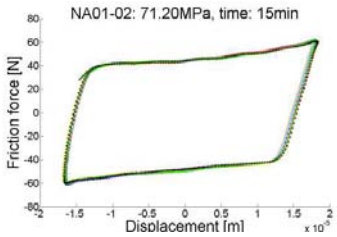
TC31 Specimen

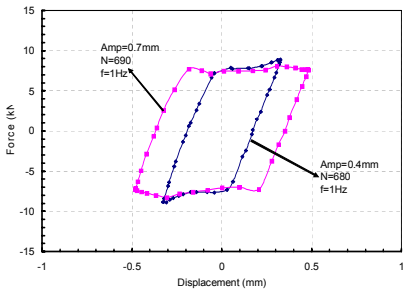
- Roughness measurements show quite severe wear is taking place

Non-uniform friction during sliding

Imperial College London

- Particularly in worn state, friction increases during sliding part of cycle
  - Seems to be associated with contact registration (macroscopic or microscopic?)
  - We have also discussed velocity dependent friction
  - Some variable amplitude tests carried out
  - Further work to be done

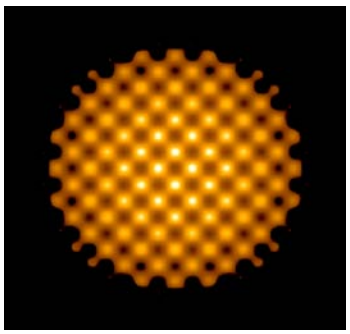




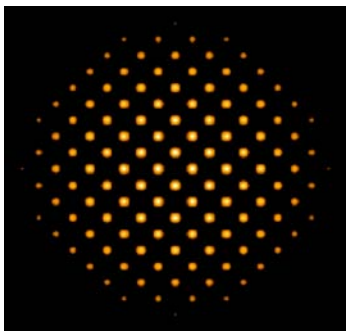
Modelling

Imperial College London

- Simon Medina (IC) has developed model for elastic contact of rough surfaces using Venner/Lubrecht approach
- Coulomb friction can be included using Ciavarella method for partial slip



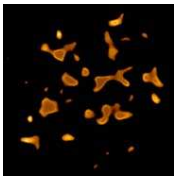
Amplitude 0.05 um



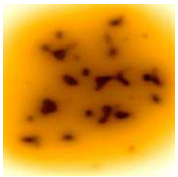
Amplitude 0.75 um

Partial slip modelling

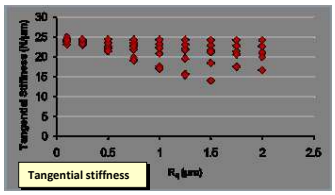
Imperial College London



Tractions



Slip

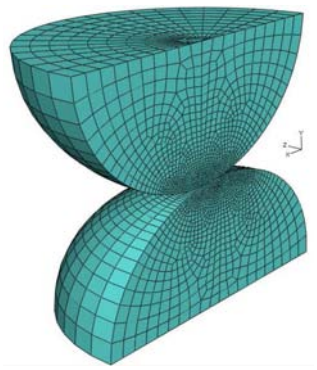


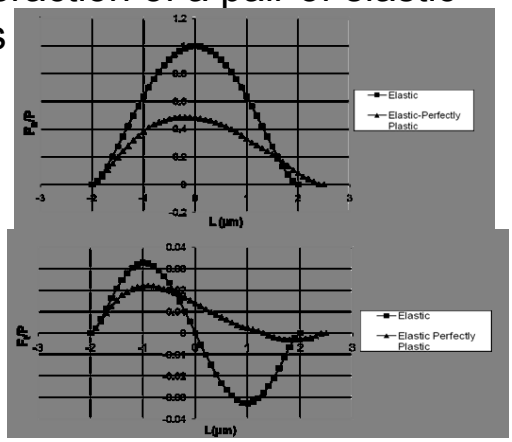
Currently possible	Further development req.d
Normal contact for any profile with/without adhesion and simplified plasticity [composite surface assumption] Tangential loading (incl. sequences) for non-adhesive contacts Prediction of friction loops for "suitable" contact geometries based on measured friction coefficient	Prediction of friction coefficient based on generalised measurable data Full plasticity solution Accounting for wear, and wear particles within contact Prediction of friction loops for contacts such as Imperial and Oxford test rigs Tangential loading with adhesion

Modelling at the asperity scale

Imperial College London

- Daniel Mulvihill (Oxford) has undertaken FE modelling of interaction of a pair of elastic-plastic asperites





Conclusions

Imperial College London

- Challenge 1 from Arlington Meeting has not yet been addressed internationally in a co-ordinated way
- However, it has been used nationally as the basis for collaborative projects
- An example of this is the Oxford/Imperial PAMFJP project
- Significant progress made in understanding and correlating experimental measurements on different rigs
- Work still ongoing in modelling at a single asperity or multi-asperity level
- Wear is more significant than was originally thought when defining the project

Workshop
Friction Damping in Joints

Dartington, Totnes, UK
27-29 April 2009

Adnan Akay
Bilkent University, Ankara, Turkey
Carnegie Mellon University, Pittsburgh, PA, USA

D.1.3 Slide Presentation of Adnan Akay, Carnegie Mellon University: *Modeling Joints*

Modeling Joints

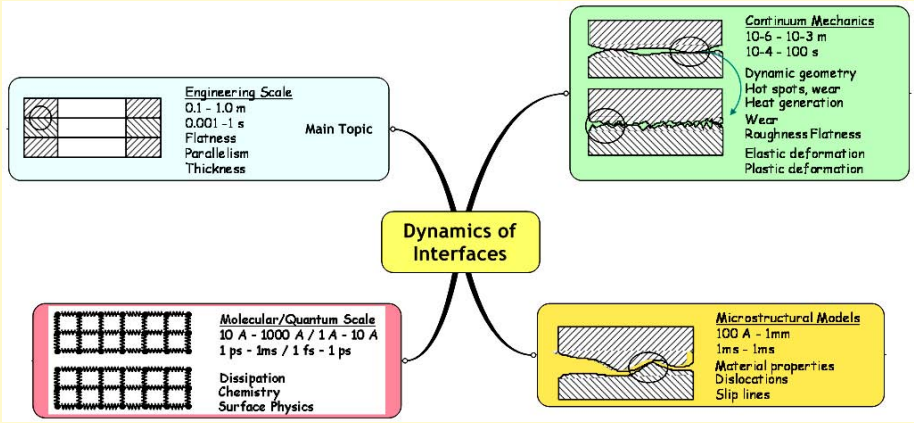
- The Challenges
  - Degree of accuracy of the models
  - Degree of repeatability of the measurements
- The Challenge for the workshop
  - Quantify the desired accuracies

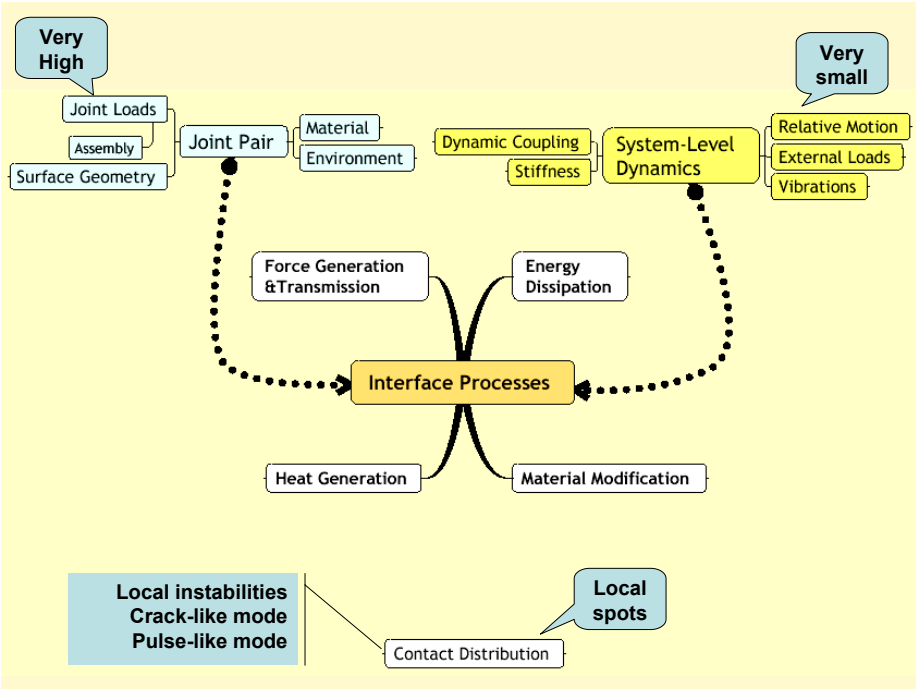
Previous Studies

Sponsors Have Different Needs & Approaches

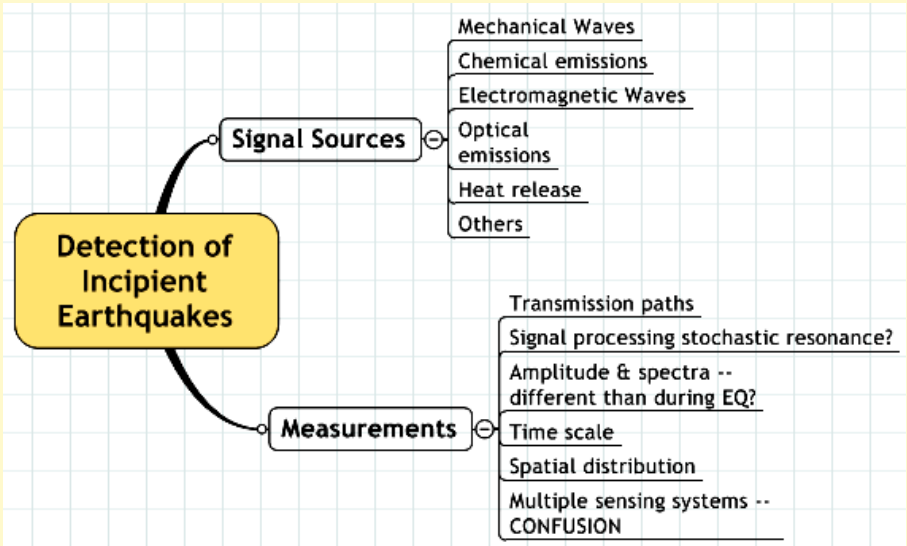
- NSF
  - Research Needs in Tribology (c. 1980)
  - Research Needs in Acoustics & Noise Control (c.1980)
  - Friction and System Dynamics (1994)
  - Friction and System Dynamics (1997)
  - Friction & Turbulence – Analogies and Contrasts (2008, 2009)
- NSF-Sandia
  - Modeling of Joints (2006)*
- ONR
  - Structural Acoustics (~1980s)
- AFOSR
  - Joint Damping (~ 1990s)*
- NATO ASI
  - 1991

DYNAMIC EFFECTS DEVELOP AT ALL LENGTH AND TIME SCALES





An Extreme Joint - Tectonic Plates



G. H. HEILMEIER'S "CATECHISM"
FOR EVALUATING A RESEARCH PROJECT

- What are you trying to do? (Articulate your objectives using absolutely no jargon.)
- How is it done today and what are the limits of current practice?
- What is new in your approach and why do you think it will be successful?
- Who cares? If it is successful, what difference will it make?
- What are the risk and the payoffs?
- How much will it cost? How long will it take?
- What are the midterm and "final" exams to check for success?

From E. Greitzer

Rolls-Royce

Joints - Rolls-Royce Perspective

Dr John Schofield / Dr Jeff Green

Joints Workshop April 2009.

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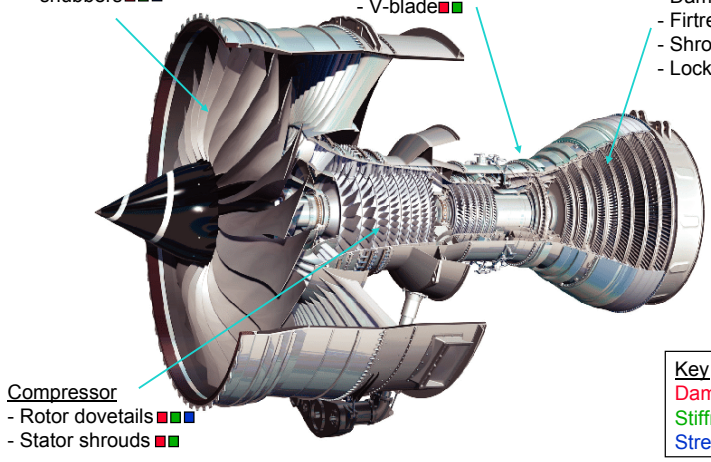
D.1.4 Slide Presentation of John Schofield and Jeff Green, Rolls-Royce: *Joints–Rolls-Royce Perspective*

Joints in Gas Turbine



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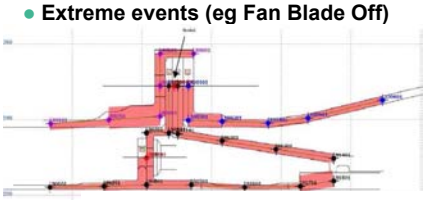
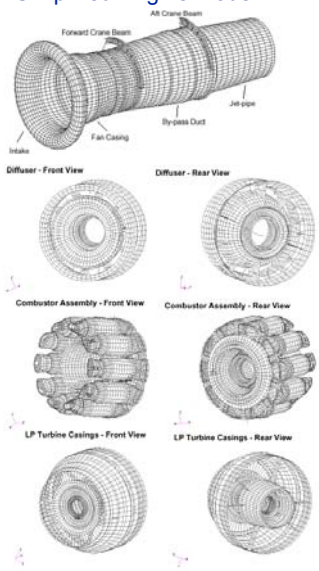
Whole Engine Issues

- **Whole engine uses a simplified model.**
  - Need simplified respresentation of joints.
- **Static loads (thrust / external loads / manoeuvres)**
  - Are stiffness effects adequate for tip clearances and load distribution etc?
- **Dynamic Loads**
  - **Engine/Wing Dynamics (0-10 Hz)**
    - Frequency, damping, loads
  - **Engine Rotordynamics (30-500 Hz)**
    - Frequency, damping, loads
  - **Extreme events (eg Fan Blade Off)**



Example Joint (and simplification)

Simplified Engine Model



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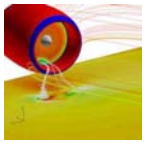
Damping

- **Drive towards prediction of vibration amplitude for design and certification.**

Structure
(known)



Force
(usually known)

Damping
(aero & mechanical)
(non-linear)
(measured ?)

Response

HCF
capability

Wear / deterioration /
contamination

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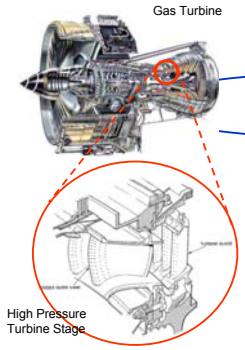
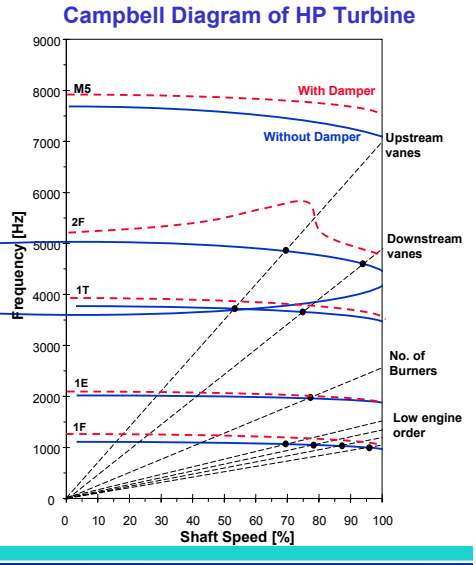
Effect of Non-linear Contact on Frequency

- Dampers can have a significant influence on resonant frequency.
  - Affect on resonant speed
  - Change in force amplitude





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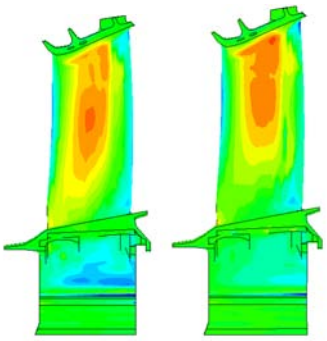
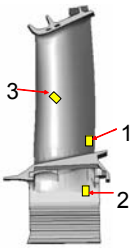
Effect of Joint on Modeshape => Stress

Contours of WP Stress At 2F resonance



No damper With damper

Typical position of strain gauges In engine tests.



	Gauge Position	Gauge Sensitivity [Mpa/mm]	
		1T Mode	2F Mode
Undamped	1	579	**779**
	2	156	**394**
	3	**731**	354
Damped	1	348	**876**
	2	62	**277**
	3	**705**	265

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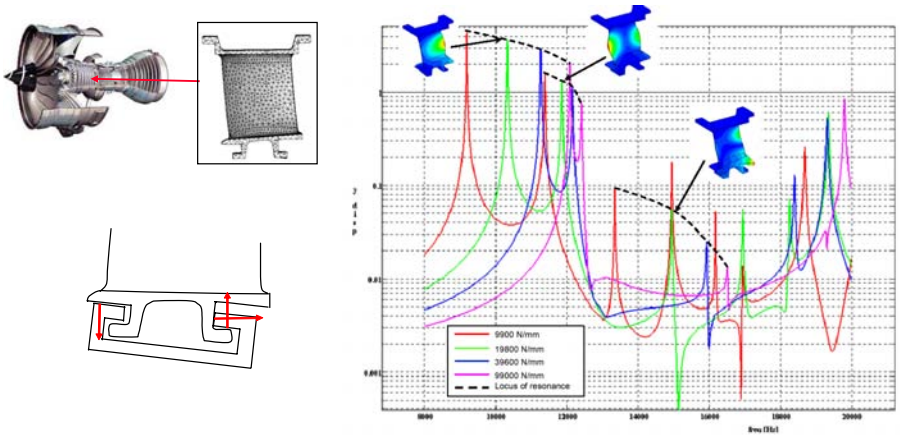
Effect of Shroud Contact on Stator Response



- Uncertainty about inner shroud restraint => variability in effective stiffness
- Change of stiffness leads to change in amplitude and frequency.
- Difficulty interpreting measured results

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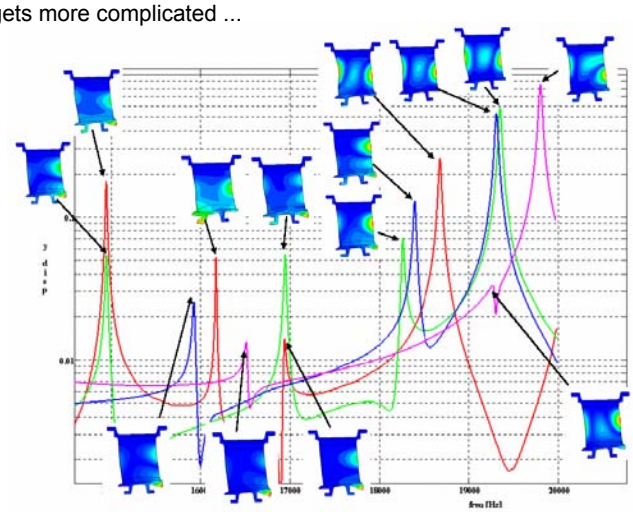
Effect of Shroud Contact on Stator Modes

- It gets more complicated ...



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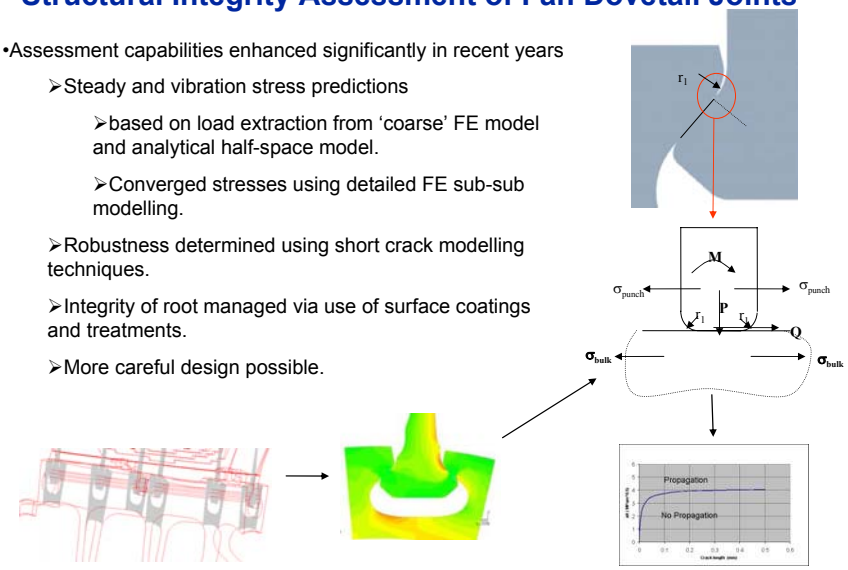
Structural Integrity Assessment of Fan Dovetail Joints

- Assessment capabilities enhanced significantly in recent years
  - Steady and vibration stress predictions
    - based on load extraction from 'coarse' FE model and analytical half-space model.
    - Converged stresses using detailed FE sub-sub modelling.
  - Robustness determined using short crack modelling techniques.
  - Integrity of root managed via use of surface coatings and treatments.
  - More careful design possible.



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Welcome and Charge

Joints Workshop
Dartington, UK
AWE/SNL/NSF

27 April 2009

Dan Segalman (SNL) and Phil Ind (AWE)

Sandia is a multi-gram laboratory operated by Sandia Corporation, a Lockheed Martin Company,
for the United States Department of Energy's National Nuclear Security Administration
under contract DE-AC04-94AL85000.

D.1.5 Slide Presentation of Dan Segalman and Phil Ind, Sandia National Labs and AWE: *Welcome and Charge*

Why SNL and AWE are Funding This

- These issues are very important to us and to others.
- This is a very difficult problem class. We shall not solve it on our own.
- It makes sense to excite interest in these problems in the general research community.

Sandia National Laboratories

Accounting for Joint Mechanics is Prerequisite to Prediction. Where We *Must* be Predictive

Where correct answers are necessary and either experiments are just too expensive or are impossible

- **satellites**
- **next generation space telescopes**
- **jet engines and jet engine failure**
- **nuclear weapons systems**

Sandia National Laboratories

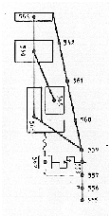
Traditional Barriers to Predictive Modeling

- **Discretization error**
- **Uncertainty in Material Properties**
- **Uncertainty in loads/boundary conditions**
- **Missing Physics - Interface Mechanics (Joints)**

We are not predictive?

Sandia National Laboratories

Discretization Error: Less of an Issue Now Than in the Past



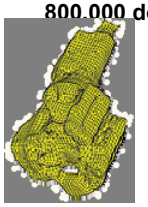
20 years ago:
Shellshock 2D
NASTRAN
200 dof

15 years ago:
NASTRAN
MC2912
30,000 dof

(Original Picture might be OUO if shown)

10 years ago:
- Fine Meshes of Subsystems
- Or Coarse Meshes of Systems

800,000 dof



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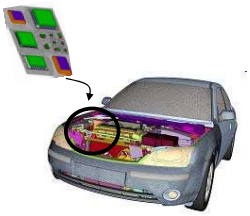
Today:
SALINAS MP
>10M dof.



(Original Picture might be OUO if shown)

5

Traditional Barriers to Predictive Modeling

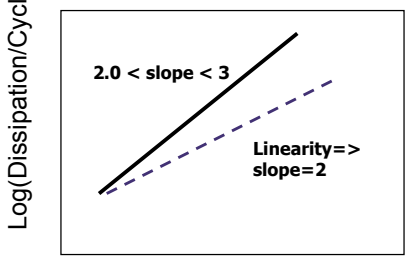
- **Discretization error**
  - **Mitigated substantially by MP technology**
- **Uncertainty in Material Properties**
  - **Subject of separate research efforts**
- **Uncertainty in loads/boundary conditions**
  - **Better measured, calculated, or bounded**
- **Missing Physics**
  - **Interface Mechanics (Joints)**
  - **The Tall Pole in the Tent**
  - **Topic of this workshop**

Topics include misfit, interference, and variability

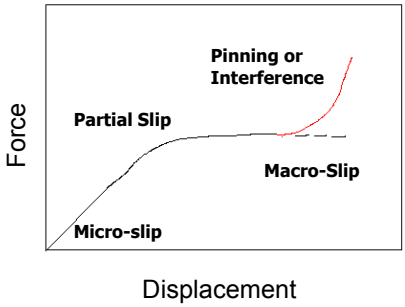
6

Empirical Nonlinearity of Joints

Dissipation from Base Excitation or Free Vibration



Monotonic Pull

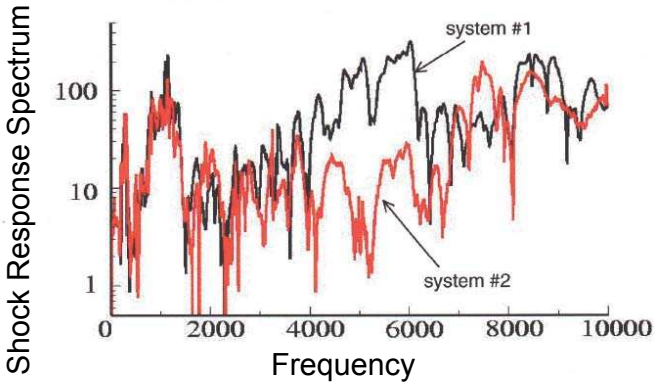


Nonlinearities even at Small Displacement

Large Displacement

7

Example of Variability Due to Joints



8

Further Complications of Joint Mechanics to Structural Dynamics

- Part-to-Part and System-to-System Variability
- Aging Effects – a little oxidation or rubbing goes a long way.

- How are we expected to predict dynamic properties of systems put into the stock pile years ago?

9

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Standard Practice for Ignoring the Nonlinearity of Joints in Structural Dynamics

How we traditionally do structural dynamics analysis

Analyst creates coarse mesh of model putting tunable springs at interfaces and postulating proportional/modal damping

10

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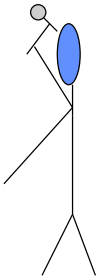
Standard Practice for Ignoring the Nonlinearity of Joints in Structural Dynamics

How we traditionally do structural dynamics analysis

Build full structure or subsystem and test in modal lab at relevant amplitudes



Analyst creates coarse mesh of model putting tunable springs at interfaces and postulating proportional/modal damping

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Standard Practice for Ignoring the Nonlinearity of Joints in Structural Dynamics

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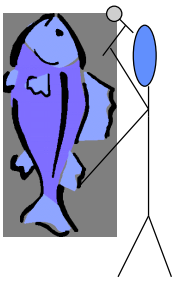

Analyst creates coarse mesh of model putting tunable springs at interfaces and postulating proportional/modal damping

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Standard Practice for Ignoring the Nonlinearity of Joints in Structural Dynamics

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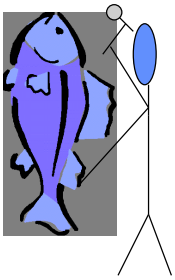

Analyst tunes joint stiffness and modal damping to match test. He then makes prediction

13

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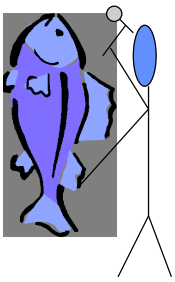
Standard Practice for Ignoring the Nonlinearity of Joints in Structural Dynamics

How we traditionally do structural dynamics analysis

Analyst creates coarse mesh of model putting tunable springs at interfaces and postulating proportional/modal damping

Build full structure or subsystem and test in modal lab at relevant amplitudes

Systems test is performed on updated model



Analyst tunes joint stiffness and modal damping to match test. He then makes prediction

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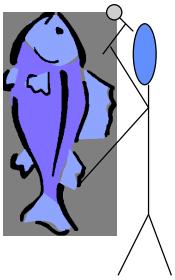
Standard Practice for Ignoring the Nonlinearity of Joints in Structural Dynamics

How we traditionally do structural dynamics analysis

Analyst creates coarse mesh of model putting tunable springs at interfaces and postulating proportional/modal damping

Build full structure or subsystem and test in modal lab at relevant amplitudes



Analyst tunes joint stiffness and modal damping to match test. He then makes prediction

Systems test is performed on updated model

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Standard Practice for Ignoring the Nonlinearity of Joints in Structural Dynamics

Elements of Process

- Assume system to be linear
- Represent each joint DOF as a linear spring
- Build and test a prototype structure
- Tune the spring stiffnesses to match frequencies
- Tune modal (or more complicated) damping to match damping of structure

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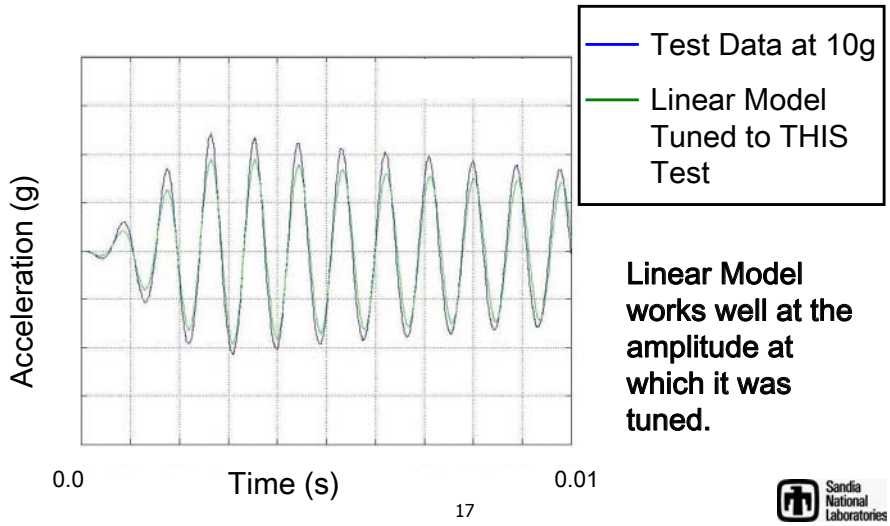
How Well Does a Linear Model Do when Tuned to a Given Experiment?



Linear Model works well at the amplitude at which it was tuned.

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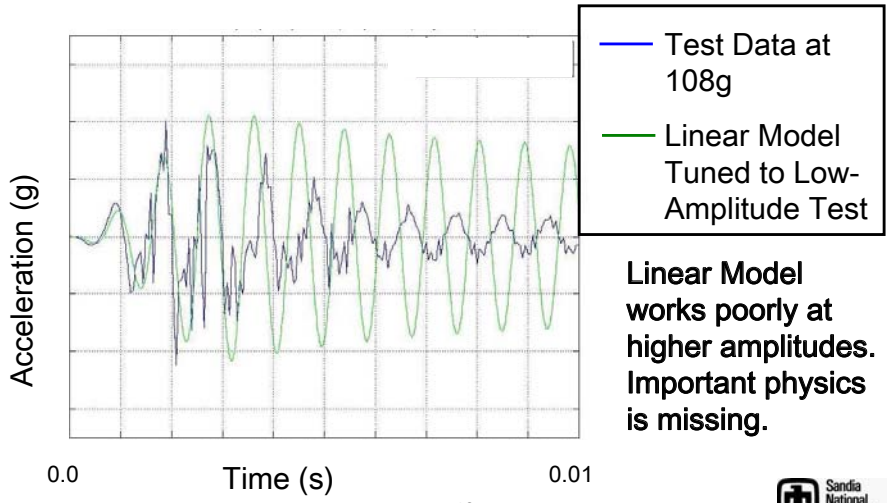
How Well Does that Linear Model Do when Tested on a Different Experiment?



Linear Model works poorly at higher amplitudes. Important physics is missing.

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Not Predictive for Real Systems

If you have to build the full structure in order to predict structural response, then you are not predictive.

The problem is fundamentally nonlinear and important phenomena cannot be captured by tuned linear models. (Silk purse/Sow's ear issue.)

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Why Big Computers Alone are Not Enough

- Multi Length Scales
- Long Duration Events (launch, steady state, …)
- Short Duration Events (blast)
- Very Low to Very High Amplitude Loads

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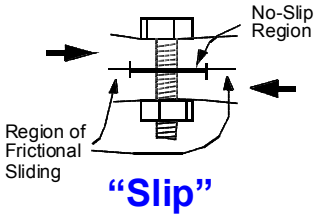
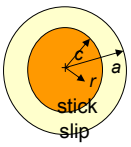
Why Joint Modeling is So Difficult

- **Moving boundaries**
- **Intrinsically multiscale**
- **Nonlocal**

Structure ~ meters

component ~ centimeters





Contact patch ~ cm

Slip zone ~100 μm

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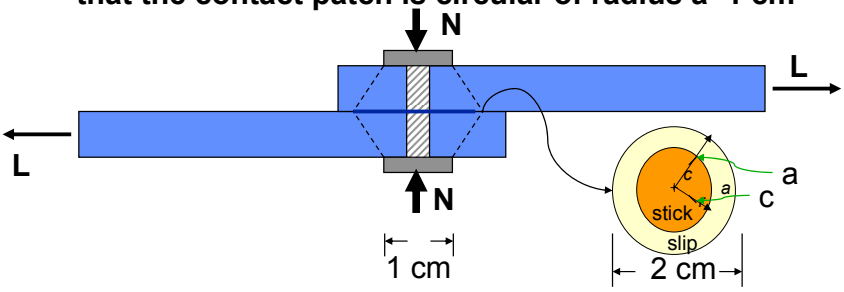
Illustration of Computational Difficulties

- **Consider a lap joint with dimensions selected so that the contact patch is circular of radius a=1 cm**



- **Approximate the elastic contact problem with the Mindlin solution for two spheres.**

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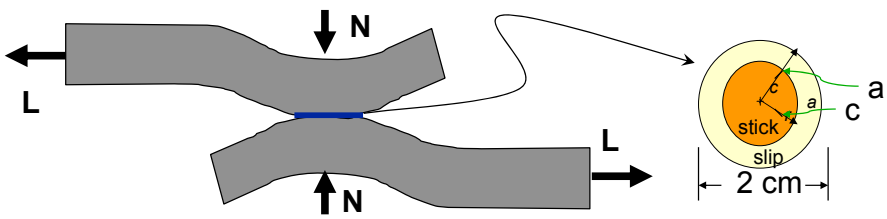
Estimation of Interface Dimensions



- **Normal Load** $N = 4000\,\text{Newtons}$
- **Lateral Loads** $L \in \left(0.05\mu N, 0.8\mu N\right)$
- **Elasticity that of Steel**
- **Slip Zone:**

Say our interest in structural response is in 100Hz-3500Hz

$$\frac{c}{a} = \left[1 - \left(\frac{L}{\mu N}\right)\right]^{1/3} \Rightarrow \frac{c}{a} \in (0.58, 0.98) \Rightarrow \frac{a-c}{a} \in (0.02, 0.42)$$

Necessary Finite Element Scales Courant Times

- **For case of small tangential loads** $L = 0.05\mu N$ **element dimension in slip zone necessary to capture dissipation is** $l = \frac{a-c}{10} = 20\mu m$ **and Courant time is 4 ns**
- **To simulate 10 ms (one cycle of 100 Hz vibration) requires 2.5E6 time steps.**

Compare this with 3E4 time steps if the problem were linear and solved implicitly

Even if This Problem is Solved Quasi-Statically

- In each load cycle, the width of the slip zone twice spans from $a-c=0$ to $a-c=0.42$
- With characteristic element size in the contact patch

$$l=\frac{a-c}{10}=20\mu m$$

- Observing that quasi-static contact has difficulty changing stick-slip status of more than one node at a time and each time step required numerous iterations
- Approximately 800 steps per cycle are required, <u>each representing hundreds of iterations</u>.

Conservation of Cussedness

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Simply Employing More Elements is not the Solution

- One cannot reasonably directly slave a micro-mechanics contact algorithm to a structural dynamics analysis.
- Tools are needed to cross the dimensions

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What We Need

- Better Models
  - Capture at least the qualitative properties of jointed structures.
  - Lend themselves to tractable – even routine – calculation.
  - Cover the full range of environments.
- Better Methods (experimental and computational) to populate the models.
- Better Methods to validate models for joints and jointed structures.

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Conclusions

- This problem appears to be intrinsically difficult. We are not expecting magic bullets.
- There is room for significant improvement on all fronts.

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Backup

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Predictive Modeling – Is that not what we already do?

- In general, engineers use simulation
  - To interpolate/extrapolate among experiments
    - Note the tuned parameters
  - To help explain experiments
  - To help design experiments
  - To provide design guidance
  - To estimate factors of safely
- We generally do not try to predict with precision
  - Finer than the intrinsic variability of the problems
  - That which requires physics for which there are no models

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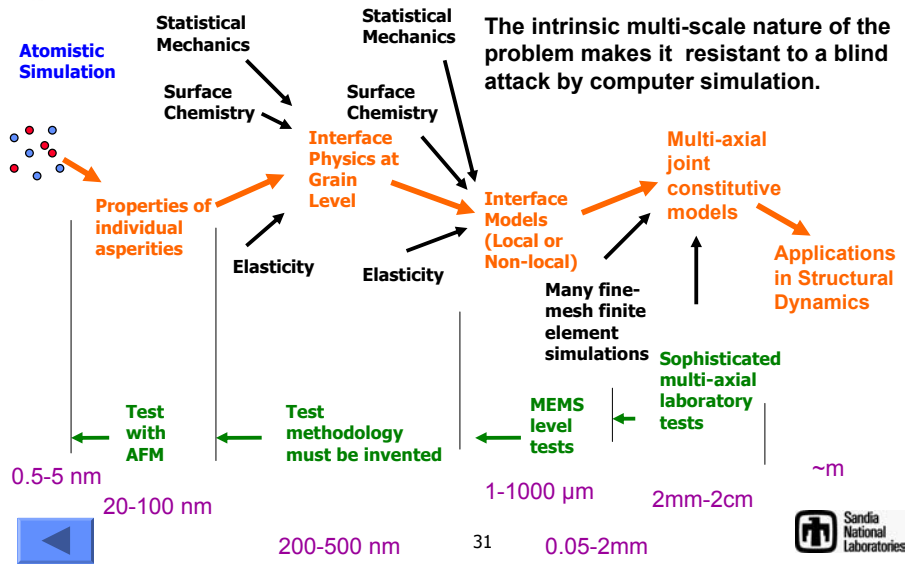
Bottom-Up and Top-Down Vision for Research in Physics of Joint Mechanics

Much of the underlying physics is not understood.

The intrinsic multi-scale nature of the problem makes it resistant to a blind attack by computer simulation.



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Submittal Details

Document Info

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Sandia Contact : SEGALMAN,DANIEL J. Submittal Type : Conference Paper

Requestor : SEGALMAN,DANIEL J. Submit Date : 04/09/2009

Comments : Sandia and AWE are co-sponsoring this workshop and this presentation is to convey some of what we want to have accomplished there.

Peer Reviewed? : N

Author(s)

Ind, Philip SEGALMAN,DANIEL J.

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Partnership Info

Partnership Involved : No

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Technical Advance : No TA Form Filed : No

SD Number :

Classification and Sensitivity Info

Title : Unclassified-Unlimited Abstract : Document : Unclassified-Unlimited

Additional Limited Release Info : None.

DUSA : None.

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Theme A: What does the engineering community need now/soon in terms of joint modelling, and what will it do when it has it?

Structural Assemblies

Larry Bergman
University of Illinois at Urbana-Champaign

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D.1.6 Slide Presentation of Larry Bergman, University of Illinois at Urbana-Champaign: *Theme A: What Does the Engineering Community Need Now/Soon in Terms of Joint Modelling, and What Will it do When it Has it?: Structural Assemblies*

Needs of the engineering community in terms of joint modelling for structural assemblies:

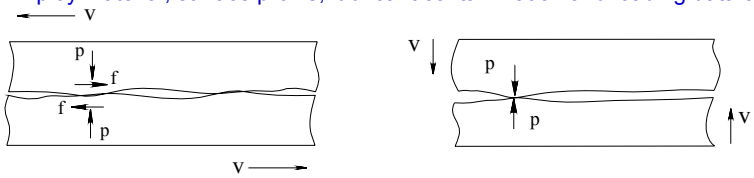
1. **Understand the fundamental physics associated with jointed interfaces**

   Interfacial interactions: friction, impact

   Strongly nonlinear phenomena: unmodeled dynamics vs. uncertainty

2. **Develop analytical and numerical models suitable for structural dynamic studies that can reproduce the physics**

   Employ material, surface profile, lubricant/contamination and loading data only



Tangential direction *Normal direction*

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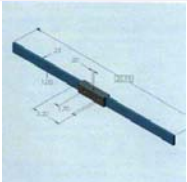
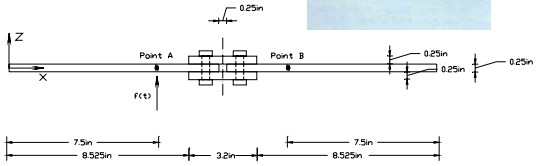
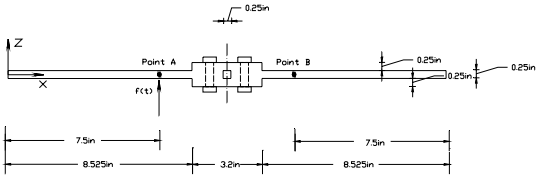
- Compare response of a jointed beam and a monolithic beam
- Free-free boundary conditions
- Transient (hammer) tests



The jointed beam



The monolithic beam



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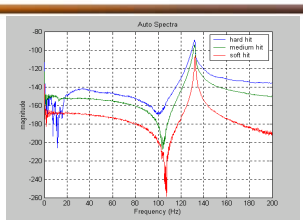
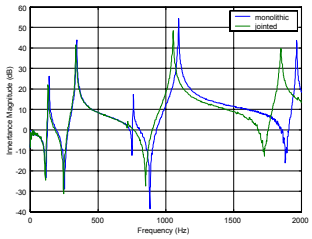


1st mode autospectra of the jointed beam under different level impulses



Inertance FRFs for the two beams (both med. hit)

Natural Frequencies (Hz)		
	Monolithic Beam	Jointed Beam
f_1	140.92	132.26 (-6.1%)
f_2	343.30	336.80 (-1.9%)
f_3	764.67	715.43 (-6.4%)
f_4	1105.80	1052.37 (-4.8%)
f_5	1970.77	1846.62 (-6.3%)
Modal Damping Ratios		
	Monolithic Beam	Jointed Beam
z_1	0.17%	0.69%(+306%)
z_2	0.05%	0.15%(+200%)
z_3	0.03%	0.13%(+333%)
z_4	0.03%	0.11%(+267%)
z_5	0.01%	0.18%(+1700%)

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As clearly seen in the previous slide, structural joints and interfaces can have a significant effect on the dynamics of structural assemblies

Make the structure more compliant
Often the major source of damping

The problem:

In general, the local damping and stiffness associated with a single joint are nonlinear, time-varying, and not repeatable when measured, either directly or indirectly, due to factors such as wear, contamination, etc.

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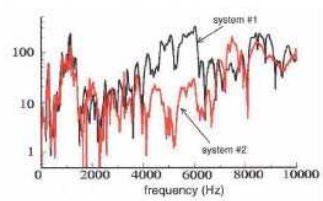

Where we are:

The computed (from measured data) shock response spectra of identical shell structures, each connected to a base by nominally identical joints. The vast difference in spectra illustrates the variability in properties among even nominally identical joints. (Segalman, et al., 2007)

What would we do with a physics-based modelling capability?

Better reconcile the computed response at mid- to high-frequencies with observed data
Determine nominal response profiles within required confidence intervals
More effective evaluation of performance, reliability, design effectiveness

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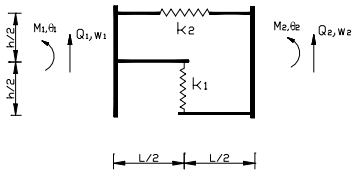
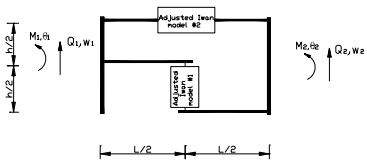
- Linear elastic beam element can be represented by the following arrangement



$$\begin{Bmatrix} Q_1 \\ M_1 \\ Q_2 \\ M_2 \end{Bmatrix} = \frac{EI}{L} \begin{bmatrix} \frac{12}{L^2} & \frac{6}{L} & -\frac{12}{L^2} & \frac{6}{L} \\ \frac{6}{L} & 4 & -\frac{6}{L} & 2 \\ -\frac{12}{L^2} & -\frac{6}{L} & \frac{12}{L^2} & -\frac{6}{L} \\ \frac{6}{L} & 2 & -\frac{6}{L} & 4 \end{bmatrix} \begin{Bmatrix} w_1 \\ \theta_1 \\ w_2 \\ \theta_2 \end{Bmatrix} \qquad k_1 = 12\frac{EI}{L^3} \qquad k_2 = 4\frac{EI}{Lh^2}$$

- The adjusted Iwan beam element is formed by replacing the linear springs with adjusted Iwan models. Iwan parameters are currently fit from whole-joint experimental data. *Can Iwan or equivalent model parameters be determined directly from material, surface and loading data?*



$$\begin{Bmatrix} Q_1 \\ M_1 \\ Q_2 \\ M_2 \end{Bmatrix} = \begin{Bmatrix} f_1(\boldsymbol{\Delta}_1, \hbar_1) \\ \frac{L}{2} f_1(\boldsymbol{\Delta}_1, \hbar_1) + \frac{h}{2} f_2(\boldsymbol{\Delta}_2, \hbar_2) \\ -f_1(\boldsymbol{\Delta}_1, \hbar_1) \\ \frac{L}{2} f_1(\boldsymbol{\Delta}_1, \hbar_1) - \frac{h}{2} f_2(\boldsymbol{\Delta}_2, \hbar_2) \end{Bmatrix}_i$$

$$\boldsymbol{\Delta}_1 = \frac{L}{2}(\theta_1 + \theta_2) + (w_1 - w_2) \qquad \boldsymbol{\Delta}_2 = \frac{h}{2}(\theta_1 - \theta_2)$$

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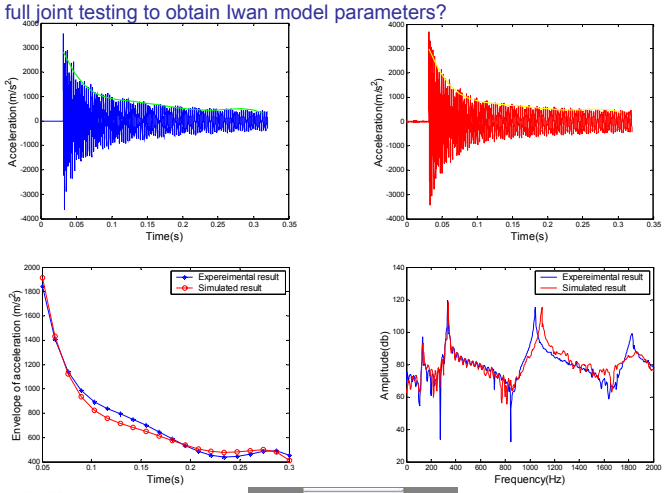
- Comparison of experimental and simulated accelerations using the identified parameters, at point A: Max(NEE)=4.2%; Ave(NEE)=2.1%. Can this level of correspondence be accomplished without full joint testing to obtain Iwan model parameters?



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Points for discussion

For structural assemblies containing joints and interfaces, *structural dynamic models* are generally reduced order models requiring full joint experimental data to define model parameters.

A more thorough understanding of joint physics may permit determination of critical joint model parameters directly from material, surface and load data obtained from micro/mesoscale experiments on standardized specimens.

How to reconcile the question of complex, often unmodeled dynamics with systematic uncertainties?

Is it time to establish a one or more benchmark problems that the community-at-large can participate in and contribute to?

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Imperial College London

Analysis of nonlinear vibrations in jointed gas-turbine structures

E.P. Petrov
Centre of Vibration Engineering
Mechanical Engineering Department

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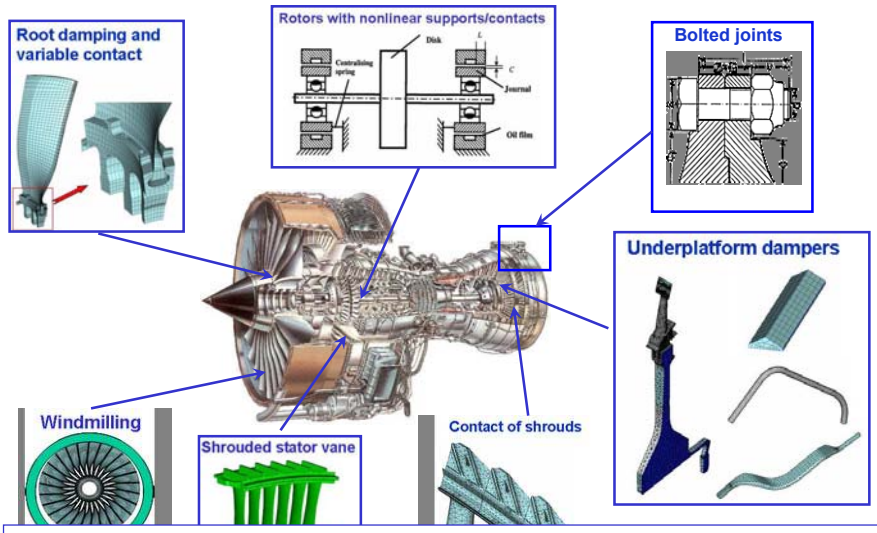
D.1.7 Slide Presentation of Evgeny Petrov, Imperial College: *Analysis of Nonlinear Vibrations in Jointed Gas-Turbine Structures*

Imperial College
London

Contact interfaces in gas-turbine structures



The challenge is to analyse fast and accurately nonlinear dynamics of assembled large-scale models of structures with contact interfaces

Imperial College London

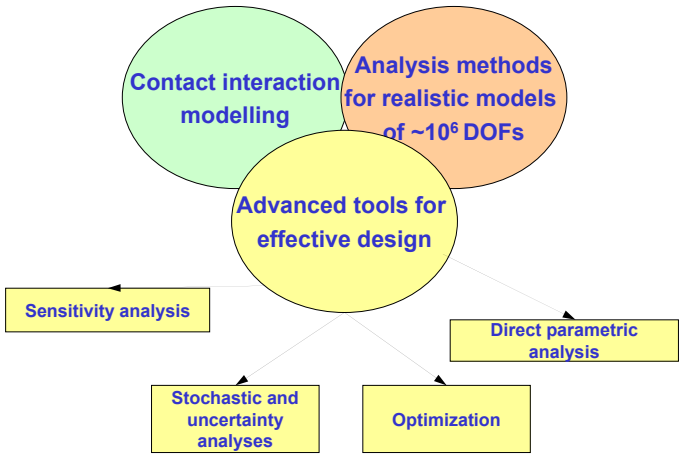
A blade containment test: windmilling

Imperial College London

Methodology for predictive analysis of dynamic problems in gas-turbine structures

Imperial College London

Major components of the methodology developed at Imperial College



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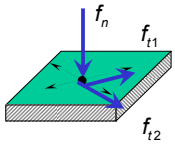
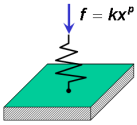
Major types of contact interactions modelled

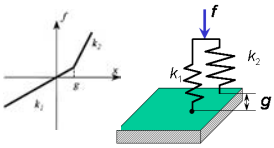
Friction contact: 3D motion with variable normal load



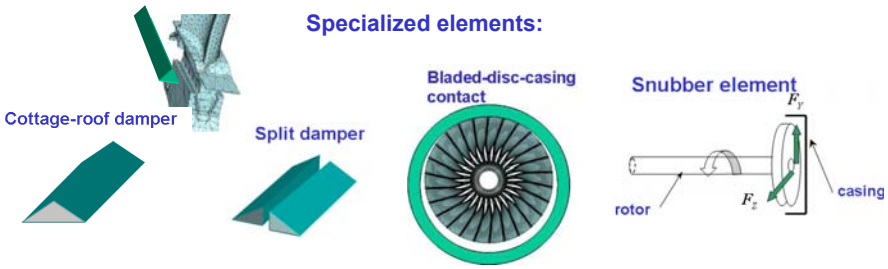
Generalised nonlinear spring element: any polynomial nonlinearity



Bilinear spring and gap



Specialized elements:



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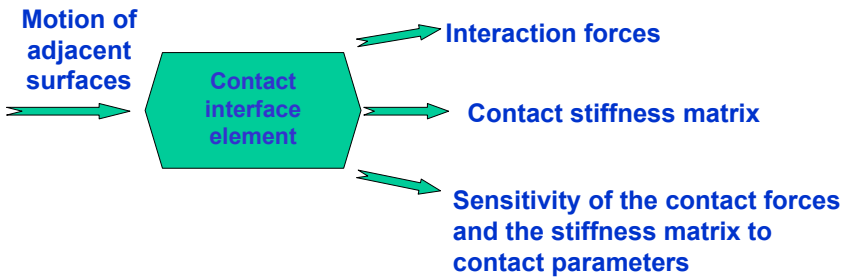
Breakthrough in the analysis of periodic steady-state vibrations: analytically derived contact interface elements



Expressions are obtained in analytical form ⇒ EXACT + extremely FAST calculations

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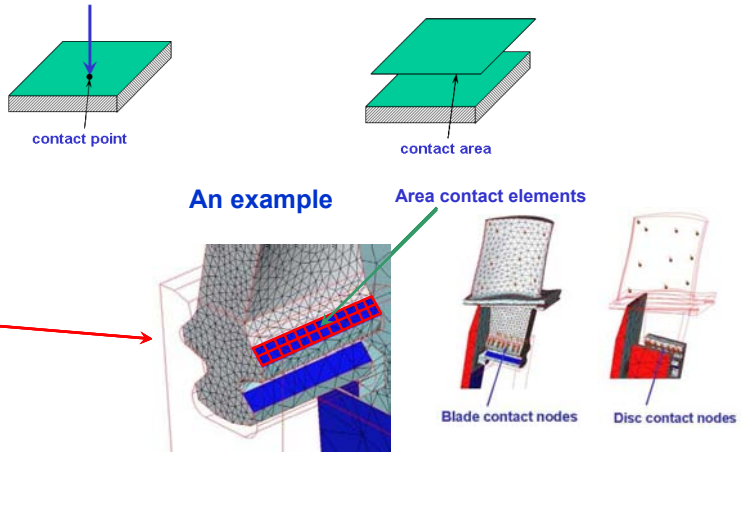
Description of contact interface interactions by the contact elements



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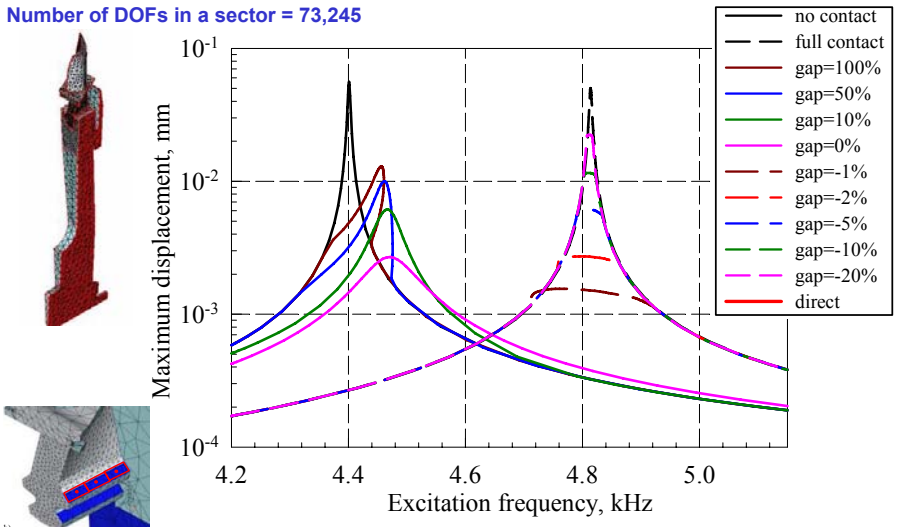
Analysis of forced response: blade root damping

Number of DOFs in a sector = 73,245



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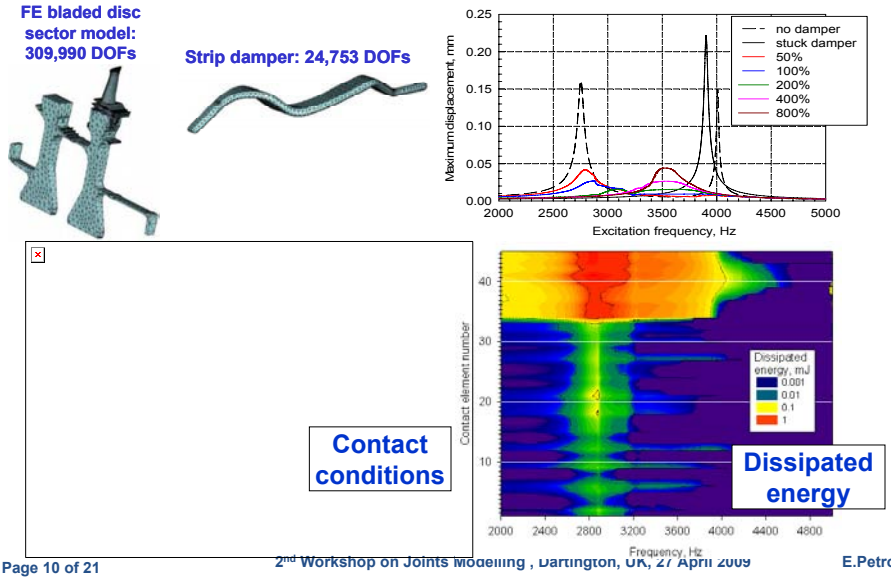
A bladed disc with u/p dampers

FE bladed disc sector model: 309,990 DOFs

Strip damper: 24,753 DOFs



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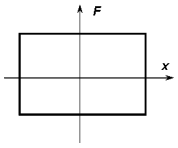
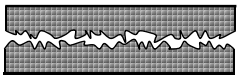
New friction models

Amonton/Coulomb model (1699/1785) $F = \mathrm{sgn}(\dot{x})\mu N$

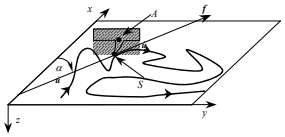


Disadvantages: disregard of normal load variation and history of motion, applicable only to large and 1D displacements, discontinuous

Rough surface contacts



3D motion of a contact point



Capabilities of new models:

1) arbitrary 3D motion, normal load variation including contact-separation
2) accounting for stiffness due to contact surface roughness
3) anisotropy and inhomogeneity of the friction parameters over contact area
4) time variation of friction parameters (due to variation in temperature, wear, lubrication, etc.)

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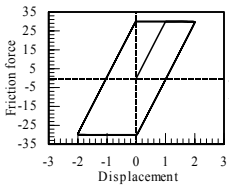
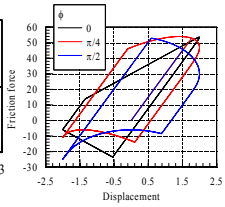
Examples of friction force modelling by new models

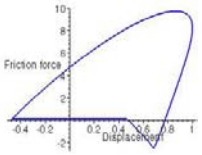
Constant normal load



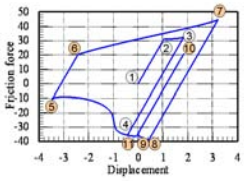
Variable normal load



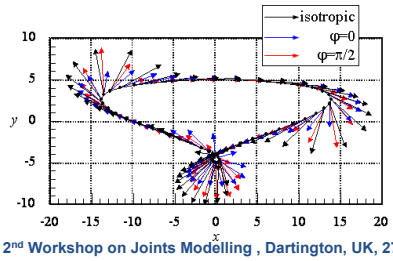
Variable normal load with separation



Multiharmonic vibration

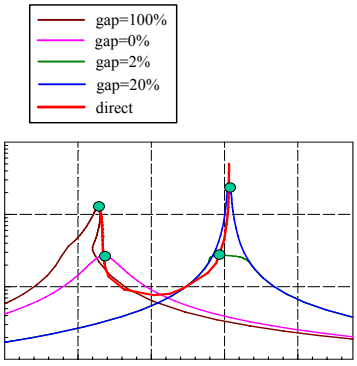


Trajectory and friction force vector (different anisotropy properties)



Advanced tools for effective design

Direct parametric analysis



Conventional forced response analysis:

Frequency response is calculated to obtain only resonance peak frequency and response level.

Many analyses are needed for different parameter values

Direct parametric analysis:

The resonance peak frequency and response level are calculated directly as functions of design parameters

$$\begin{cases} \omega^{res}(\lambda) = \omega^{res}\left(\boldsymbol{b}(\lambda)\right) \\ \boldsymbol{a}^{res}(\lambda) = \boldsymbol{a}^{res}(\boldsymbol{b}(\lambda)) \end{cases}$$

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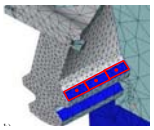
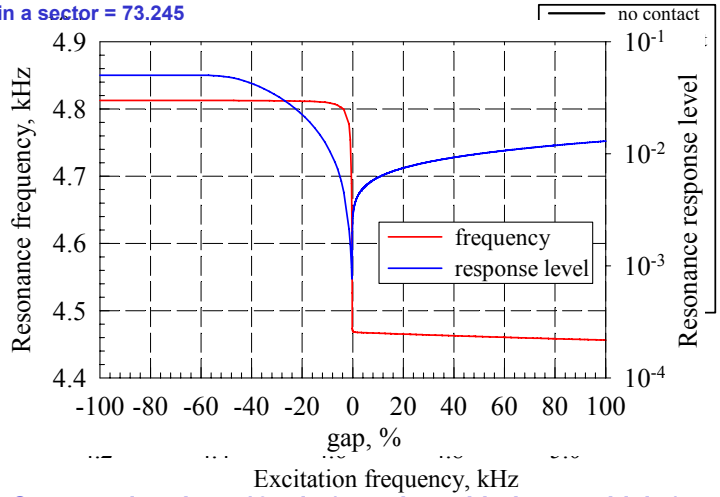
Direct parametric analysis: blade root damping

Number of DOFs in a sector = 73.245





Computation time: 10 min (together with the sensitivity)

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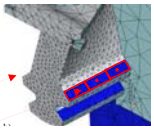
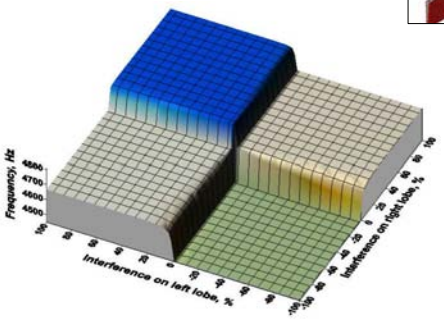
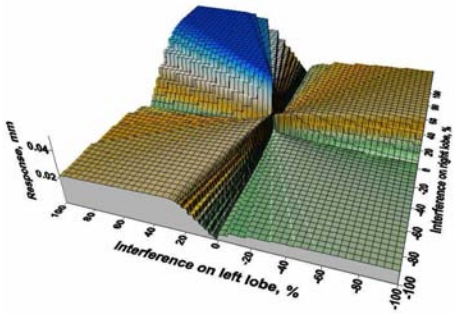
Resonance peak amplitude and frequency: dependency on the interference values at the contact interfaces







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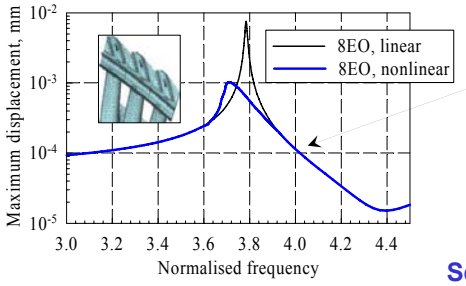
Sensitivity analysis for forced response



Forced response:

$$x(\omega)$$

First-order sensitivity coefficients

$$\frac{\partial x(\omega)}{\partial \lambda}$$

Second-order sensitivity coefficients

$$\frac{\partial^2 x(\omega)}{\partial \lambda^2}$$

λ = clearance, interference, friction coefficient, contact stiffness, mass of u/p damper, or other design parameters and their combinations

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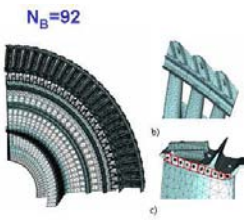
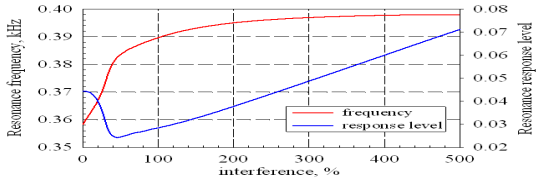
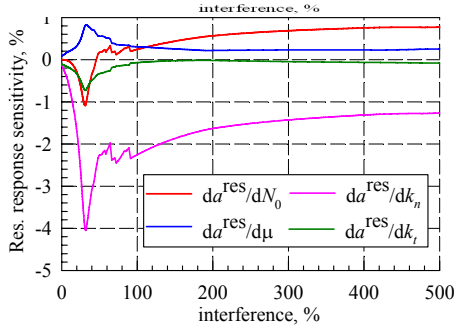
Example: forced response sensitivity to contact interface parameters for a shrouded bladed disc







The analytical derivation provides outstanding qualities of the method: very fast speed of calculations and high accuracy

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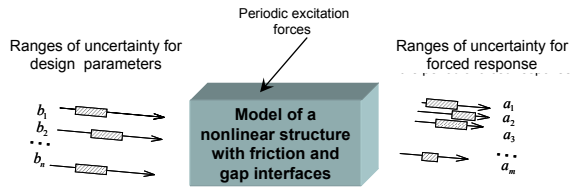
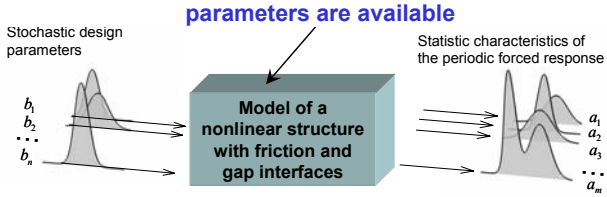
Analysis of effects of design parameters uncertainty and variability on forced response

Uncertainty analysis of forced response for given design parameter uncertainties



Stochastic analysis for forced response: when stochastic characteristics of parameters are available



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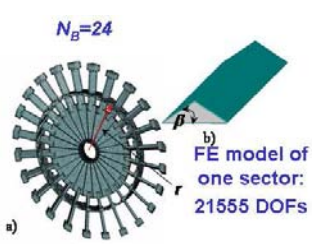
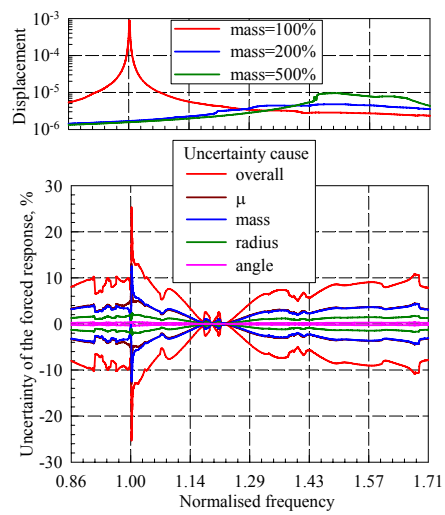
Example: analysis for ranges of uncertainty of the forced response





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Challenges

1) **Validated constitutive equations describing forces at friction contacts (friction laws):**

- for contacts in micro-slip, with small relative motion (bolted joints, flanges, blade-disc joints)
- for large high-energy rubbing motions and impacts (e.g. windmilling)
- for new materials (e.g. composites, rubber, polymers)

2) **Friction contact parameters (e.g. friction and contact stiffness coefficients):**

- prediction
- allowing for dependence and effects of operating conditions: temperature, contact stresses, wear, oxidation, etc.

MODEL UNCERTAINTY

Marc P. Mignolet
Department of Mechanical and Aerospace Engineering
Arizona State University
Tempe, AZ 85287-6106

Ira A. Fulton School of Engineering
Department of Mechanical and Aerospace Engineering
Structural Dynamics Group

D.1.8 Slide Presentation of Marc Mignolet, Arizona State University: *Model Uncertainty*

PLAN

* Uncertainty: parameter uncertainty - model uncertainty

* Benefits of Reduced Order Models

* Entropy Maximization

* Model Identification within an Uncertain Framework

PARAMETER UNCERTAINTY - MODEL UNCERTAINTY

= two sources of uncertainty preventing a perfect match of clean experimental measurements (no measurements uncertainty) with computational predictions:

Uncertainty on the parameters of the computational Model:

* variations (from part to part) of the values entered in the computational model, e.g. Young's modulus, Poisson's ratio, density, dimensions, etc.

* Test article and computational model are different parts from the same stock

PARAMETER UNCERTAINTY - MODEL UNCERTAINTY

Uncertainty (lack of "realism") of the computational model:

* computational geometry *approximates* certain features of the physical model, e.g.
  _ fasteners (rivets, welds, bolts, lap joints,…),
  _ plate/beam models of slender components, …
  _ no warping, out of straight,…

* boundary conditions typically differ from those in test/structure

* constitutive behavior is *modeled*: use of linear structural damping isotropic or orthotropic properties, linear elasticity,…

PARAMETER UNCERTAINTY - MODEL UNCERTAINTY

Notes:

* Separation between model and parameter uncertainty may be gray
  _ thickness can be varied as a parameter in plate/beam models but not in 3D blocks
  _ real boundary conditions can be approximated by linear springs (whose stiffness can become parameter uncertainty)
  _ meshless methods may help in making geometry uncertainty become parameter uncertainty.

* Model uncertainty cannot be eliminated, it may be reduced by great increases in parameter uncertainty.

REPRESENTATION/HANDLING OF UNCERTAINTY

"Throw randomness into the computations"

Not that easy if one desires to be physical/representative:

*Data Uncertainty (easiest): represent each of the uncertainty data as a random variable (if fixed within the element, random process or field otherwise).

The complete representation of a set of *n* random variables requires the specification of the *joint* probability density function, which is a function of *n* variables and includes variation levels (standard deviations) and "correlation/dependence" type information. Is such a complete data available? (*No*) Does it matter?

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REPRESENTATION/HANDLING OF UNCERTAINTY

*Model Uncertainty: one could run a *few* different models. Is that sufficient? (*most likely not*). Maybe one can introduce randomness somewhere as in data uncertainty…?

*Model Updating of Uncertain Structures:

_ Accuracy of "mean (computational) model" is not primary focus, rather it is accuracy of the predicted band of uncertainty around the mean model predictions.

_ Mean model updating need to be carried out to capture the physics that affects the band of uncertainty

BENEFITS OF REDUCED ORDER MODELS

*Reduced Order Models: *when successful/appropriate*

_ reduce the complexity of the uncertainty modeling problem, i.e. number of uncertain parameters

_ transform *model* uncertainty into *parameter* uncertainty

_ should be carried out with *fixed basis* appropriately determined

_ reduce computational cost of carrying out Monte Carlo simulations to assess uncertainty effects

_ eliminate *topological* model constraints (e.g. banded stiffness matrix)

_ reduce/eliminate obvious correlation between various uncertain parameters of the model

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ENTROPY MAXIMIZATION

What do we do if we don't have (as is usual) the complete model of all random variables describing data uncertainty in the ROM?

One answer: (postulating the specific model is another)
Derive the necessary model from an engineering vision of /desire for the uncertainty.

One such vision/desire is that the uncertainty is not simply limited to a small neighborhood of the mean model but spreads broadly as allowed given a set of *constraints.*

This approach leads to the
constrained maximization of the statistical entropy
for the determination of the probabilistic model of uncertainty

EXAMPLE

Linear structural dynamic reduced order model involves mass, damping, and stiffness matrices with following properties:

These matrices are: (i) symmetric, and (ii) positive definite.

The first problem of maximization of entropy was thus about simulating random matrices with properties (i) and (ii) and
(iii) mean of random matrices = matrices of "mean model"
(iv) no zero eigenvalue should occur in the random matrices if none exists in the mean model.

Solution of this problem by Christian Soize in 2000.

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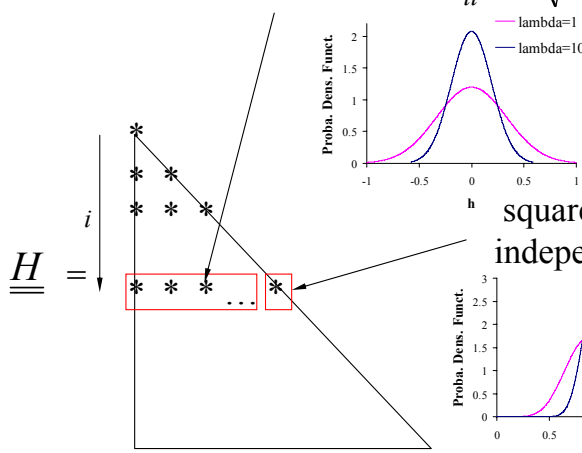
SOLUTION

zero mean Gaussian, independent of each other with standard dev. $\sigma_{il} = 1/\sqrt{2\mu}$

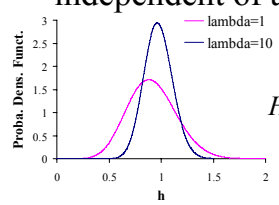
$$\overline{\underline{\underline{A}}} = \overline{\underline{\underline{L}}}\,\overline{\underline{\underline{L}}}^T$$

$$\underline{\underline{A}} = \overline{\underline{\underline{L}}}\,\underline{\underline{G}}\,\overline{\underline{\underline{L}}}^T$$

$$\underline{\underline{G}} = \underline{\underline{H}}\,\underline{\underline{H}}^T$$



square root of Gamma, independent of all others



$$H_{ii} = \sqrt{\frac{Y_{ii}}{\mu}}$$

EXTENSIONS

* problems with rigid body modes
* matrices that are not symmetric, positive definite:
  acoustic-structure interface, bearing stiffness/damping matrices,…
* uncertainty on linear boundary/attachment conditions
* more information on the level of uncertainty (variance of nat. freq.)
* nonlinear geometric ROMs, i.e. linear and nonlinear stiffness terms
* coupled matrices, e.g. mass - gyroscopic matrices in rotordynamics
* …

Entropy Maximization can also serve as basis for simulation of random processes (e.g., friction coefficients), and fields (e.g., random elasticity tensor)

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MODEL IDENTIFICATION IN UNCERTAIN FRAMEWORK

The uncertainty model obtained from entropy maximization has parameters =

(a) parameters of mean model (e.g. natural frequencies, damping ratios)

(b) parameters describing the uncertainty level (Lagrange multipliers associated with the constraints).

These uncertainty model parameters can be obtained using classical estimation approaches (e.g., maximum likelihood) to provide an updating of the mean model from an ensemble of measurements on random parts.

Nonlinear System Identification: Challenges and Open Questions

A.F. Vakakis
University of Illinois
avakakis@illinois.edu

ILLINOIS UNIVERSITY OF ILLINOIS AT URBANA-CHAMPAIGN LNDVL NES Tech 1

D.1.9 Slide Presentation of A.F. Vakakis, University of Illinois at Urbana-Champaign: *Nonlinear System Identification: Challenges and Open Questions*

Nonlinear System Identification: Overview

- The difficulty in developing NSI methodologies that are valid for broad classes of dynamical systems is due to the well-recognized highly individualistic nature of nonlinear systems which restricts the unifying dynamical features that are amenable to system identification
- Some limitations of current methods:
  - Weak nonlinearities
  - Relatively simple systems (i.e., one or two-DOF)
  - Computationally intensive
  - No general methods for systems with non-smooth characteristics (e.g., clearances, friction), strong nonlinearities (e.g., ultra-flexible wings)

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Nonlinear System Identification: Challenges

- One of the challenges of NSI is the well known sensitivity of nonlinear systems to initial and forcing conditions
- So even the simple task of identifying a set of (linearized) modal matrices modified ('perturbed') by nonlinear corrections might be an oversimplification of the problem.
- Effects of non-smooth effects on the dynamics?
- How to address multi-physics and multi-scale effects in NSI?
- How to tie uncertainty modeling with NSI?
- How to identify strongly nonlinear complex nonlinear modal (resonance) interactions?
- Effects of middle- or high-frequency dynamics?

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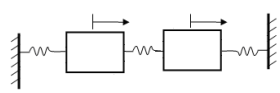
Nonlinear System Identification: Preliminaries

- Simple two-DOF system:

$$\ddot{x}_1 + (2x_1 - x_2) = 0$$
$$\ddot{x}_2 + (2x_2 - x_1) = 0$$



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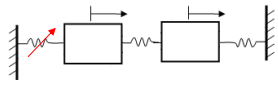
Nonlinear System Identification: Preliminaries

- Now we add a simple (and common) cubic nonlinearity:

$$\ddot{x}_1 + (2x_1 - x_2) + 0.5x_1^3 = 0$$
$$\ddot{x}_2 + (2x_2 - x_1) = 0$$



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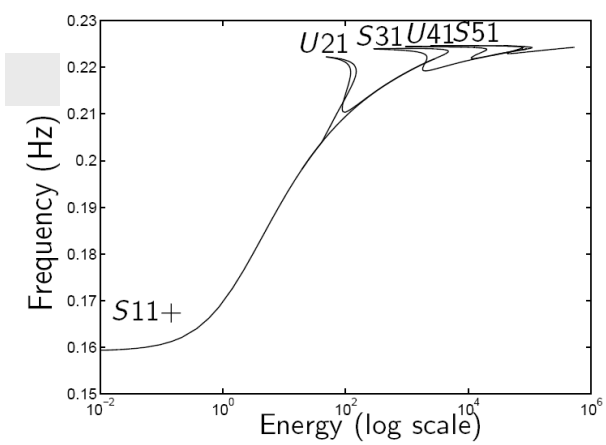
Nonlinear System Identification: Preliminaries

High-frequency strongly nonlinear modal interactions are unavoidable!



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Nonlinear System Identification: Preliminaries

- A very useful feature of the FEP is its relation to the transient dynamics of the corresponding weakly damped system.
- This is due to the fact that the effect on the dynamics of weak damping is parasitic: Instead of introducing 'new' dynamics, it just causes transitions of the dynamics between branches of normal modes leading to multi-frequency nonlinear dynamical transitions.
- Different initial or forcing conditions may lead to drastically different transitions in the FEP. We demonstrate this by a simple two-DOF nonlinear system.

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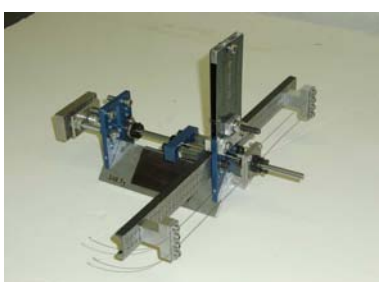
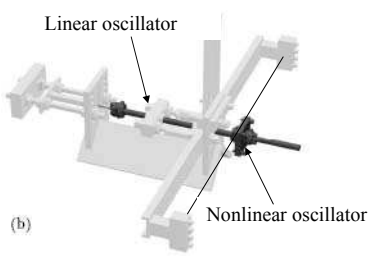
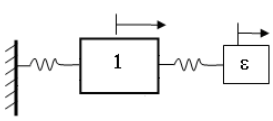
Nonlinear System Identification: Example 1





$$\ddot{x}_1 + \omega_n x_1 + C(x_1 - x_2)^3 = 0$$
$$\varepsilon \ddot{x}_2 + C(x_2 - x_1)^3 = 0$$



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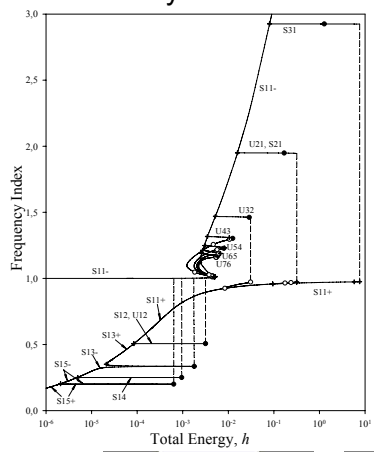
Nonlinear System Identification: Example 1

- Very complicated structure of nonlinear modes in the FEP for this two-DOF system!



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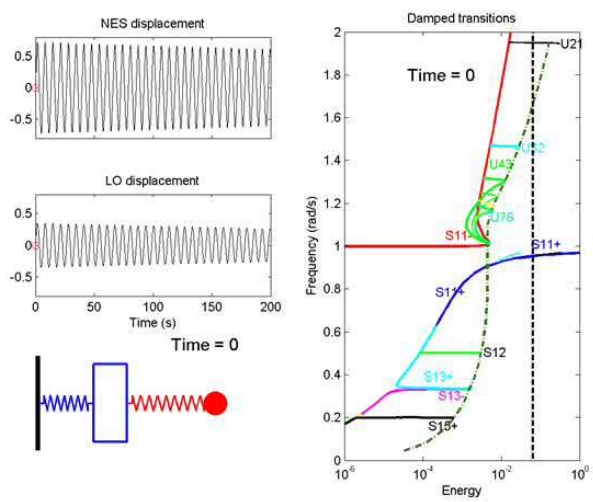
Nonlinear System Identification: Sensitivity on ICs



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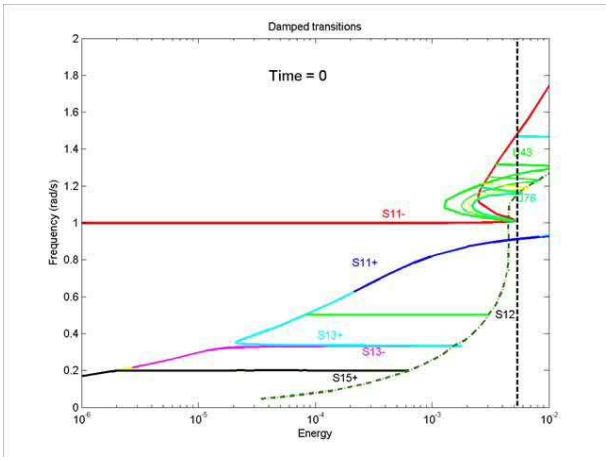
Nonlinear System Identification: Sensitivity on ICs



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Nonlinear System Identification: Challenges

- The previous highlight the important challenges that the analyst is faced with when performing NSI:
  - First, to address the (generic) feature of nonlinear systems to exhibit qualitatively different responses with varying energy and/or initial conditions → need to adopt a global approach for identifying the dynamics over broad frequency and energy ranges.
  - Second, identify complex multi-frequency transitions for particular initial conditions → dictates a local approach, whereby a specific nonlinear transition is considered and the task is to identify the nonlinear modal interactions that govern this transition.

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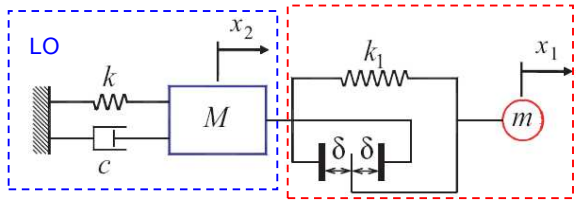
Nonlinear System Identification: Example 2

Example 2: Joint with clearance



$|x_1 - x_2| < \delta \;\Rightarrow$ Vibrations of 2-DOF linear oscillator
$|x_1 - x_2| = \delta \;\Rightarrow$ Vibro-impacts occur

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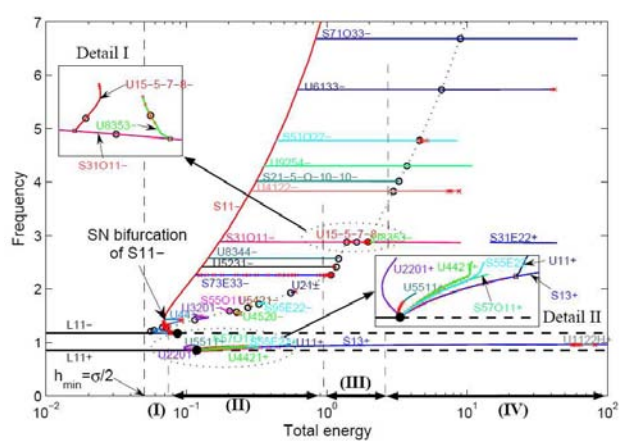
Nonlinear System Identification: Example 2



The FEP shows that the system possesses a very complicated topological structure of dynamics due to the strong vibro-impact nonlinearity.

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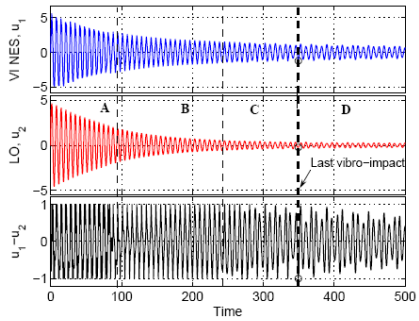
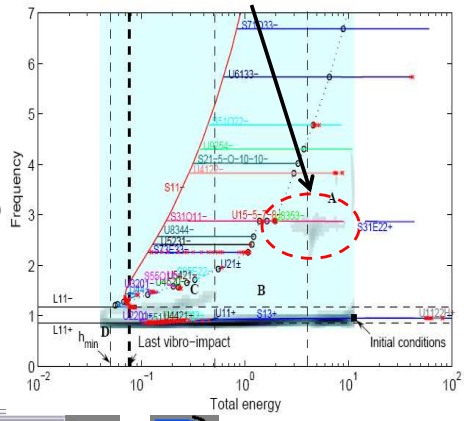
Nonlinear System Identification: Example 2



The initial conditions at 1:1 in-phase NNM are imposed on the system; that is, the resulting VI responses at the initial transient should exhibit 1:1 transient resonance capture.

Does this portion of VI damped transitions on FEP represent superharmonic resonant interactions, instead of 1:1 resonance?



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NSI Methodology: Major Challenges

- Global / local aspects of dynamics need to be addressed
- Need for ROMs that take into account the generic sensitivity of nonlinear systems to changes in initial and forcing conditions.
- Capacity for separating smooth from non-smooth effects, identifying both (especially friction and clearance effects)
- Based on direct analysis of response time series?
- Natural nonlinear extension of linear modal analysis?
- Data driven ROMs in multi-physics problems?
- Probe middle / high frequency dynamics coupling vibrations- and wave-based approaches?
- …

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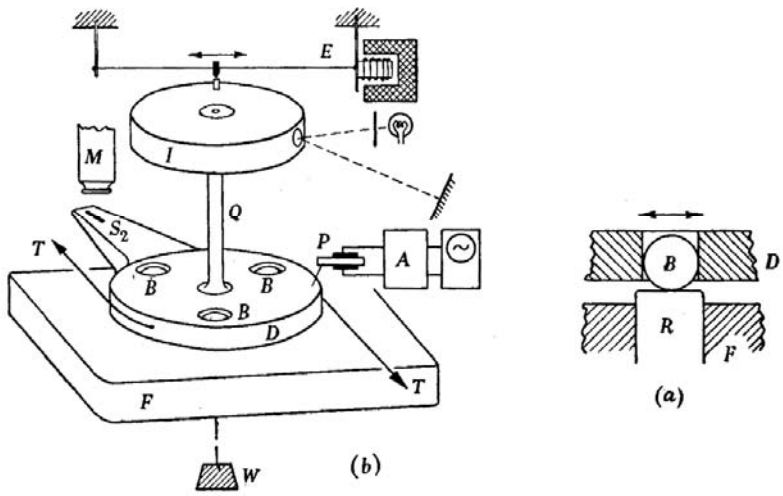


FIGURE 2. (*a*) The contact of a steel ball *B* held in disk *D* with flat end of steel roller *R* held in base plate *F*. (*b*) Diagram of apparatus. Normal load is provided by *W*. Static shear forces are applied by strings *T* and static displacements measured by microscope *M*. Oscillating shear forces are applied by torsional vibration of the disk *I* and the displacements measured by the pick-up *P* and amplifier *A*.

Johnson, K. L. 'Surface interaction between elastically loaded bodies under tangential forces' Proc. Roy. Soc A230 (1955) 531-548

Mindlin, R. D. 'Compliance of elastic bodies in contact' J. Appl. Mech., Trans ASME 71 (1949) 259

Bowden, F. P. and Tabor, D. 'Friction and Lubrication of Solids', Oxford University Press, 1950

D.1.10 Slide Presentation of K.L. Johnson, University of Cambridge

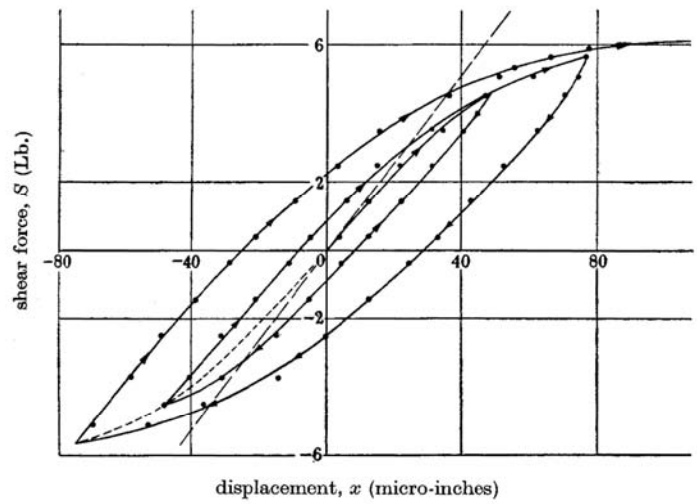


FIGURE 5. Static hysteresis loops. These results show that the major component of the displacement is elastic. On reversing the load a closed symmetrical loop is traced out in which $x(-\bar{S}) \triangleq -x(\bar{S})$. $D = 0{\cdot}375$ in.; $N = 13{\cdot}9$ Lb.

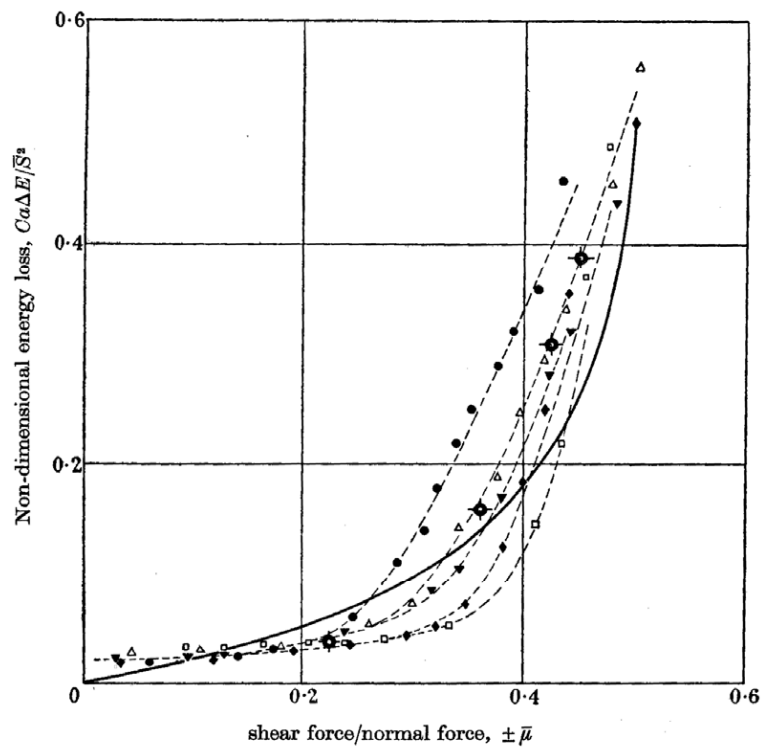


FIGURE 9. Non-dimensional correlation of the results of the damping tests. The full curve represents Mindlin's theoretical relationship given by equation (3) taking $\mu_s = 0{\cdot}5$. The values indicated $\oplus$ were obtained from static hysteresis loops.

Johnson, K. L. ‘Energy disspiation at spherical surfaces in contact transmitting oscillating forces J. Mech. Engng. Sci. 3 (1961) 362-368

Mindlin, R. D. and Deresiewicz, H. ‘Elastic spheres in contact under varying oblique forces’ J. Appl. Mech., Trans ASME 75 (1953) 237

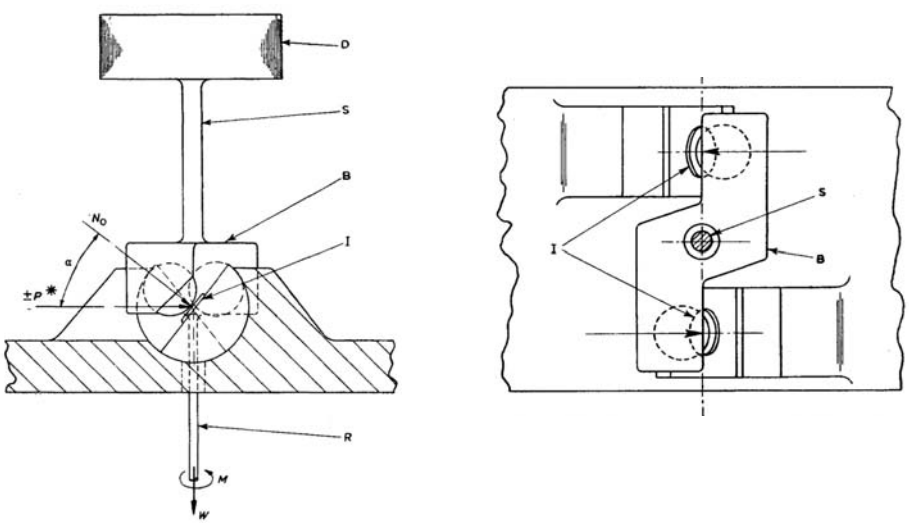


Fig. 3. *Apparatus for measuring the energy dissipation at the contact of a sphere and a plane under the action of an oscillating oblique force*

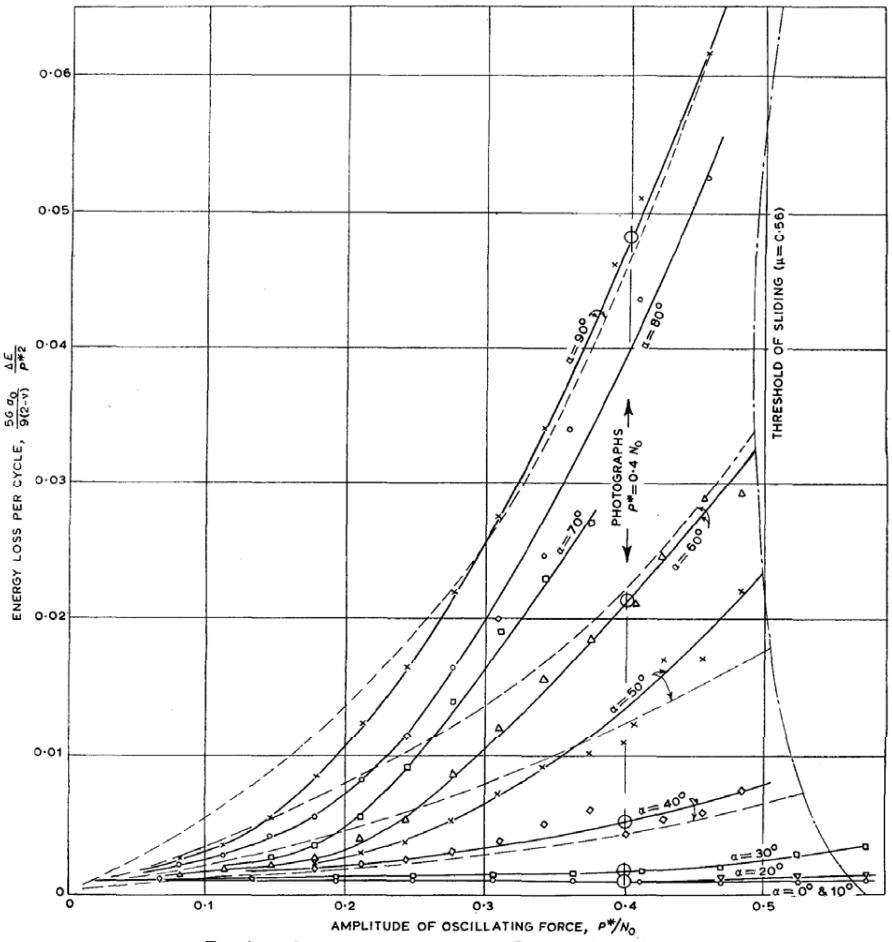
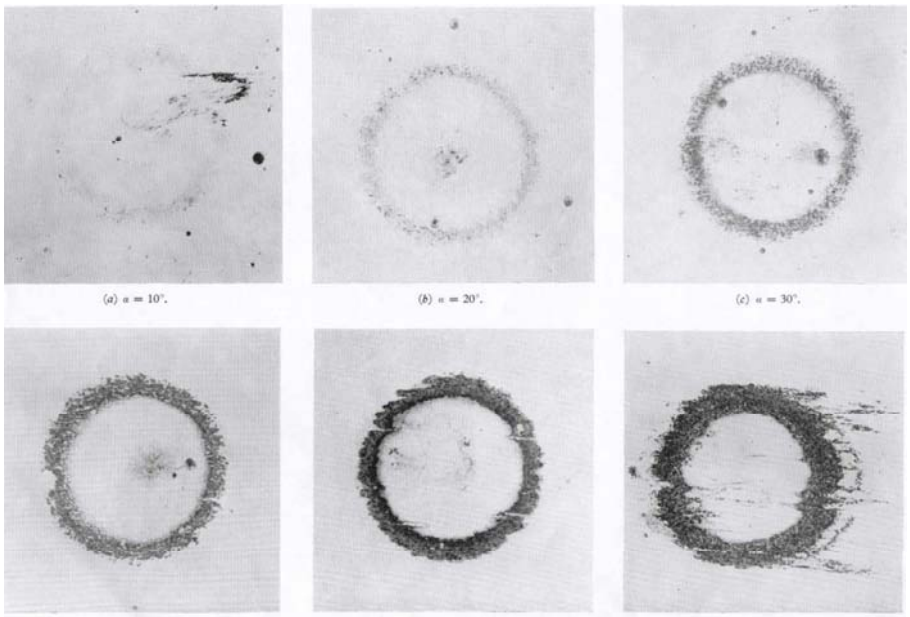


Fig. 4. *Results of energy dissipation measurements at various angles of obliquity*



(a) $\alpha = 10°$. (b) $\alpha = 20°$. (c) $\alpha = 30°$.

(d) $\alpha = 40°$. (e) $\alpha = 60°$. (f) $\alpha = 90°$.

$N_0 = 20$ lb, $a_0 = 0{\cdot}0077$ in.

Fig. 5. *Fretting damage after 10 000 cycles at a constant amplitude* $P^* = 0{\cdot}4N_0$

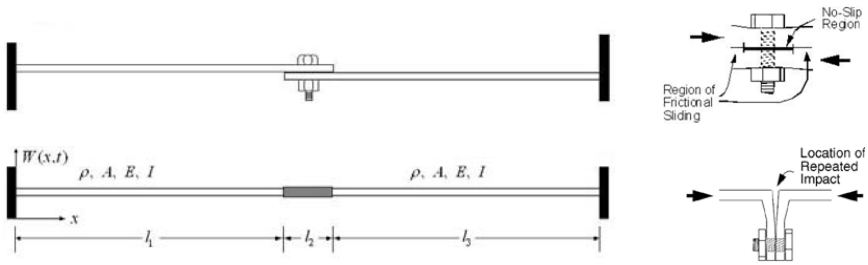
Identification of Nonlinear Bolted Lap-Joint Parameters using Force-State Mapping

International Journal of Solids and Structures, 44 (2007) 8087-8108

Hassan Jalali, Hamed Ahmadian and John E Mottershead

1

D.1.11 Slide Presentation of Hassan Jalali, Hamed Ahmadian, and John E. Mottershead, University of Liverpool, *Identification of Nonlinear Bolted Lap Joint Parameters using Force State Mapping*

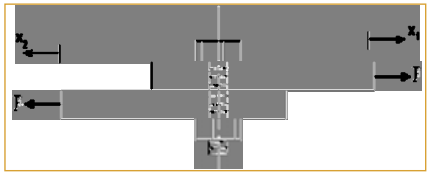


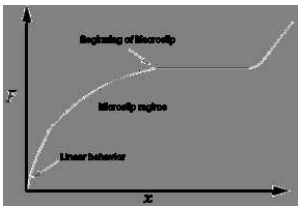
$$\bar{\mathbf{\Gamma}}\,\bar{d} = \bar{F} \qquad \bar{\mathbf{\Gamma}} = \begin{bmatrix} \mathbf{\Gamma} & \mathbf{0} \\ \mathbf{0} & \mathbf{\Gamma}_J \end{bmatrix}$$

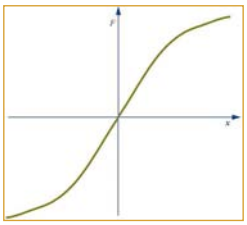
$$\boxed{\Gamma_J = \Gamma_L + \Gamma_{NL}} \Rightarrow \boxed{\Gamma_{NL} = h(x, \dot{x})}$$

- Joint modeling means identification of the joint operator both qualitatively and quantitatively.

2



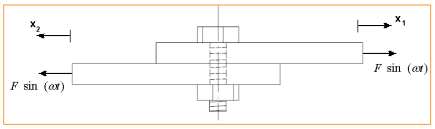


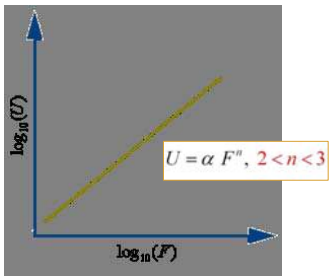


- Static tangential loading leads to a softening stiffness effect:

$$F(x) = k_1 x + k_3 x^3 + k_5 x^5 + \ldots$$

2nd Workshop on Joints Modelling – Dartington – 26-29 April 2009

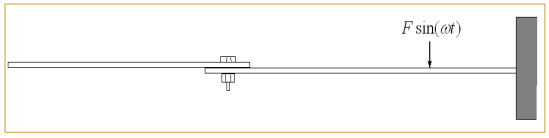


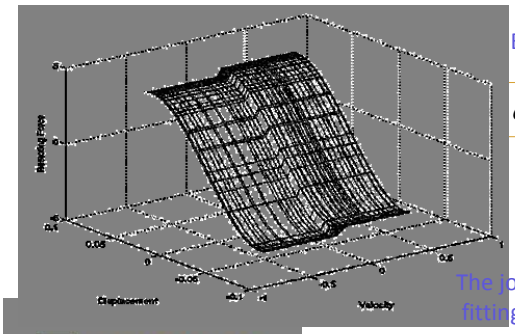


- Harmonic force at the bolted lap joint
- Energy dissipated vs. the force amplitude follows a power-law relationship.
- Experimental results show that $2 \leq n \leq 3$
- Linear viscous damping – dependent upon maximum steady-state displacement amplitude

$$c(x_{\max}) = c_0 + c_1 x_{\max} + c_2 x_{\max}^2 + \cdots$$

Force-State Mapping





Equations in modal coordinates:

$$\ddot{q}(t) + \omega_1^2 q(t) + h(\dot{q}, q) = Q(t)$$

$$\omega_1^2\, q(t) + h(q, \dot{q}) = Q(t) - \ddot{q}(t)$$

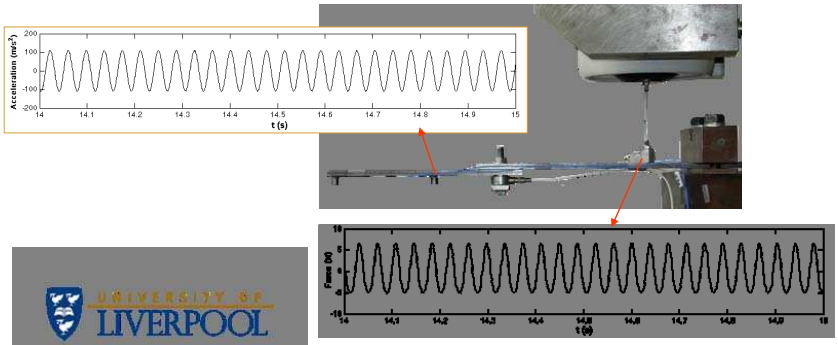
The joint parameters are identified by fitting a function on the surface .

Experiment

- Single-frequency excitation close to a natural frequency
- Measured low-level force and acceleration converted to a single modal coordinate using an updated FE model
- Other (higher) modes are eliminated
- Modal acceleration integrated analytically to obtain velocity and displacement

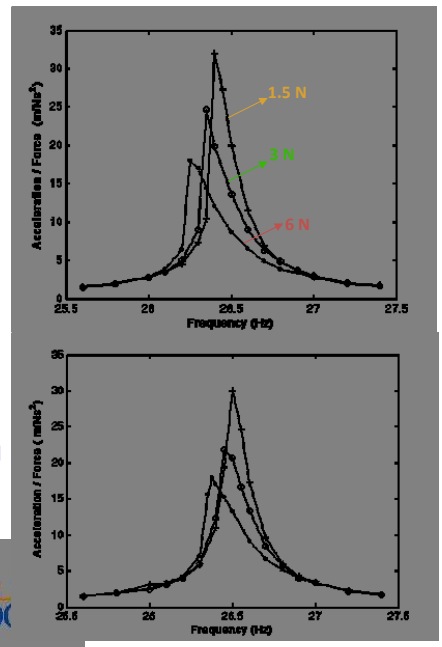
$$\ddot{q}(t) = \sum_{i}^{n} \left(A_i \sin(i\omega t) + B_i \cos(i\omega t) \right)$$

- Truncated after the fourth harmonic



Preload=120N

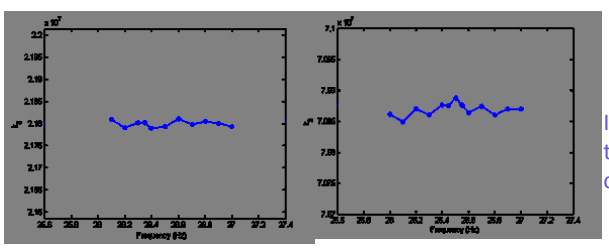
Preload=540N



3 Excitation levels

2 Preload conditions

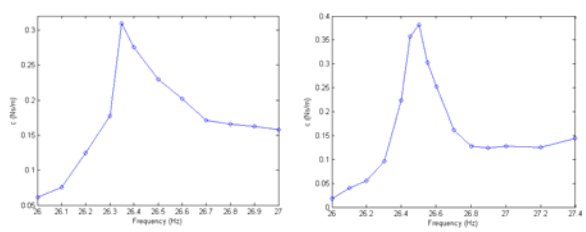
7



Identification shows that k_3 is almost constant

Preload=120N

Preload=540N



Viscous damping c peaks at the natural frequency – it depends upon the amplitude of vibration

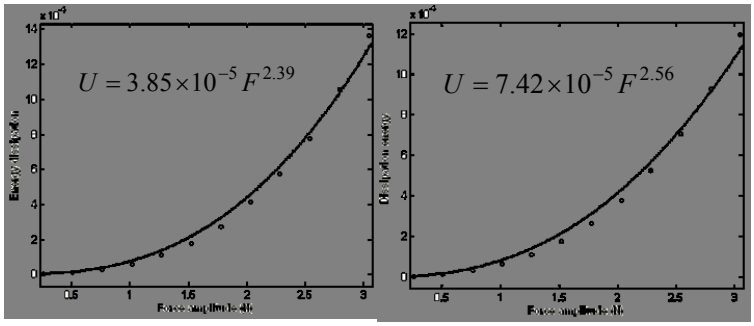
$$Q(t) - \ddot{q}(t) = \omega_1^2 q(t) - k_3 q(t)^3 + c(q_{max})\dot{q}(t)$$

$$c(q_{\max}) = c_0 + c_1 q_{\max} + c_2 q_{\max}^2 + \cdots$$

8

Displacement – dependent damping $c(q_{\max}) = c_0 + c_1 q_{\max} + c_2 q_{\max}^2 + \cdots$

Energy constraint $U = \alpha\, F^n,\ 2 < n < 3$



Preload 120N

Preload 540N

Excitation at 6N

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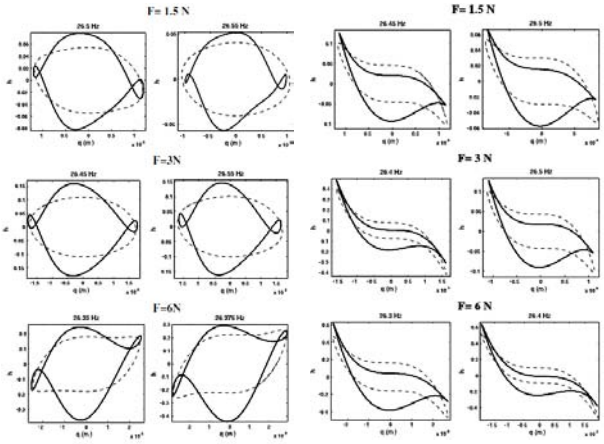

Full line – experimental
Dashed line- analytical

Hysteresis Loops – Preload=120N

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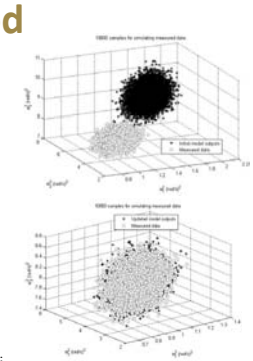
- Area and orientation approximately correct
- Possible underestimation of the cubic stiffening term

Perturbation Methods for the Estimation of Parameter Variability in Stochastic Model Updating

Mechanical Systems and Signal Processing 22 (2008) 1751-1773

Hamed Haddad Khodaparast, John E Mottershead and Michael I Friswell

The Perturbation Method



Classical model updating: $\boldsymbol{\theta}_{j+1} = \boldsymbol{\theta}_j + \mathbf{T}_j(\mathbf{z}_m - \mathbf{z}_j)$

Measurement: $\mathbf{z}_m = \overline{\mathbf{z}}_m + \Delta\mathbf{z}_m$

Prediction: $\mathbf{z}_j = \overline{\mathbf{z}}_j + \Delta\mathbf{z}_j$

Parameters: $\boldsymbol{\theta} = \overline{\boldsymbol{\theta}} + \Delta\boldsymbol{\theta}$

Mean $\overline{\bullet}$
Variability Δ

Transformation matrix: $\mathbf{T} = \overline{\mathbf{T}} + \Delta\mathbf{T}_j$ $\quad \Delta\mathbf{T}_j = \sum_{k=1}^{n} \frac{\partial \mathbf{T}_j}{\partial z_{mk}} \Delta z_{mk}$

Stochastic model updating:

$$\overline{\boldsymbol{\theta}}_{j+1} + \Delta\boldsymbol{\theta}_{j+1} = \overline{\boldsymbol{\theta}}_j + \Delta\boldsymbol{\theta}_j + \left(\overline{\mathbf{T}}_j + \Delta\mathbf{T}_j\right)\left(\overline{\mathbf{z}}_m + \Delta\mathbf{z}_m - \overline{\mathbf{z}}_j - \Delta\mathbf{z}_j\right)$$

The Perturbation Method

$$\mathbf{O}\left(\Delta^0\right): \qquad \overline{\boldsymbol{\theta}}_{j+1} = \overline{\boldsymbol{\theta}}_j + \overline{\mathbf{T}}_j\left(\overline{\mathbf{z}}_m - \overline{\mathbf{z}}_j\right)$$

- the mean values of the updating parameters

$$\mathbf{O}(\Delta^1): \qquad \Delta\boldsymbol{\theta}_{j+1} = \Delta\boldsymbol{\theta}_j + \overline{\mathbf{T}}_j\left(\Delta\mathbf{z}_m - \Delta\mathbf{z}_j\right) + \left(\left(\sum_{k=1}^{n}\frac{\partial\overline{\mathbf{T}}_j}{\partial z_{mk}}\Delta z_{mk}\right)\left(\overline{\mathbf{z}}_m - \overline{\mathbf{z}}_j\right)\right)$$

- leads to an expression for the parameter covariances

Parameter Covariances

$$\mathrm{Cov}\left(\Delta\boldsymbol{\theta}_{j+1},\Delta\boldsymbol{\theta}_{j+1}\right) = \mathrm{Cov}\left(\Delta\boldsymbol{\theta}_j + \mathbf{A}_j\,\Delta\mathbf{z}_m + \overline{\mathbf{T}}_j\left(\Delta\mathbf{z}_m - \Delta\mathbf{z}_j\right),\ \Delta\boldsymbol{\theta}_j + \mathbf{A}_j\,\Delta\mathbf{z}_m + \overline{\mathbf{T}}_j\left(\Delta\mathbf{z}_m - \Delta\mathbf{z}_j\right)\right)$$

$$\begin{aligned}
&= \mathrm{Cov}\left(\Delta\boldsymbol{\theta}_j,\Delta\boldsymbol{\theta}_j\right) + \mathrm{Cov}\left(\Delta\boldsymbol{\theta}_j,\Delta\mathbf{z}_m\right)\mathbf{A}_j^T + \mathrm{Cov}\left(\Delta\boldsymbol{\theta}_j,\Delta\mathbf{z}_m\right)\overline{\mathbf{T}}_j^T - \mathrm{Cov}\left(\Delta\boldsymbol{\theta}_j,\Delta\mathbf{z}_j\right)\overline{\mathbf{T}}_j^T \\
&\quad + \left(\mathrm{Cov}\left(\Delta\boldsymbol{\theta}_j,\Delta\mathbf{z}_m\right)\mathbf{A}_j^T\right)^T + \mathbf{A}_j\mathrm{Cov}\left(\Delta\mathbf{z}_m,\Delta\mathbf{z}_m\right)\mathbf{A}_j^T + \mathbf{A}_j\mathrm{Cov}\left(\Delta\mathbf{z}_m,\Delta\mathbf{z}_m\right)\overline{\mathbf{T}}_j^T - \mathbf{A}_j\mathrm{Cov}\left(\Delta\mathbf{z}_m,\Delta\mathbf{z}_j\right)\overline{\mathbf{T}}_j^T \\
&\quad + \left(\mathrm{Cov}\left(\Delta\boldsymbol{\theta}_j,\Delta\mathbf{z}_m\right)\overline{\mathbf{T}}_j^T\right)^T + \left(\mathbf{A}_j\mathrm{Cov}\left(\Delta\mathbf{z}_m,\Delta\mathbf{z}_m\right)\overline{\mathbf{T}}_j^T\right)^T + \overline{\mathbf{T}}_j\mathrm{Cov}\left(\Delta\mathbf{z}_m,\Delta\mathbf{z}_m\right)\overline{\mathbf{T}}_j^T - \overline{\mathbf{T}}_j\mathrm{Cov}\left(\Delta\mathbf{z}_m,\Delta\mathbf{z}_j\right)\overline{\mathbf{T}}_j^T \\
&\quad - \left(\mathrm{Cov}\left(\Delta\boldsymbol{\theta}_j,\Delta\mathbf{z}_j\right)\overline{\mathbf{T}}_j^T\right)^T - \left(\mathbf{A}_j\mathrm{Cov}\left(\Delta\mathbf{z}_m,\Delta\mathbf{z}_j\right)\overline{\mathbf{T}}_j^T\right)^T - \left(\overline{\mathbf{T}}_j\mathrm{Cov}\left(\Delta\mathbf{z}_m,\Delta\mathbf{z}_j\right)\overline{\mathbf{T}}_j^T\right)^T + \overline{\mathbf{T}}_j\mathrm{Cov}\left(\Delta\mathbf{z}_j,\Delta\mathbf{z}_j\right)\overline{\mathbf{T}}_j^T
\end{aligned}$$

$$\mathbf{A} = \left[\frac{\partial\overline{\mathbf{T}}_j}{\partial z_{m1}}\left(\overline{\mathbf{z}}_m - \overline{\mathbf{z}}_j\right) \quad \frac{\partial\overline{\mathbf{T}}_j}{\partial z_{m2}}\left(\overline{\mathbf{z}}_m - \overline{\mathbf{z}}_j\right) \cdots \frac{\partial\overline{\mathbf{T}}_j}{\partial z_{mn}}\left(\overline{\mathbf{z}}_m - \overline{\mathbf{z}}_j\right)\right] \qquad \overline{\mathbf{T}}_j = \left(\overline{\mathbf{S}}_j^T\mathbf{W}_1\overline{\mathbf{S}}_j + \mathbf{W}_2\right)^{-1}\overline{\mathbf{S}}_j^T\mathbf{W}_1$$

$\mathrm{Cov}(\Delta\boldsymbol{\theta}_j,\Delta\mathbf{z}_j)$ and $\mathrm{Cov}(\Delta\mathbf{z}_j,\Delta\mathbf{z}_j)$ are determined by forward propagation

$$\mathrm{Cov}\left(\Delta\boldsymbol{\theta}_{j+1},\Delta\mathbf{z}_{\mathrm{m}}\right)=\mathrm{Cov}\left(\Delta\boldsymbol{\theta}_{j}+\mathbf{A}_{j}\Delta\mathbf{z}_{m}+\overline{\mathbf{T}}_{\mathrm{j}}\left(\Delta\mathbf{z}_{m}-\Delta\mathbf{z}_{j}\right),\Delta\mathbf{z}_{m}\right)$$
$$=\mathrm{Cov}\left(\Delta\boldsymbol{\theta}_{j},\Delta\mathbf{z}_{m}\right)+\left(\mathbf{A}_{j}+\overline{\mathbf{T}}_{j}\right)\mathrm{Cov}\left(\Delta\mathbf{z}_{m},\Delta\mathbf{z}_{m}\right)-\overline{\mathbf{T}}_{j}\mathrm{Cov}\left(\Delta\mathbf{z}_{j},\Delta\mathbf{z}_{m}\right)$$
$$\mathrm{Cov}\left(\boldsymbol{\Delta}\mathbf{z}_{j},\boldsymbol{\Delta}\mathbf{z}_{m}\right)=\overline{\mathbf{S}}_{\mathrm{j}}\mathrm{Cov}\left(\boldsymbol{\Delta}\boldsymbol{\theta}_{j},\boldsymbol{\Delta}\mathbf{z}_{m}\right)$$

$$\mathbf{A}_{j+1}=\left[\left.\frac{\partial\overline{\mathbf{T}}_{j+1}}{\partial z_{m1}}\right|_{z_{m1}=\bar{z}_{m1}}\left(\overline{\mathbf{z}}_{m}-\overline{\mathbf{z}}_{j+1}\right)\quad\left.\frac{\partial\overline{\mathbf{T}}_{j+1}}{\partial z_{m2}}\right|_{z_{m2}=\bar{z}_{m2}}\left(\overline{\mathbf{z}}_{m}-\overline{\mathbf{z}}_{j+1}\right)\quad\cdots\quad\left.\frac{\partial\overline{\mathbf{T}}_{j+1}}{\partial z_{mn}}\right|_{z_{mn}=\bar{z}_{mn}}\left(\overline{\mathbf{z}}_{m}-\overline{\mathbf{z}}_{j+1}\right)\right]$$

$$\left.\frac{\partial\overline{\mathbf{T}}_{j+1}}{\partial z_{mk}}\right|_{z_{mk}=\bar{z}_{mk}}=\sum_{i=1}^{m}\frac{\partial\overline{\mathbf{T}}_{j+1}}{\partial\overline{\theta}_{(j+1),i}}\left.\frac{\partial\overline{\theta}_{(j+1),i}}{\partial z_{mk}}\right|_{z_{mk}=\bar{z}_{mk}}\quad;\quad k=1,2,\ldots,n$$

$$\frac{\partial\overline{\mathbf{T}}_{j+1}}{\partial\theta_{(j+1),i}}=\left(\overline{\mathbf{S}}_{j+1}^{T}\mathbf{W}_{1}\overline{\mathbf{S}}_{\mathrm{j+1}}+\mathbf{W}_{2}\right)^{-1}\frac{\partial\overline{\mathbf{S}}_{j+1}^{T}}{\partial\theta_{(j+1),i}}\mathbf{W}_{1}$$
$$-\left(\overline{\mathbf{S}}_{j+1}^{T}\mathbf{W}_{1}\overline{\mathbf{S}}_{j+1}+\mathbf{W}_{2}\right)^{-1}\left(\frac{\partial\overline{\mathbf{S}}_{j+1}^{T}}{\partial\theta_{(j+1),i}}\mathbf{W}_{1}\overline{\mathbf{S}}_{j+1}+\overline{\mathbf{S}}_{j+1}^{T}\mathbf{W}_{1}\frac{\partial\overline{\mathbf{S}}_{j+1}}{\partial\overline{\theta}_{(j+1),i}}\right)\left(\overline{\mathbf{S}}_{j+1}^{T}\mathbf{W}_{1}\overline{\mathbf{S}}_{j+1}+\mathbf{W}_{2}\right)^{-1}\overline{\mathbf{S}}_{j+1}^{T}\mathbf{W}_{1}$$

2nd order sensitivity needed

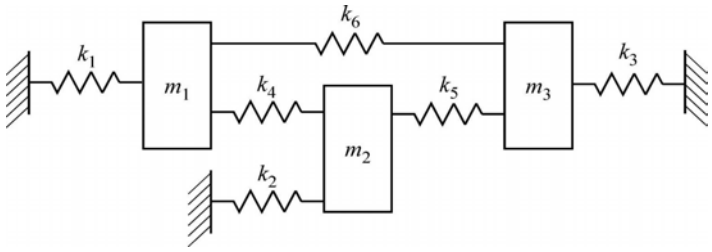
Simplification

If the measurements and parameters are assumed to be uncorrelated then, $\mathrm{Cov}(\Delta\mathbf{z}_m,\Delta\boldsymbol{\theta}_j)=0$, $\mathrm{Cov}(\Delta\mathbf{z}_m, \Delta\mathbf{z}_j)=0$

The parameter covariances become;

$$\mathrm{Cov}\left(\boldsymbol{\Delta}\boldsymbol{\theta}_{j+1},\boldsymbol{\Delta}\boldsymbol{\theta}_{j+1}\right)=\mathrm{Cov}\left(\boldsymbol{\Delta}\boldsymbol{\theta}_{j},\boldsymbol{\Delta}\boldsymbol{\theta}_{j}\right)-\mathrm{Cov}\left(\boldsymbol{\Delta}\boldsymbol{\theta}_{j},\boldsymbol{\Delta}\mathbf{z}_{j}\right)\overline{\mathbf{T}}_{j}^{T}+\overline{\mathbf{T}}_{j}\mathrm{Cov}\left(\boldsymbol{\Delta}\mathbf{z}_{m},\boldsymbol{\Delta}\mathbf{z}_{m}\right)\overline{\mathbf{T}}_{j}^{T}$$
$$-\overline{\mathbf{T}}_{j}\mathrm{Cov}\left(\boldsymbol{\Delta}\mathbf{z}_{j},\boldsymbol{\Delta}\boldsymbol{\theta}_{j}\right)+\overline{\mathbf{T}}_{j}\mathrm{Cov}\left(\boldsymbol{\Delta}\mathbf{z}_{j},\boldsymbol{\Delta}\mathbf{z}_{j}\right)\overline{\mathbf{T}}_{j}^{T}$$

No requirement for the second-order sensitivities.



Known deterministic parameters

$m_i = 1.0\,\text{kg} \quad (\text{i} = 1,2,3) \qquad k_i = 1.0\,\text{N/m} \quad (i = 3,4) \qquad k_6 = 3.0 \ \text{N/m}$

Unknown Gaussian random variables with mean values and standard deviations given by

$\mu_{k_1} = 1.0\,\text{N/m} \quad , \mu_{k_2} = 1.0\,\text{N/m}\,,\ \mu_{k_5} = 1.0\,\text{N/m}$

$\sigma_{k1} = 0.20\text{N/m}, \quad \sigma_{k2} = 0.20\text{N/m}, \quad \sigma_{k5} = 0.20\text{N/m}$

The measured data are obtained by using Monte Carlo simulation.
The initial estimates of the unknown random parameters are,

$\bar{k}_1 = \bar{k}_2 = \bar{k}_5 = 2.0\,\text{N/m} \quad \text{Cov}(k_i) = (0.3^2)\text{N}^2/\text{m}^2 \ \ i = 1,2,5$

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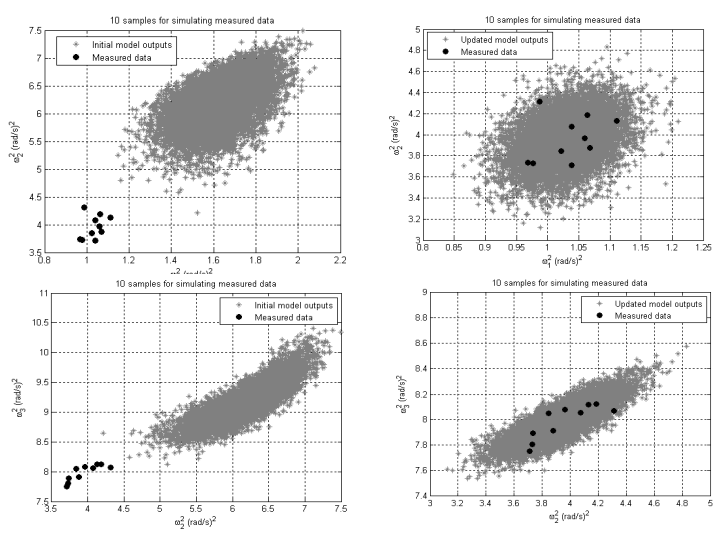

Convergence of natural frequency distributions

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Converged Distributions – 10000 Samples

Parameters	Initial % Error	% Error (1)	% Error (2)	% Error (3)	% Error (4)	% Error (5)
$\bar{k}_1$	100	1.29	1.43	1.22	1.88	16.40
$\bar{k}_2$	100	-2.45	-2.70	-2.68	-2.62	36.56
$\bar{k}_5$	100	0.56	0.61	0.61	2.13	58.58
$std\ (\bar{k}_1)$	50	0.14	0.45	1.50	-89.88	-14.82
$std\ (\bar{k}_2)$	50	-0.68	1.96	0.75	-89.96	-13.07
$std\ (\bar{k}_5)$	50	1.22	0.62	0.34	-90.07	-59.87

Methods:
1.Simplified perturbation method
2.Full perturbation method
3.Perturbation method by Hua
4.Minimum variance – Collins et al.
5.Minimum variance - Friswell

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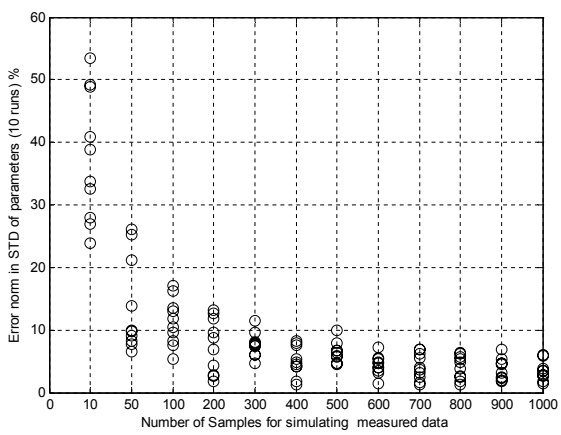
Effect of sample size



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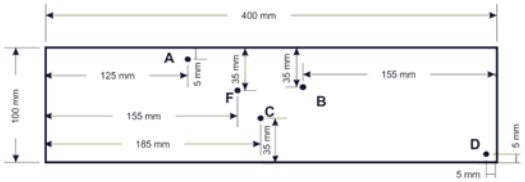
Experimental Case Study – Plate Thickness Variability

Arrangement of accelerometers (A, B, C, D) and driving point (F)



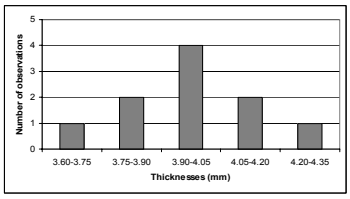


Plate thickness distribution

Measurement Distribution from 10 plates

	Mode Number				
Plate No.	1	2	3	4	5
1	119.774	284.283	331.970	589.404	656.359
2	121.615	291.922	337.186	605.160	665.854
3	123.156	291.440	340.184	602.603	673.357
4	128.048	298.163	355.210	620.139	700.798
5	128.533	303.809	357.110	630.809	704.505
6	128.596	301.010	361.488	635.533	713.207
7	129.796	311.726	361.114	646.765	712.792
8	135.058	315.393	374.368	653.584	738.395
9	134.478	312.215	374.406	649.130	737.256
10	138.141	321.812	382.932	667.203	755.189
Mean	128.720	303.177	357.597	630.033	705.771
STD	6.011	12.032	17.048	25.235	32.854

	Mode Number				
Plate No.	6	7	8	9	10
1	932.576	1091.603	1343.097	1628.879	1825.215
2	953.666	1106.861	1372.890	1650.395	1860.225
3	955.515	1119.445	1376.298	1669.899	1868.071
4	980.403	1165.177	1414.181	1736.714	1924.260
5	995.188	1169.660	1433.020	1743.750	1946.155
6	999.248	1184.455	1440.134	1765.415	1957.581
7	1019.052	1184.608	1467.366	1766.361	1987.556
8	1031.837	1225.375	1487.512	1825.602	2021.640
9	1023.229	1224.420	1479.268	1824.121	2013.354
10	1053.974	1253.610	1519.011	1866.665	2031.377
Mean	994.469	1172.521	1433.278	1747.780	1943.543
STD	38.877	53.840	56.771	79.232	72.908

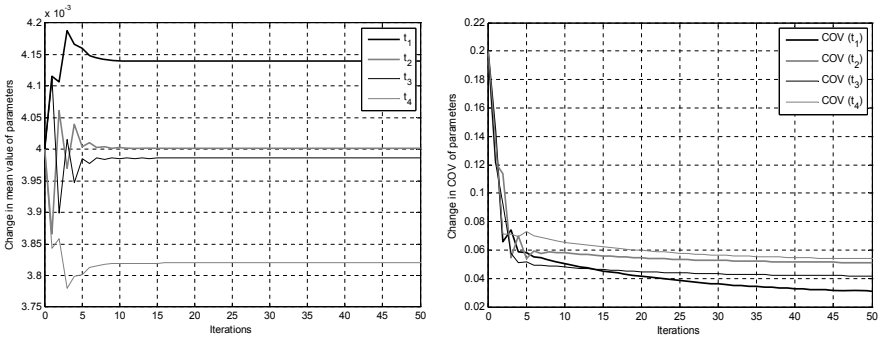
Parameterisation into Four Regions

1	2	3	4

Initial parameters

$$\bar{t}_i = 4\,\text{mm}, \quad \text{std}(t_i) = 0.8\,\text{mm}, \quad i = 1,\ldots,4.$$

Convergence of Parameter Estimates



Measured, Initial and Updated Mean and Standard Deviation of Parameters

mm	Measured Parameters	Initial Parameters	Updated Parameters	Initial FE % error	Updated FE % error
$\bar{t}_1$	3.978	4.000	4.140	0.553	4.072
$\mathrm{std}(t_1)$	0.159	0.8	0.129	403.145	-18.868
$\bar{t}_2$	3.969	4.000	4.002	0.781	0.831
$\mathrm{std}(t_2)$	0.161	0.8	0.204	396.894	26.708
$\bar{t}_3$	3.982	4.000	3.986	0.452	0.100
$\mathrm{std}(t_3)$	0.164	0.8	0.166	387.805	1.219
$\bar{t}_4$	3.981	4.000	3.820	0.477	-4.044
$\mathrm{std}(t_4)$	0.167	0.8	0.206	379.042	23.353

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Measured, initial and updated mean natural frequencies

	Measured (Hz)	Initial FE (Hz)	Updated FE (Hz)	Initial FE % error	Updated FE % error
Mode (1)	128.720	128.321	128.111	-0.310	-0.473
Mode (2)	303.177	307.147	306.339	1.310	1.043
Mode (3)	357.597	356.645	355.185	-0.266	-0.675
Mode (4)	630.033	637.433	633.188	1.175	0.501
Mode (5)	705.771	705.467	701.777	-0.043	-0.566
Mode (6)	994.469	1002.229	996.865	0.780	0.241

Measured, initial and updated std of natural frequencies

	Measured (Hz)	Initial FE (Hz)	Updated FE (Hz)	Initial FE % error	Updated FE % error
Mode (1)	6.011	20.943	5.750	248.411	-4.342
Mode (2)	12.032	47.385	13.777	293.825	14.503
Mode (3)	17.048	39.231	15.180	130.121	-10.957
Mode (4)	25.235	65.655	26.797	160.175	6.190
Mode (5)	32.854	71.379	28.644	117.261	-12.814
Mode (6)	38.877	108.445	40.166	178.944	3.316

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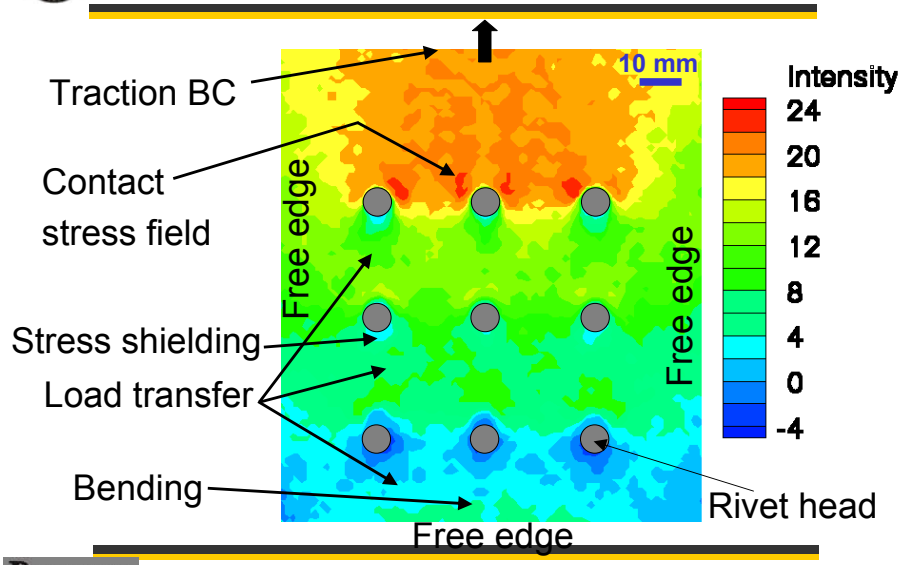
Recent Developments for Conformal Contacts with Friction

N Sundaram
TN Farris
Purdue University

Center for Materials Processing and Tribology

D.1.12 Slide Presentation of N. Sundaram and Tom Farris, Purdue University, *Recent Developments for Conformal Contacts with Friction*

Applications (σ_o =110 MPa)



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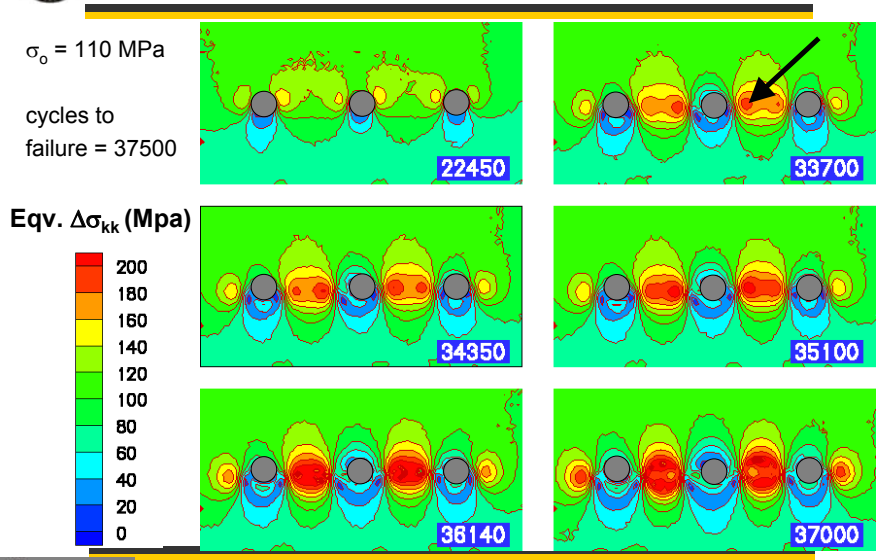
Crack Evolution

σ_o = 110 MPa

cycles to failure = 37500



Rigid-on-iso, Full Sliding

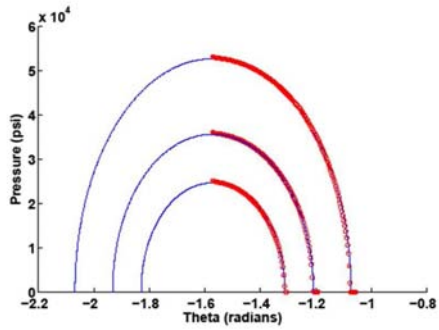
- Full sliding SIDE obtained from the coupled system derived earlier Solution scheme: Transform Hilbert kernel to Cauchy kernel, solve with Gauss-Chebyshev quadrature and Newton-Raphson search.
- Full sliding solution is interpreted as being caused by applied CW / CCW moments under the prevailing P,Q, remote stress conditions
- The moments inducing sliding are obtained as part of the solution

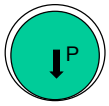
$$k'' = \kappa - 1 \qquad k' = \frac{\kappa+1}{2} \qquad \mathscr{A} = \sigma_{xx}^{\infty} + \sigma_{yy}^{\infty} \qquad \mathscr{D} = \sigma_{xx}^{\infty} - \sigma_{yy}^{\infty}$$

$$\boxed{\begin{aligned} -R + R_D - \frac{Rk'}{2G}\left[\frac{A}{2} - 3D\cos(2\theta)\right] = \frac{R}{4\pi G}\Bigg[& k'\int_{\alpha}^{\beta} N(\xi)d\xi + 2\kappa\int_{\alpha}^{\beta}\cos(\theta-\xi)N(\xi)d\xi \\ & -k'\int_{\alpha}^{\beta}\cot\left(\frac{\theta-\xi}{2}\right)N'(\xi)d\xi - \pi k''N(\theta) \pm \mu k'\int_{\alpha}^{\beta}\cot\left(\frac{\theta-\xi}{2}\right)N(\xi)d\xi \\ & \mp\mu 2\kappa\int_{\alpha}^{\beta}\sin(\theta-\xi)N(\xi)d\xi \mp \pi k''\mu N'(\theta) \quad \forall\theta\in(\alpha,\beta)\Bigg] \end{aligned}}$$

PIN LOAD ONLY

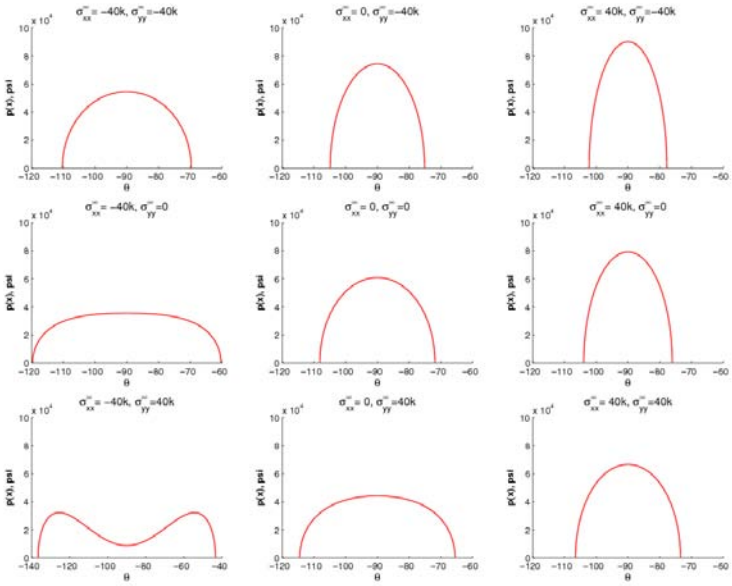
- Conformal contacts μ = 0
- R = 1.0” Rd = 0.99”
- E = 1.708 x 10^7 ν = 0.316
- Plane strain, κ = 1.735
- P = 10k,20k, 40k
- Solution of Singular Integro-Differential Equation using modified Gauss-Chebyshev method
- Solid line (SIDE), Marker (FEM)





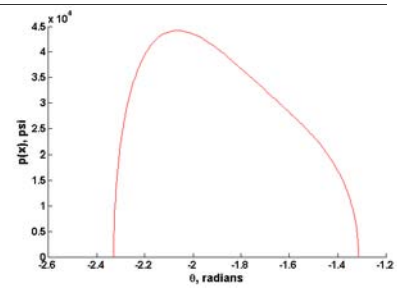
REMOTE STRESSES + PIN LOAD

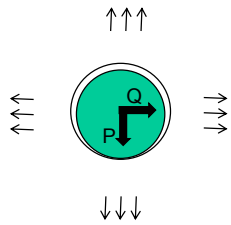
R = 1.0”
Rd =0.98”
κ = 1.735
P = 30k
(all cases)
σ_{xx} =
-40,0,40 ksi
σ_{yy} =
-40,0,40 ksi

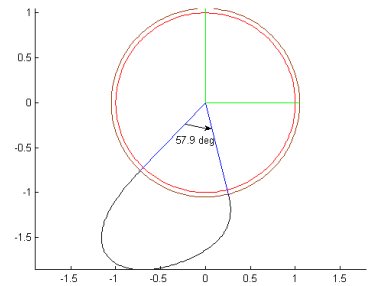


Oblique pin load+remote stresses

- Equivalent to P only + additional σ_{xy} at infinity when $\mu = 0$
- Essential to include Q in formulation because in presence of friction this is not possible without loss of generality
- $\kappa = 1.735$ R=1", Rd=0.98", P=30k, Q=-10k σ_{xx} = -10ksi, σ_{yy} = 50 ksi

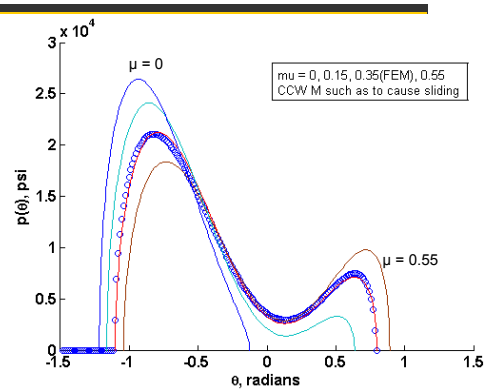


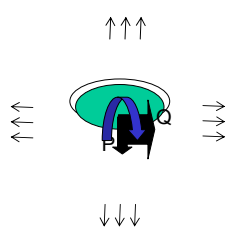




Generalized case of bulk-sliding

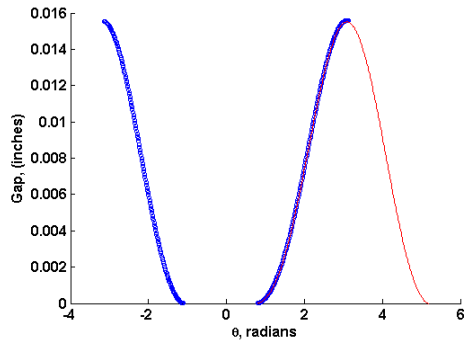
- μ = 0.0, 0.15, 0.35, 0.55
- R = 1.0" Rd = 0.995"
- E = 17.08 x 10^6 ν = 0.316
- Plane strain, $\kappa = 1.735$
- P = 12.5k, Q=12.5k
- σ_{xx} = -12 ksi, σ_{yy} = 12 ksi
- Solid lines (SIDE), Marker (FEM, very near sliding)



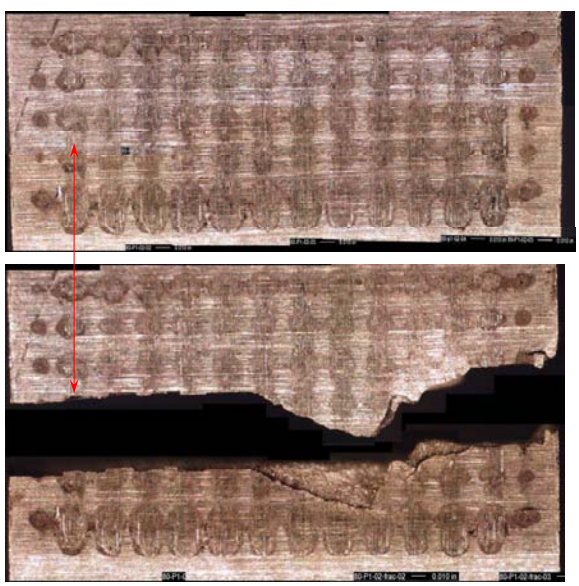


Results: Gap and approach for a full sliding problem

- Problem same as discussed before: has P,Q, stresses
- mu = 0.35
- Solid line SIDE + 2pt. bvp gap
- Markers FEM
- Points separated by 2*pi are identical
- SIDE predicted sliding moment M_{cr} = -6608, FEM ~ -6635
- **Previous results due to Persson and Ciavarella and Decuzzi**



LSP Contact Surface



- Top picture is prior to testing
- Bottom picture indicates fracture along middle of contact
- Arrow indicates origin of fatigue crack

Frictional contacts

David A. Hills

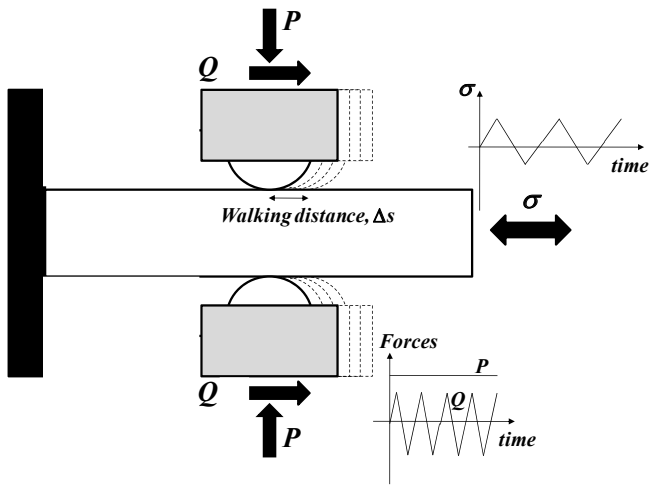
Department of Engineering Science
University of Oxford
28/4/2009

Rolls-Royce

D.2 Presentations of April 28, 2009 Current State of the Art in Joint Modelling

D.2.1 Slide Presentation of David Hills, University of Oxford: *Frictional contacts*

Friction Measurements without Sliding



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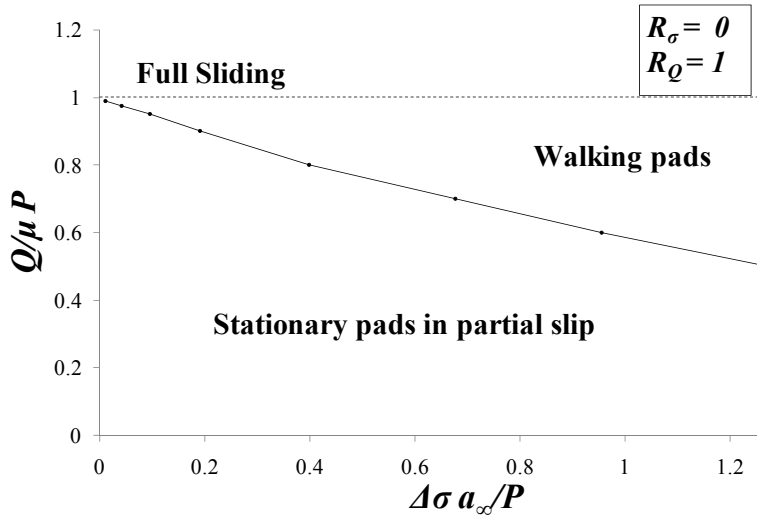
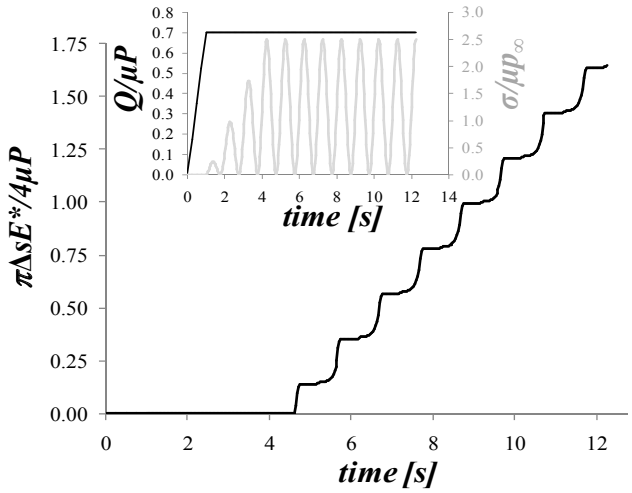
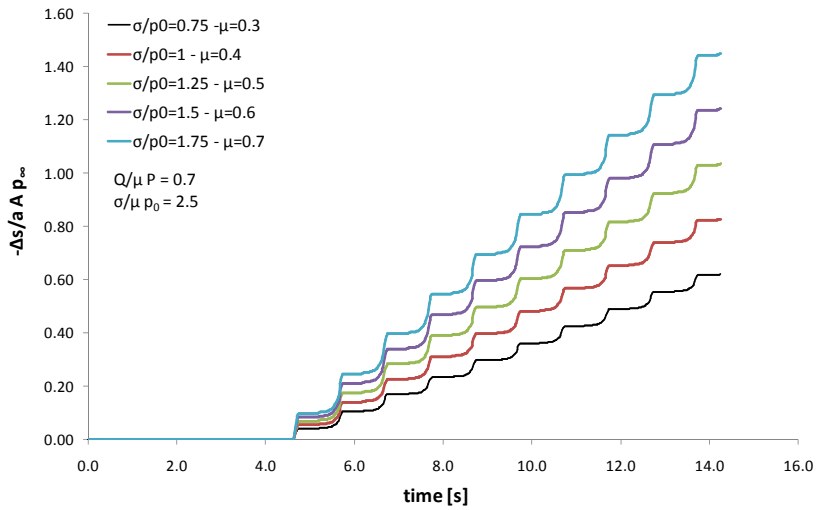


Rolls-Royce



Rolls-Royce



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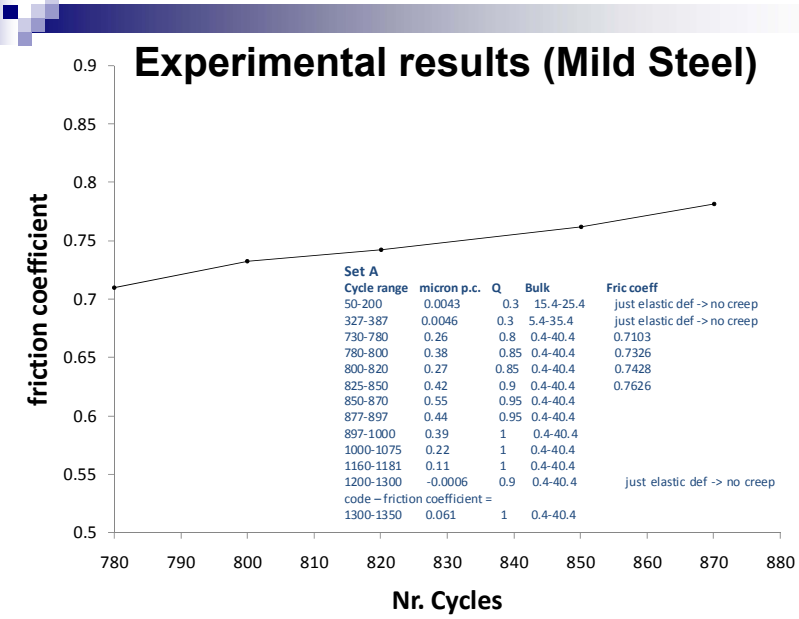

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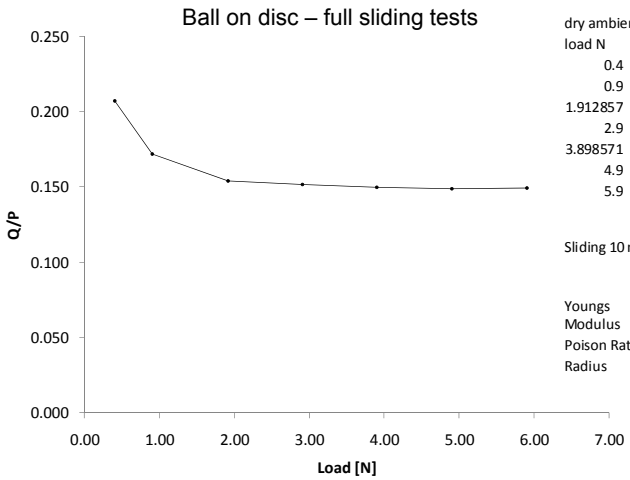
Pressure dependent friction?



dry ambient

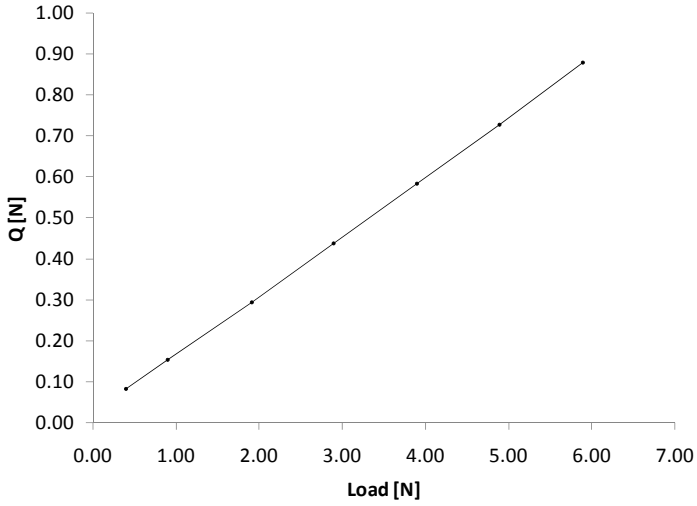
load N	friction coef
0.4	0.206947329
0.9	0.1714776
1.912857	0.153897914
2.9	0.151195286
3.898571	0.149629586
4.9	0.1485512
5.9	0.149006757

Sliding 10 mm/s

	Disc	Ball
Youngs Modulus	1.20E+08	2.07E+11
Poison Ratio	0.05	0.3
Radius		9.53E-03

Rolls-Royce

Pressure dependent friction?



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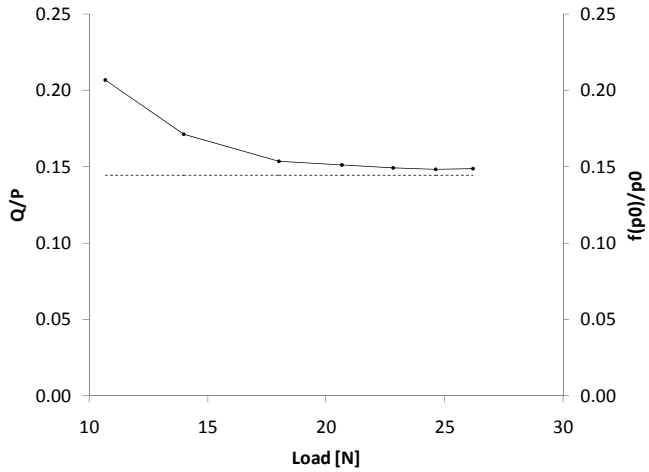
Pressure dependent friction?



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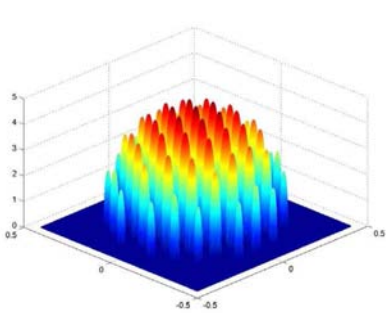
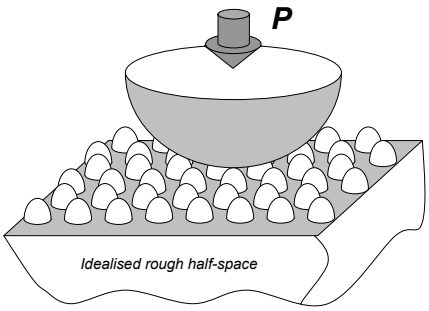
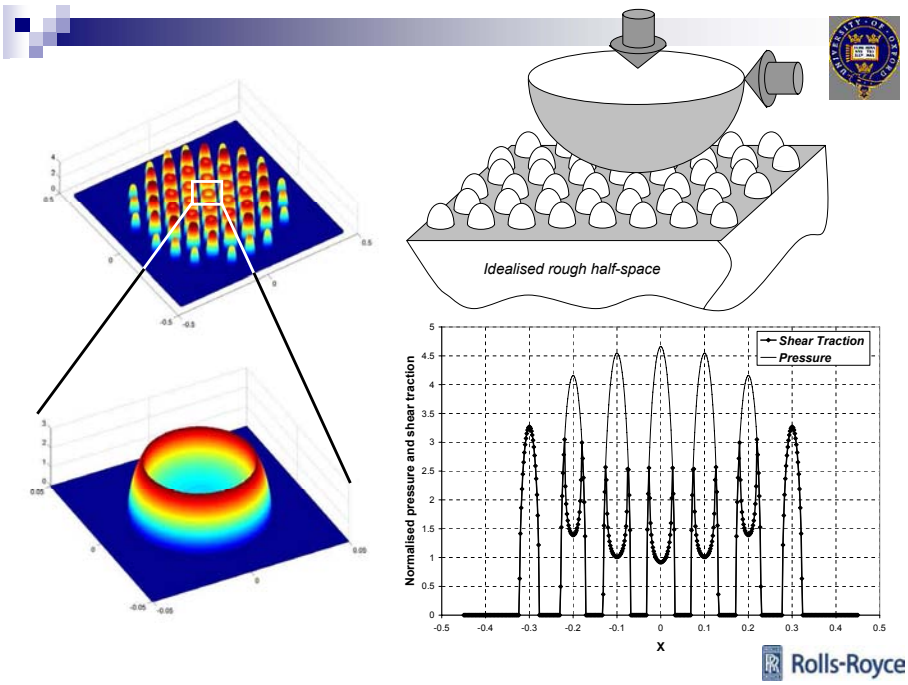
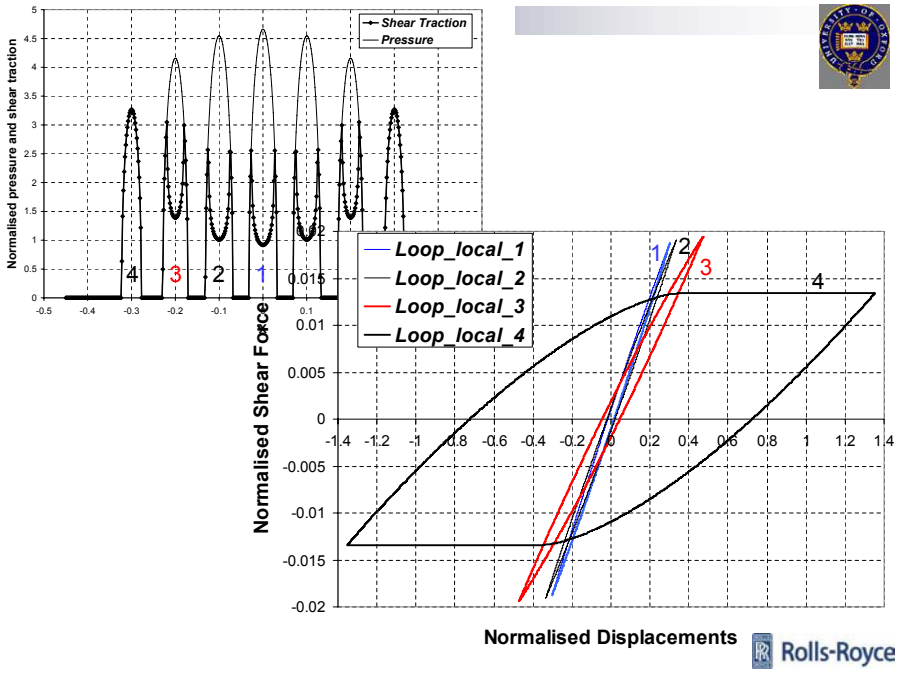
Energy Dissipation in Rough Contacts

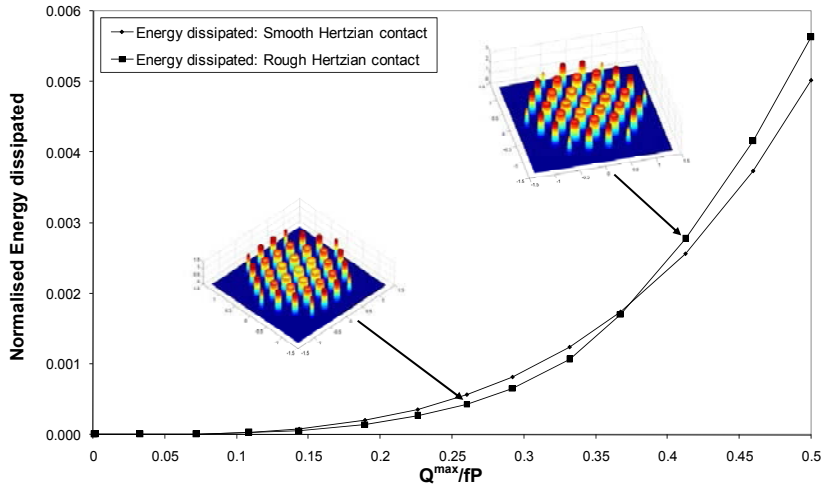




Rolls-Royce

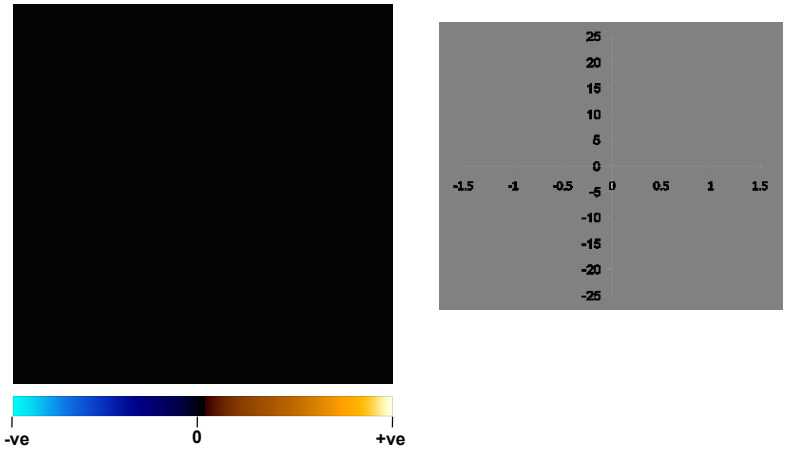

Idealised rough half-space
Shear Traction
Pressure
Normalised pressure and shear traction
x
Rolls-Royce


Shear Traction
Pressure
Normalised pressure and shear traction
Loop_local_1
Loop_local_2
Loop_local_3
Loop_local_4
Normalised Shear Force
Normalised Displacements
Rolls-Royce



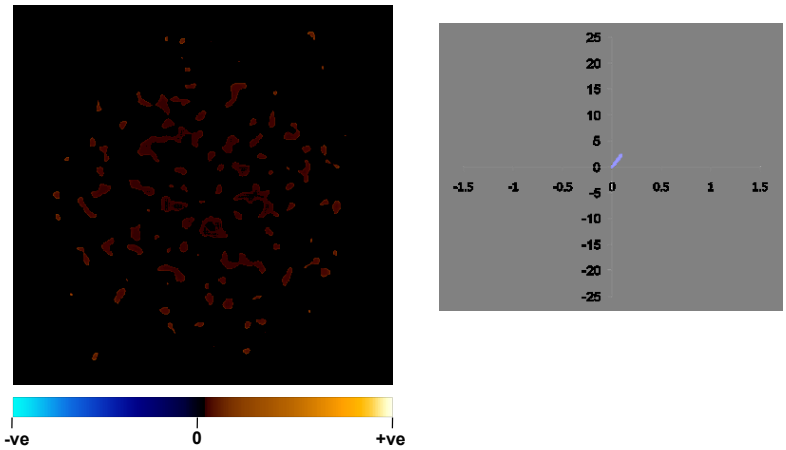
Rolls-Royce

Rough surface – Typical friction loop (traction)



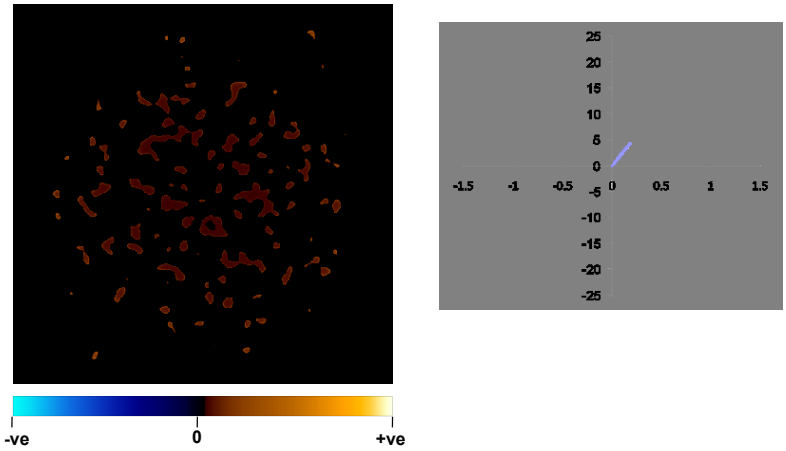
Rolls-Royce

Rough surface – Typical friction loop (traction)



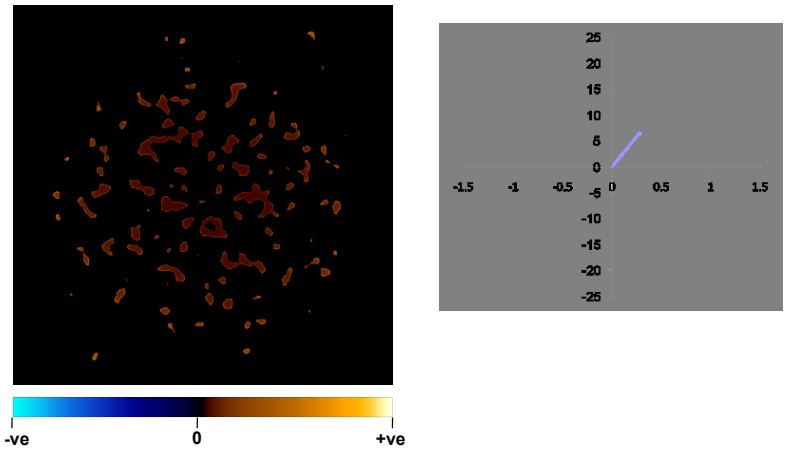
Rolls-Royce

Rough surface – Typical friction loop (traction)



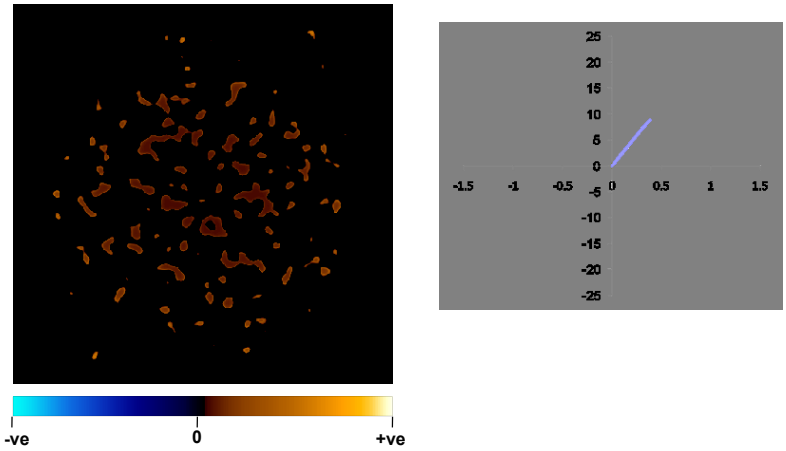
Rolls-Royce

Rough surface – Typical friction loop (traction)



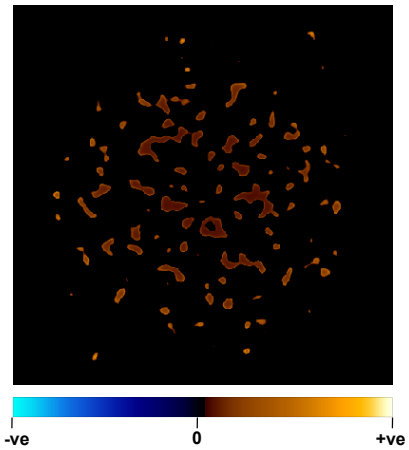
Rolls-Royce

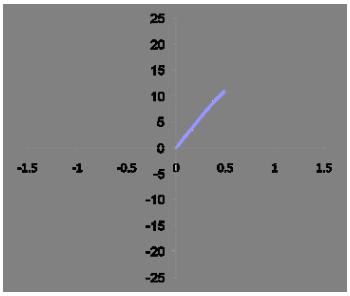
Rough surface – Typical friction loop (traction)



Rolls-Royce

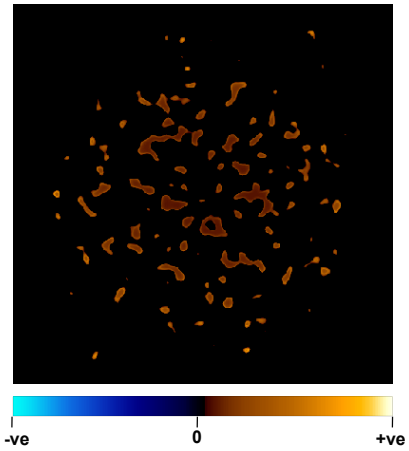
Rough surface – Typical friction loop (traction)

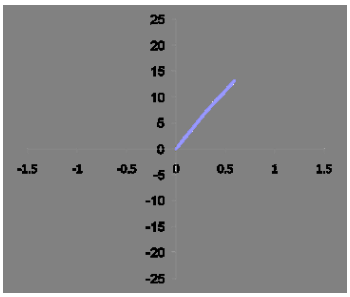




Rolls-Royce

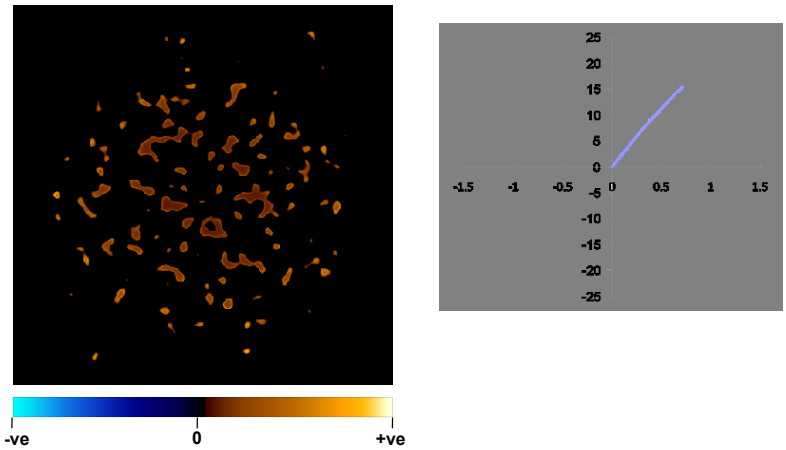
Rough surface – Typical friction loop (traction)





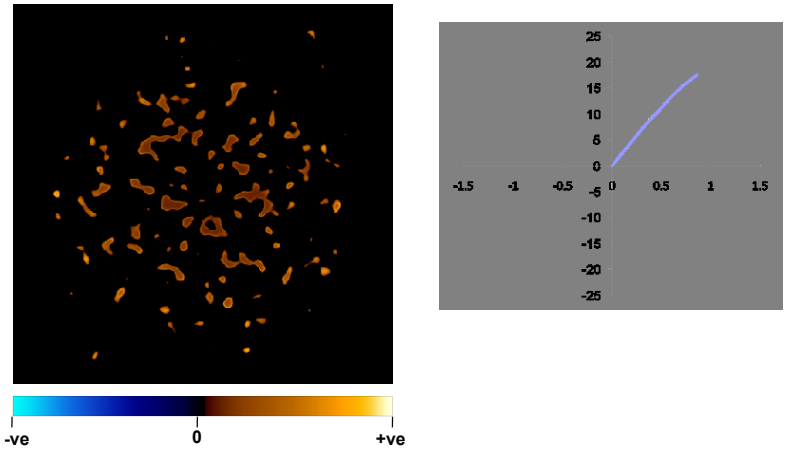
Rolls-Royce

Rough surface – Typical friction loop (traction)



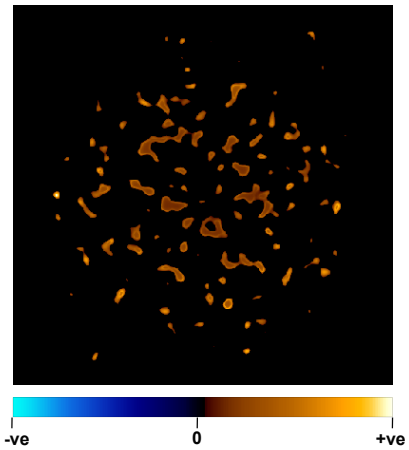
Rolls-Royce

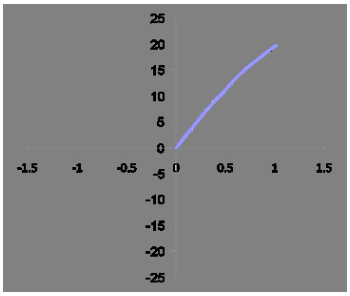
Rough surface – Typical friction loop (traction)



Rolls-Royce

Rough surface – Typical friction loop (traction)





Rolls-Royce

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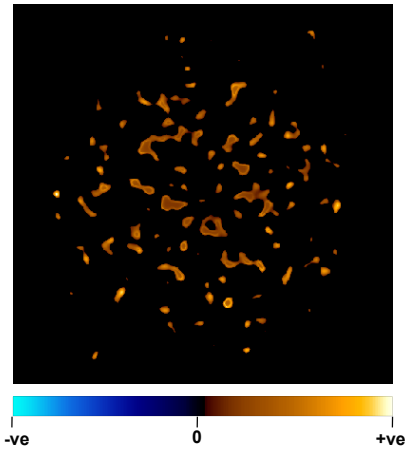
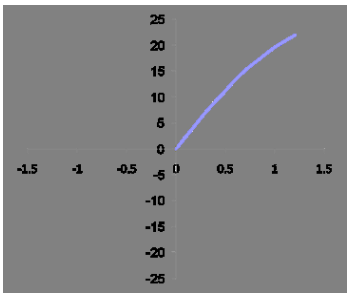
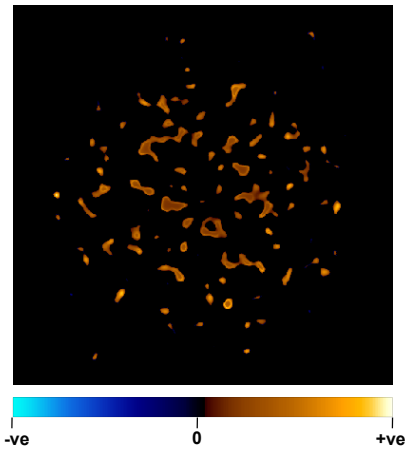
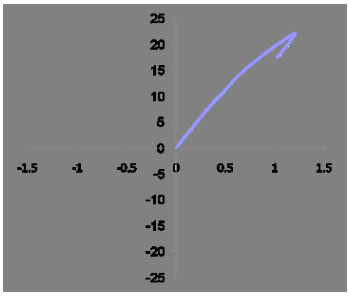




Rolls-Royce

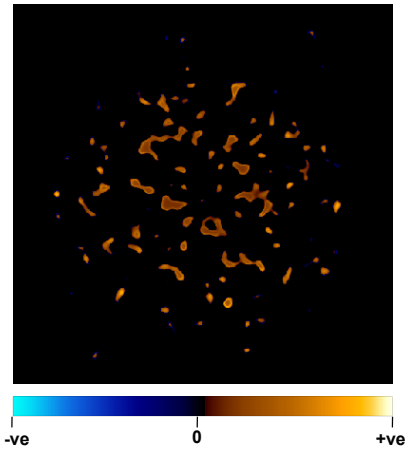
Rough surface – Typical friction loop (traction)

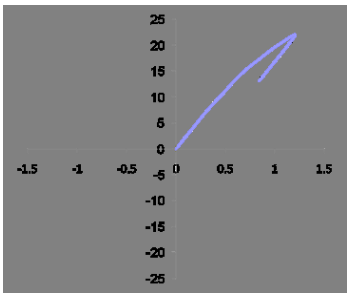




Rolls-Royce

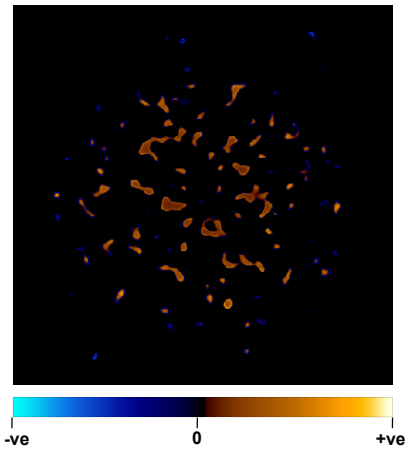
Rough surface – Typical friction loop (traction)

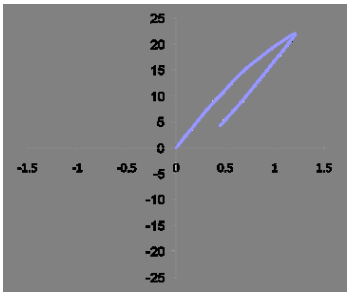




Rolls-Royce

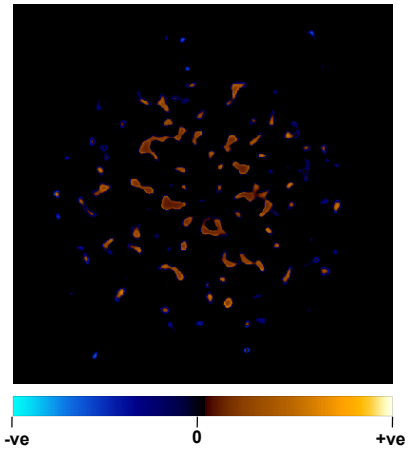
Rough surface – Typical friction loop (traction)

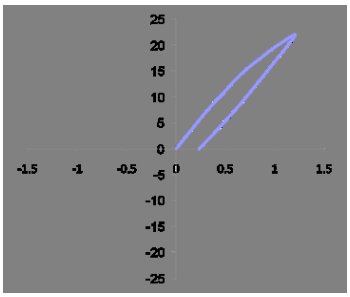




Rolls-Royce

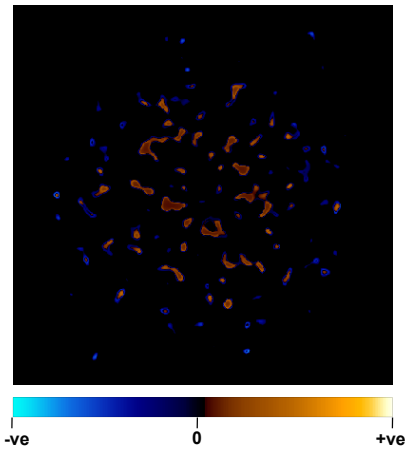
Rough surface – Typical friction loop (traction)

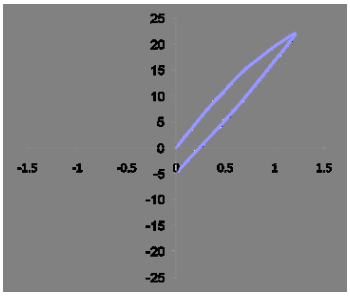




Rolls-Royce

Rough surface – Typical friction loop (traction)





Rolls-Royce

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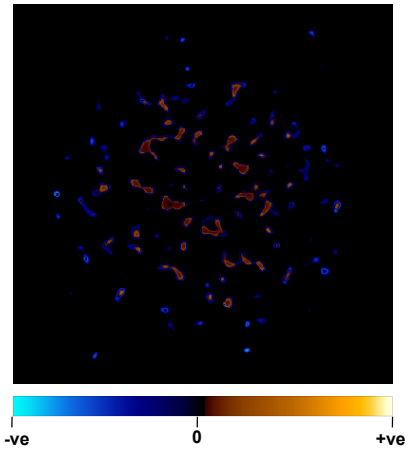
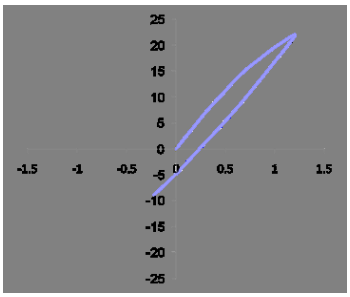
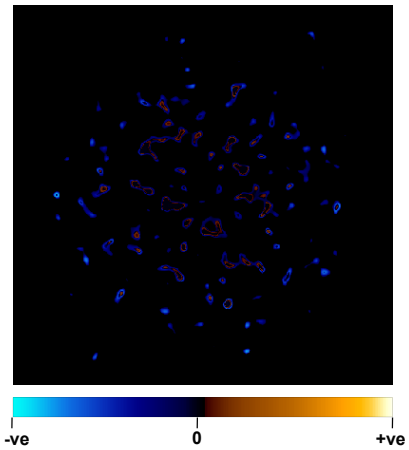
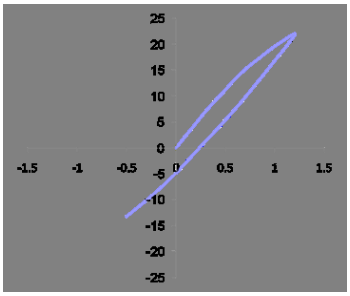




Rolls-Royce

Rough surface – Typical friction loop (traction)





Rolls-Royce

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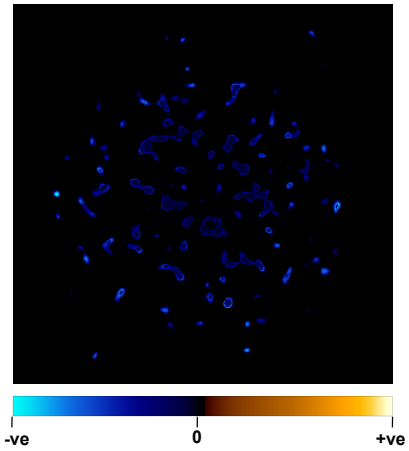
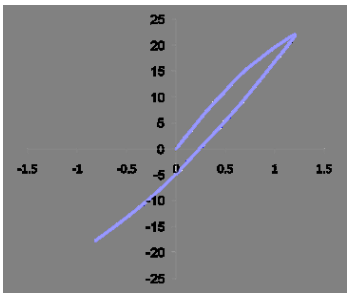
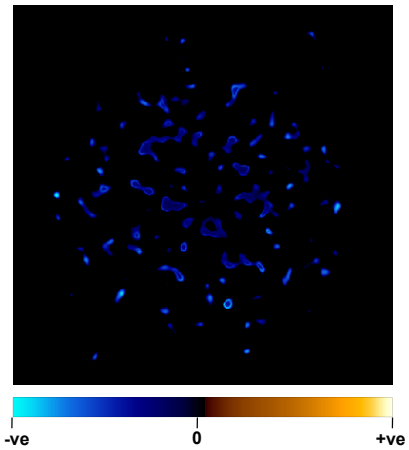
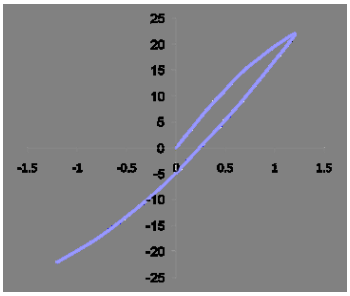




Rolls-Royce

Rough surface – Typical friction loop (traction)





Rolls-Royce

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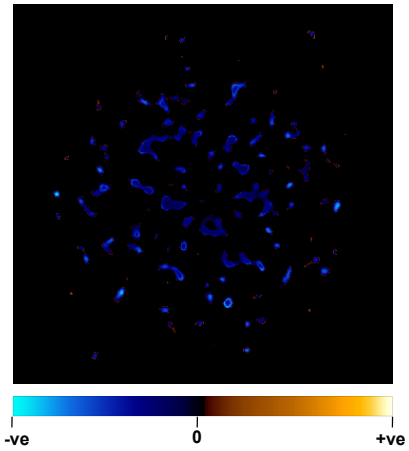
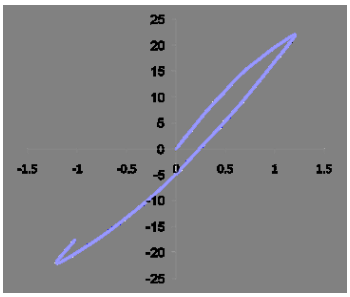
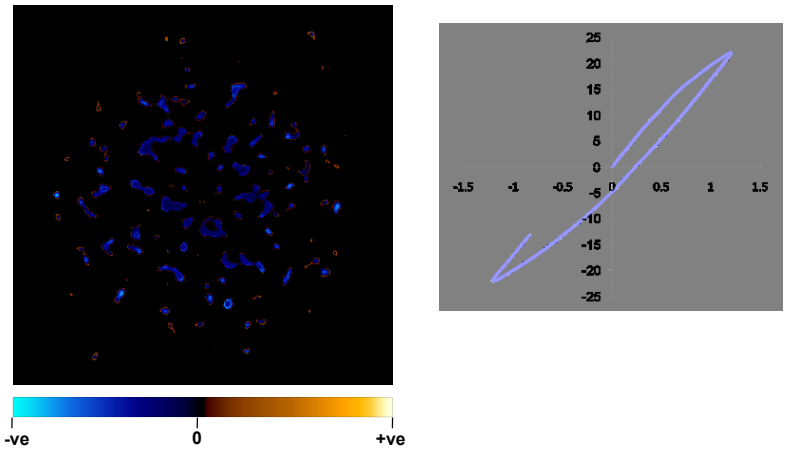




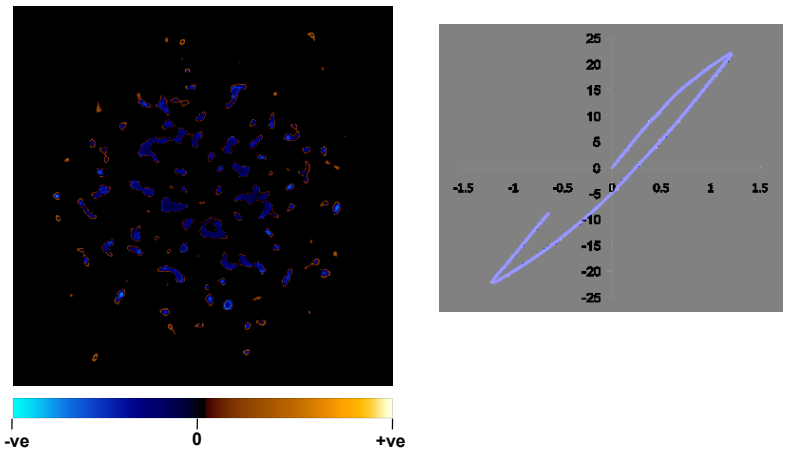
Rolls-Royce

Rough surface – Typical friction loop (traction)



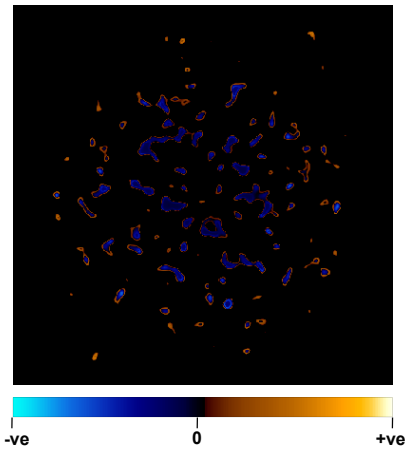
Rolls-Royce

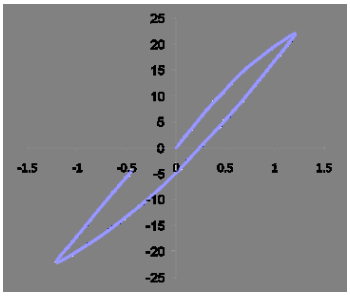
Rough surface – Typical friction loop (traction)



Rolls-Royce

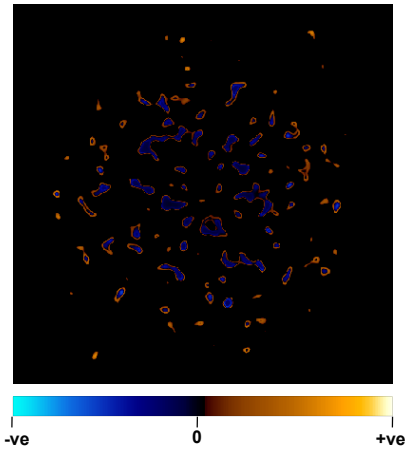
Rough surface – Typical friction loop (traction)

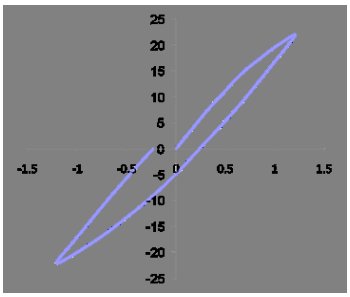




Rolls-Royce

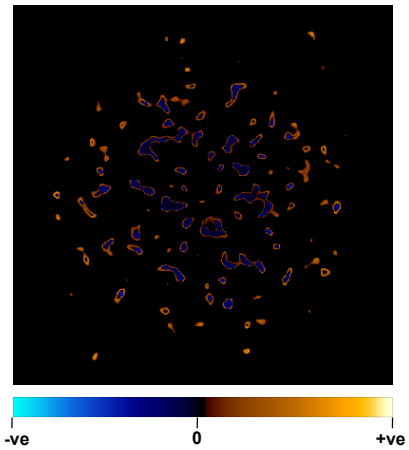
Rough surface – Typical friction loop (traction)

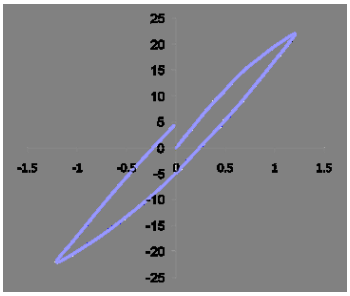




Rolls-Royce

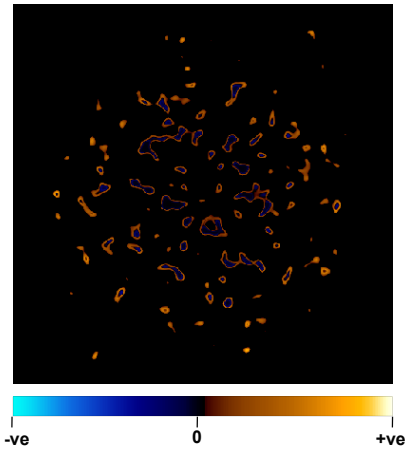
Rough surface – Typical friction loop (traction)

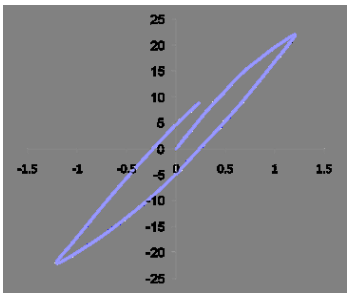




Rolls-Royce

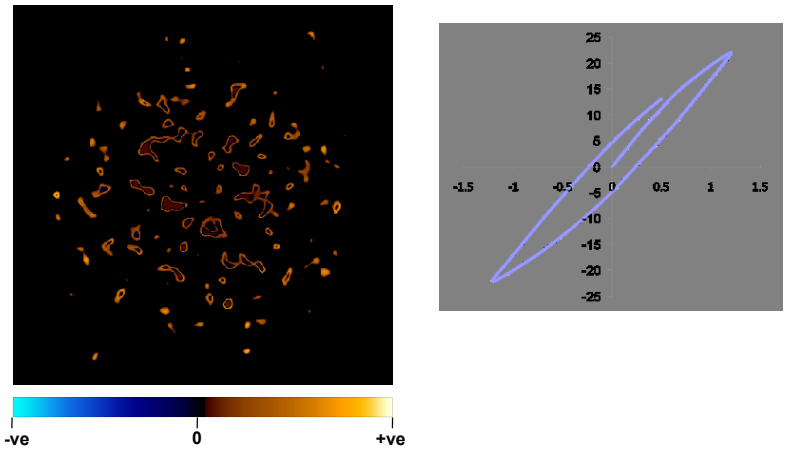
Rough surface – Typical friction loop (traction)





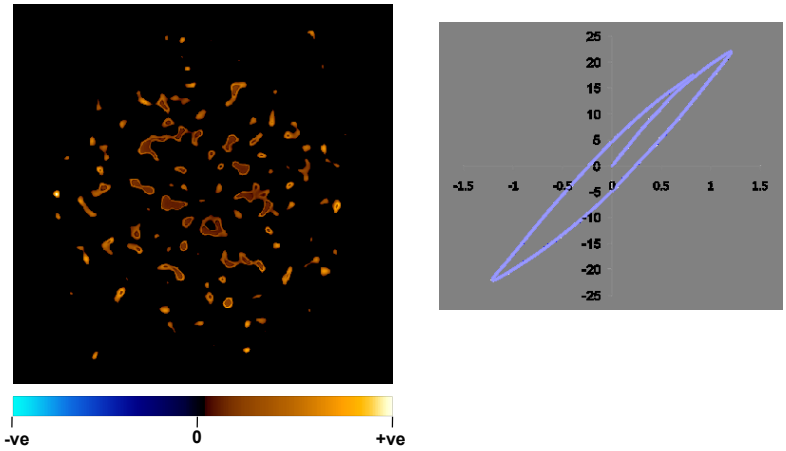
Rolls-Royce

Rough surface – Typical friction loop (traction)



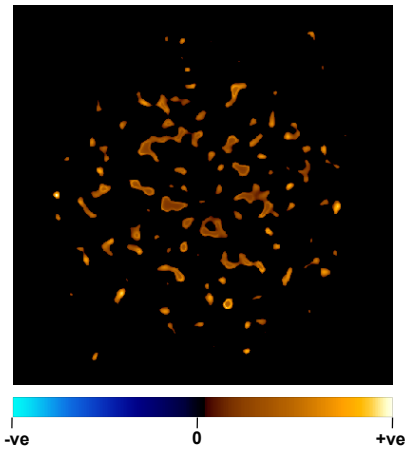
Rolls-Royce

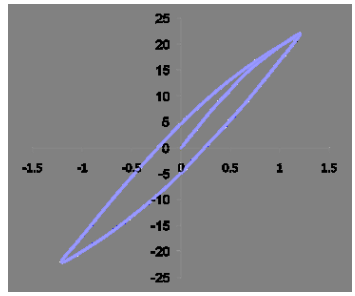
Rough surface – Typical friction loop (traction)



Rolls-Royce

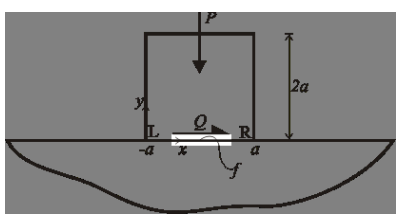
Rough surface – Typical friction loop (traction)



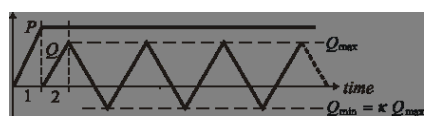


Rolls-Royce

Complete contacts – (1) Static

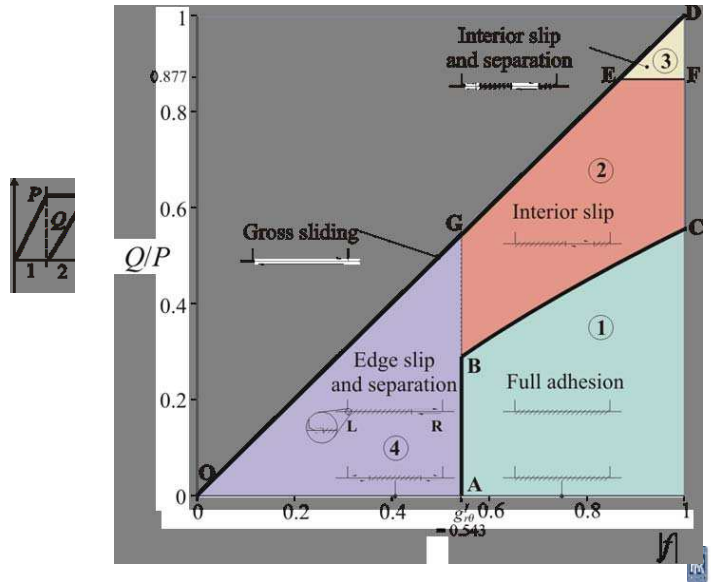


Example contact geometry



Loading history

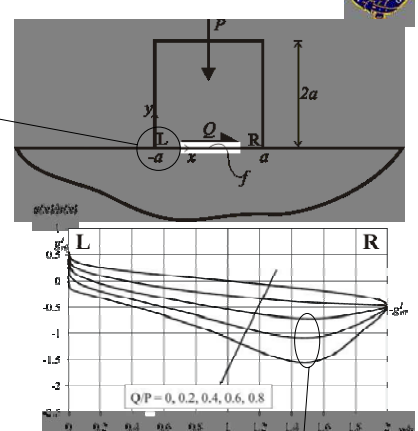
Rolls-Royce



Rolls-Royce

The traction ratio near the contact edge is

$$\frac{q(x)}{p(x)} = \frac{K_I^o g_{r\theta}^I x^{\lambda_I - 1} + K_{II}^o g_{r\theta}^{II} x^{\lambda_{II} - 1}}{K_I^o x^{\lambda_I - 1} + K_{II}^o x^{\lambda_{II} - 1}}$$



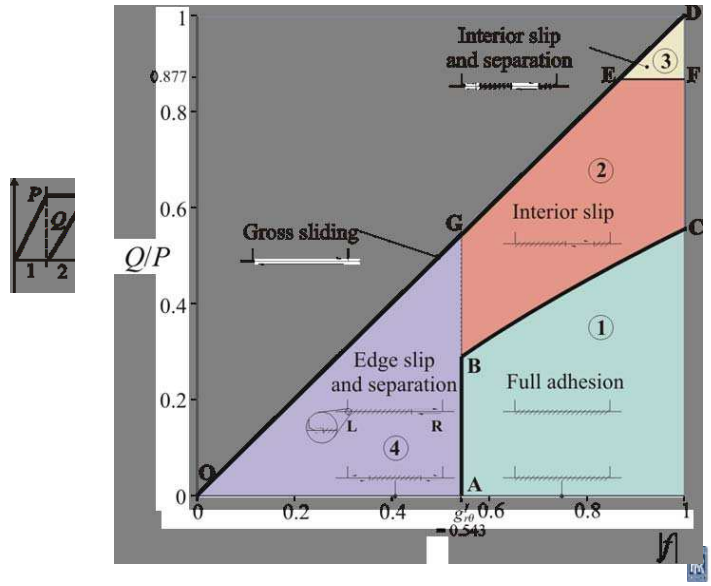
and because $\lambda_I - 1 < \lambda_{II} - 1 < 0$ the mode I singularity dominates the mode II singularity as $x \to 0$ to give

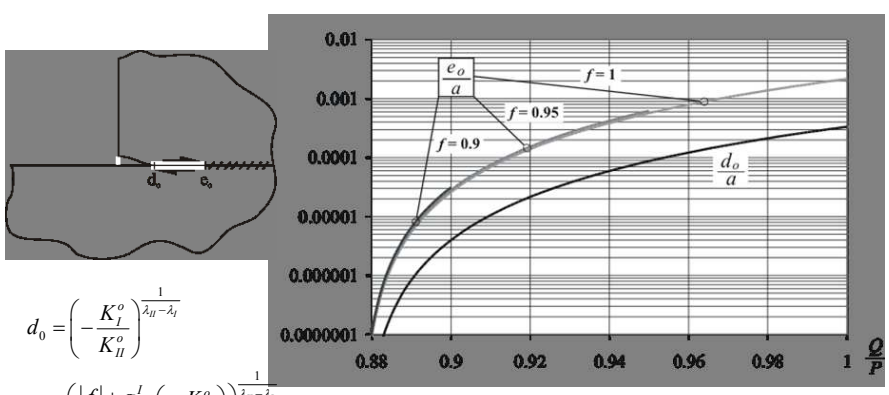
$$\left.\frac{q(x)}{p(x)}\right|_{x \to 0} = g_{r\theta}^I = 0.543$$

Therefore, *irrespective of the loading applied*, the traction ratio at the corner is **constant**.

$\lvert f \rvert < 0.543$ Slip at the edge of the contact	If interior maximum in traction ratio exceeds $\lvert f \rvert$ there is slip at an interior point.
$\lvert f \rvert > 0.543$ Adhesion at the edge	

Rolls-Royce

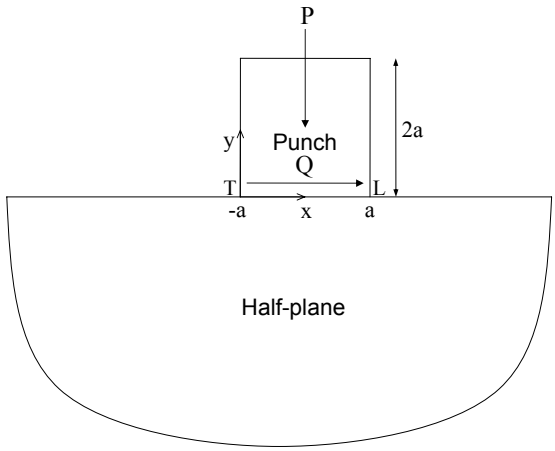




$$d_0 = \left(-\frac{K_I^o}{K_II^o}\right)^{\frac{1}{\lambda_{II}-\lambda_I}}$$

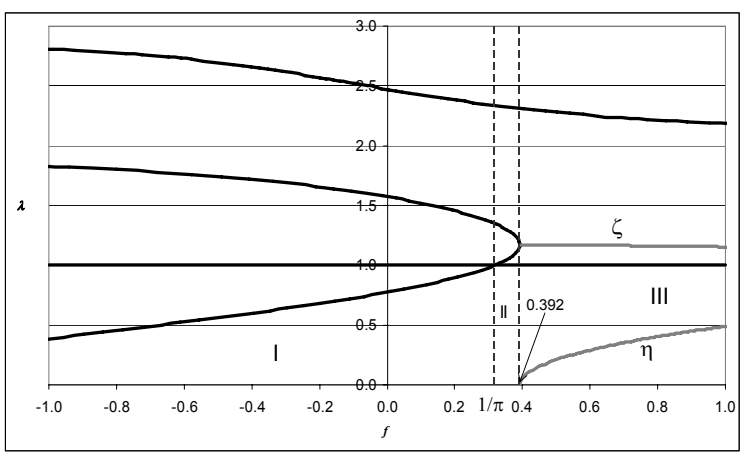
$$e_0 = \left(\frac{|f| + g_{r\theta}^{I}}{|f| + g_{r\theta}^{II}}\left(-\frac{K_I^o}{K_{II}^o}\right)\right)^{\frac{1}{\lambda_{II}-\lambda_I}}$$

Complete contacts – (2) Sliding



Rolls-Royce

Eigenvalues Plot



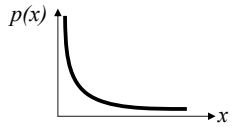
Leading edge Trailing edge

Rolls-Royce

Fundamental forms of the contact pressure

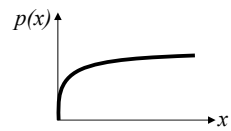
Region I : $f < 1/\pi$ (≈ 0.318)

$$p(x) = K_s x^{\lambda_s - 1} \qquad 0 < \lambda_s < 1$$



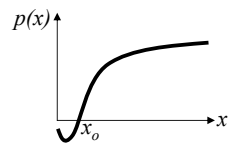
Region II : $0.318 < f < 0.392$

$$p(x) = K_{b1} x^{\lambda_{b1} - 1} + K_{b2} x^{\lambda_{b2} - 1} \qquad 1 < \lambda_{b1}, \lambda_{b2} < 2$$



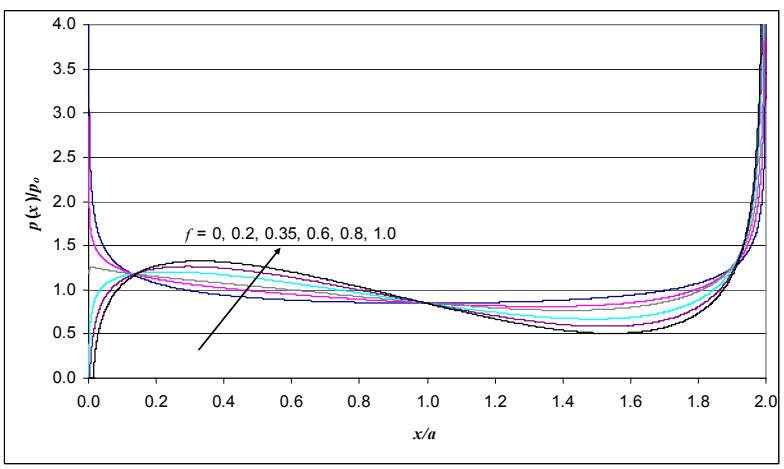
Region III : $f > 0.392$

$$p(x) = K_c x^{\zeta - 1} \sin\left(\eta \ln\left(\frac{x}{x_o}\right)\right) \qquad 1 < \zeta < 2; 0 < \eta < 1$$



Rolls-Royce

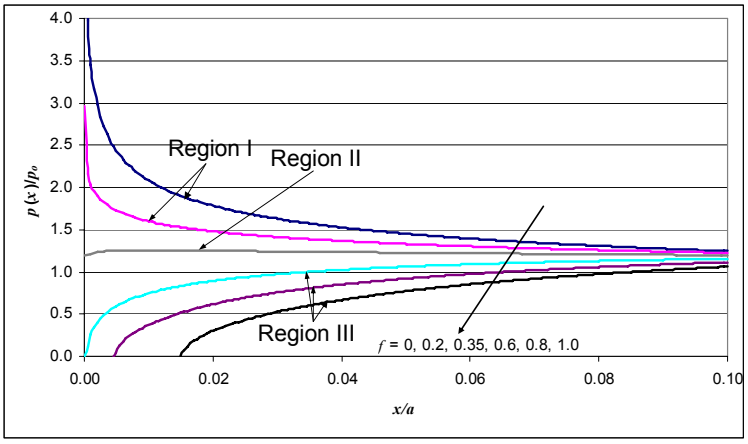
Contact pressure comparison



$$p_o = \frac{P}{2a}$$

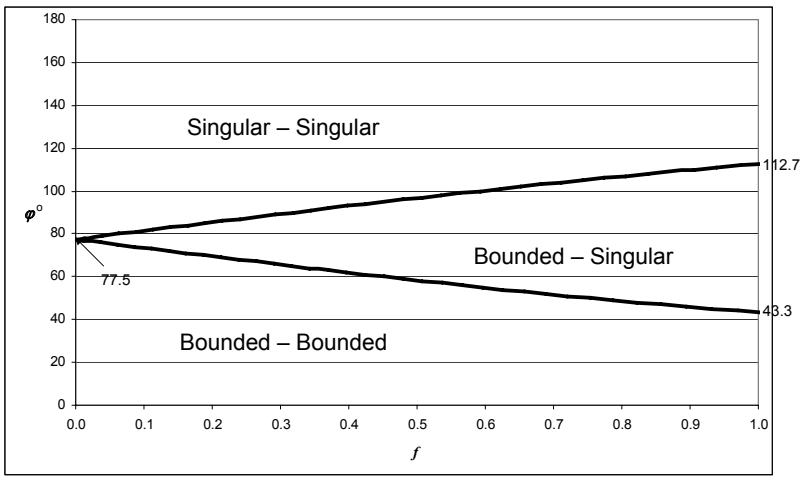
Rolls-Royce

Contact pressure comparison (trailing edge)

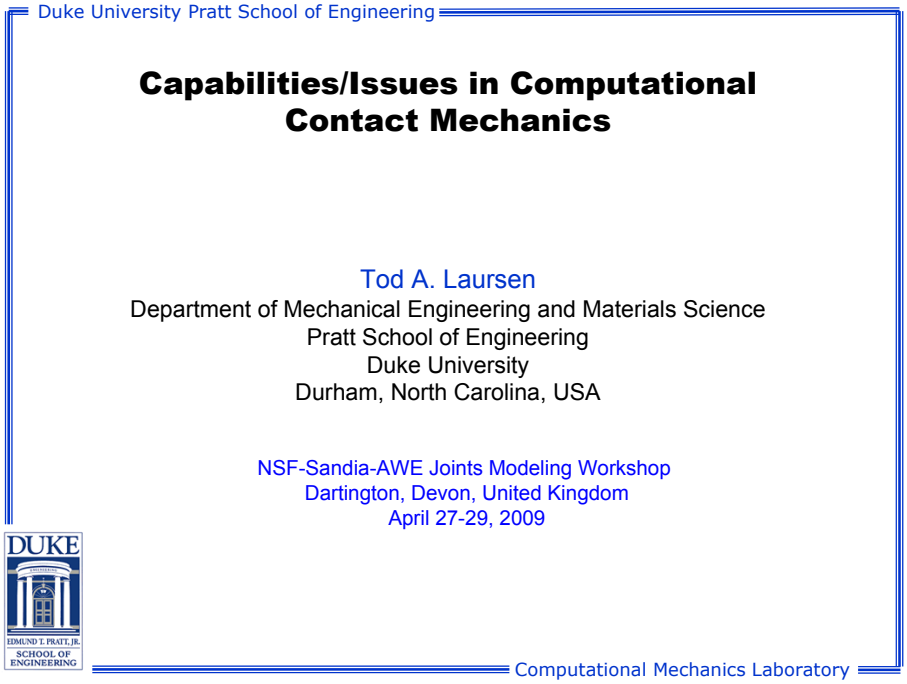


Rolls-Royce

General Behaviour Map of Sliding Contacts



Rolls-Royce



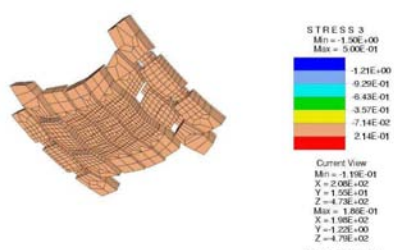
D.2.2 Slide Presentation of Tod Laursen, Duke University: *Capabilities/Issues in Computational Contact Mechanics*

Goal of our Research

Finite Element algorithms for large deformation, deformable-to-deformable contact in quasistatic and implicit dynamic analysis, suitable for:

- High fidelity prediction of frictional behaviors in a wide variety of physical settings (stick slip behavior in forming operations; microslip damping phenomena giving rise to structural damping; self-contact and frictional dissipation in tire rolling)
- Accurate treatment of impact phenomena, with careful attention in particular paid to conservation/dissipation of momenta and energy
- Increasingly, incorporation of tribological complexity in our capabilities for contact simulation (including lubrication)

An example from our collaboration with Michelin:



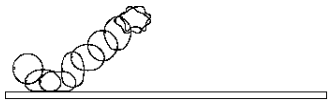
2

Some Past Efforts Relevant to this Goal

Energy-Momentum Formulation of Impact Interaction

Motivation: many traditional finite element integrators for impact interaction are only linearly stable, and in nonlinear impact calculations can readily produce unstable behavior

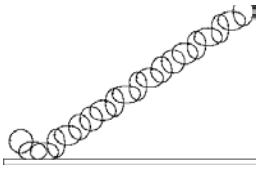
Example:HHT integration of ring impact



Idea: develop algorithms for impact that explicitly conserve energy (when appropriate), as well as linear and angular momentum

Accomplishments of this work (see Laursen & Chawla [1997]; Chawla & Laursen [1998]; Laursen & Love [2002]; Love & Laursen [2003]):

- Stable algorithms for conservative (frictionless) contact without introduction of nonphysical damping
- Introduction of surface and bulk dissipation (inelasticity) in a manner consistent with underlying thermodynamics
- New notions of temporal accuracy, and corresponding implementations,within an energy-momentum framework



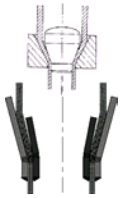
Stable, energy-momentum solution

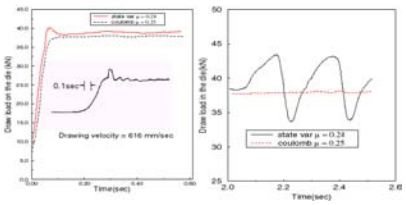
Past Efforts (Cont.)

Complexity on Interfaces: Multifield Coupling and Tribological Modeling

Motivation: many applications demand sophisticated interface constitutive laws to describe observed phenomena

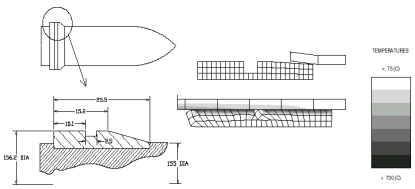
Example: chatter instabilities in drawing applications (Oancea and Laursen [1997, 1998]





Accomplishments of our research:

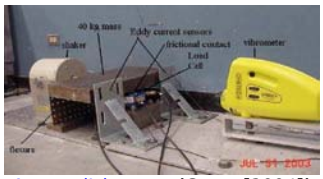
- Theoretical framework enabling stable extension of mechanical descriptions to encompass thermomechanical coupling, enabling simulation of frictional heating, thermal softening (as in shell firing simulation, right)
- Implementations of frictional rate dependence, enabling simulation of unstable slip (see above)

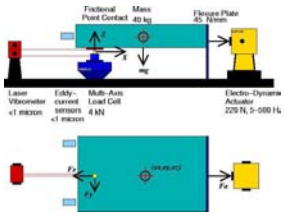


Past Efforts (Cont.)

Microslip Damping/Hysteresis Prediction without Phenomenology

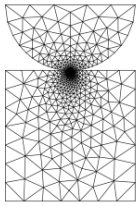
Motivation: Many structural damping applications are limited by reliance on phenomenological results, in which distinctions between bulk compliance and surface effects cannot be drawn

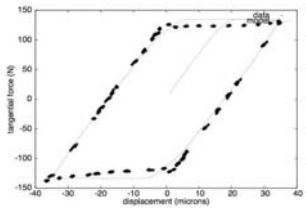




Recent Accomplishments (Greer [2004]):

- Good representation of hysteretic behavior in finite element models of Mindlin-type experiments (friction law requires two inputs which are readily measured experimentally: mu and an interface stiffness)

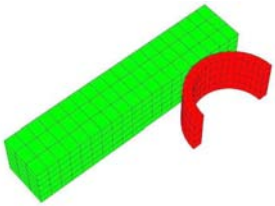




Some Preliminary Observations

From this (very brief!) examination of the types of problems we are interested in, we infer some needs that have driven our research in the past few years:

- We would like methods of contact analysis that are accurate: (ideally, presence of contact should not degrade spatial convergence rates expected from underlying finite element methods)
- We need numerical robustness (particulaly in large sliding and/or deformation applications, where connectivity continually changes throughout the simulation)
- We want broad applicability: two and three dimensions, with a variety of material models, with and without friction, extendible to tribologically complex settings (including rate dependence, anisotropy, lubrication)



Finite Element Formulation

If we approximate this system by imposing a finite element grid, we end up with an equation system of the form

$$\boldsymbol{M}\ddot{\boldsymbol{d}}(t) + \boldsymbol{F}^{int}(\boldsymbol{d}(t)) + \boldsymbol{F}_c(\boldsymbol{d}(t)) = \boldsymbol{F}^{ext}(t)$$

where

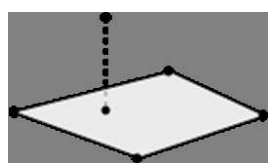
- $\boldsymbol{M}$ is the mass matrix
- $\boldsymbol{F}^{int}$ is the internal force vector, a generally nonlinear function of $\boldsymbol{d}$
- $\boldsymbol{F}_c$ is the contact force vector, subject to the aforementioned restrictions
- $\boldsymbol{F}^{ext}$ is the imposed external loading

In solving such a system, several challenges manifest themselves:

- Nonlinear equation solving (subject to nonsmooth constraints)
- Potential ill-conditioning
- Stability problems in dynamics
- Detection of contact (i.e., searching)
- Spatial discretization of contact constraints, and its effect on the results obtained

Traditional Approach to Contact Mechanics in Finite Element Analysis

"Node to Surface" Contact, where constraints are imposed for nodes with respect to opposing element surfaces:



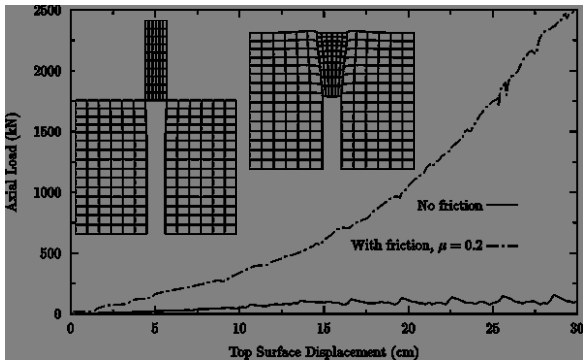
We can think of this as sort of a collocation approach, with the collocation points being the nodes of one side (or both)

Some problems with this approach are evident

- When nodes slide across element boundaries, nonsmoothnesses are introduced
- Low order solutions are not admitted by the formulation
- Accurate contact traction recovery is difficult
- Non-conforming FE approximation→suboptimal convergence

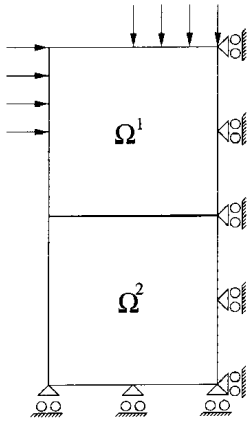
One Limitation of Traditional Approach: Nonsmoothnesses from Faceted Geometries

Because node to surface schemes enforce constraints with respect to a faceted geometry, both convergence difficulties and nonphysical results are to be expected in deformable interface contact problems.

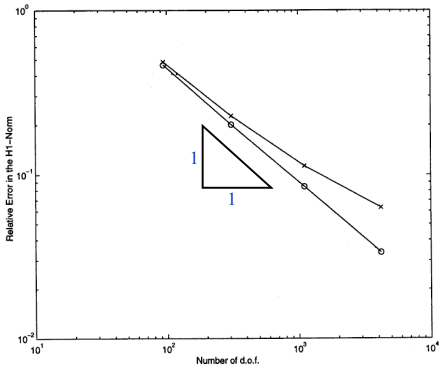


Another Limitation: Convergence is Demonstrably Degraded in Node to Surface Treatments

Convergence Study due to Hild [2000]



Test problem



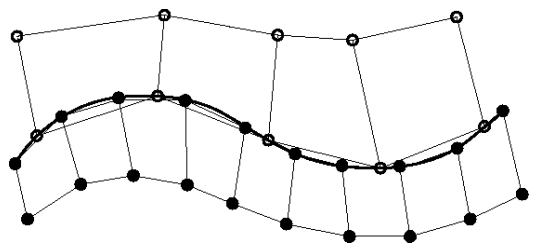
○ Mortar Method

× Node-to-Segment

A (Relatively) New Approach: Mortar-Finite Element Methods

The "node to surface" contact formulation is avoided by considering an integral formulation of contact conditions

- To demonstrate this idea, consider the mesh tying problem as a template
- This is a problem of great practical importance: dissimilar discretizations of the same curve give rise not only to "contact-like" geometries, but also to areas of gaps and overlaps between surfaces to be joined



Basic Idea of Mortar Concept

For the tying problem, we enforce compatibility of a least squares projection of one displacement field with that of the opposing surface.

Key ideas:

- Integral representation of displacement continuity

$$0 = \int_{\Gamma_c^{(1)^h}} \boldsymbol{q}^h \cdot (\boldsymbol{U}^{(1)^h} - \boldsymbol{U}^{(2)^h})\, d\Gamma$$

where the multipliers (tractions) $\boldsymbol{\lambda}^h$ are interpolated via

$$\boldsymbol{\lambda}^h = \sum_{A=1}^{nnod(1)} \boldsymbol{\lambda}_A N_A$$

- Combination of the above leads to constraints c_A of the form

$$0 = \boldsymbol{c}_A = \int_{\Gamma_c^{(1)^h}} N_A \left(\sum_{B=1}^{nnod(1)} \boldsymbol{U}_B N_B - \sum_{\hat{B}=1}^{nnod(2)} \boldsymbol{U}_{\hat{B}} N_{\hat{B}} \right) d\Gamma$$

involving inner products of shape functions, i.e. $M_{AB} = \int_{\Gamma_c^{(1)^h}} N_A N_A \, d\Gamma$

We Use this Idea for Discretization of Contact Interaction, but Recognize that Mortar Integrals must Depend on Deformation

Contact virtual work:

$$G^{cm}(\boldsymbol{\varphi}^h, \overset{*}{\boldsymbol{\varphi}}{}^h) := -\int_{\gamma_c^{(1)^h}} \boldsymbol{\lambda}^h(\boldsymbol{X}) \cdot \left(\overset{*}{\boldsymbol{\varphi}}{}^{(1)^h}(\boldsymbol{X}) - \overset{*}{\boldsymbol{\varphi}}{}^{(2)^h}(\bar{\boldsymbol{Y}})\right) d\gamma$$

- The discretized contact traction and deformation fields are defined as:

$$\boldsymbol{\lambda}^h(\boldsymbol{X}) = \sum_{A=1}^{n_s} N_A^{(1)}\left(\boldsymbol{\xi}^{(1)}(\boldsymbol{X})\right)\boldsymbol{\lambda}_A \qquad \boldsymbol{\varphi}^{(1)^h}(\boldsymbol{X}) = \sum_{D=1}^{n_s} N_D^{(1)}\left(\boldsymbol{\xi}^{(1)}(\boldsymbol{X})\right)\boldsymbol{\varphi}_D^{(1)}$$

(etc. for the other fields)

- Finally, the discretized contact virtual work is

$$G^{cm}(\boldsymbol{\varphi}^h, \overset{*}{\boldsymbol{\varphi}}{}^h) = -\sum_A \sum_B \sum_C \boldsymbol{\lambda}_A \cdot \left[n_{AB}^{(1)} \overset{*}{\boldsymbol{\varphi}}{}_B^{(1)} - n_{AC}^{(2)} \overset{*}{\boldsymbol{\varphi}}{}_C^{(2)}\right]$$

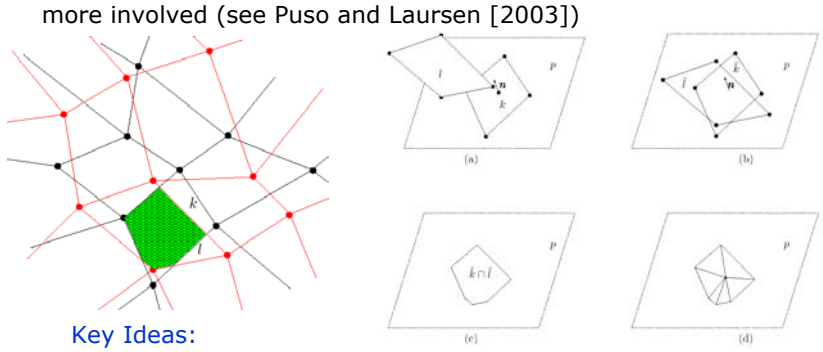
- Where the mortar integrals are now computed in the current configuration

$$n_{AB}^{(1)} = \int_{\gamma^{(1)^h}} N_A^{(1)}\left(\boldsymbol{\xi}^{(1)}(\boldsymbol{X})\right) N_B^{(1)}\left(\boldsymbol{\xi}^{(1)}(\boldsymbol{X})\right) d\gamma$$

$$n_{AC}^{(2)} = \int_{\gamma^{(1)^h}} N_A^{(1)}\left(\boldsymbol{\xi}^{(1)}(\boldsymbol{X})\right) N_C^{(2)}\left(\boldsymbol{\xi}^{(2)}\left(\bar{\boldsymbol{Y}}(\boldsymbol{X})\right)\right) d\gamma$$

Computation of mortar integrals: three dimensions

Computation of $n_{AB}^{(1)}$ and $n_{AC}^{(2)}$ in three dimensions uses extension of same idea, but algorithm is necessarily much more involved (see Puso and Laursen [2003])



Key Ideas:

(1) Define the flat projection surface *p*

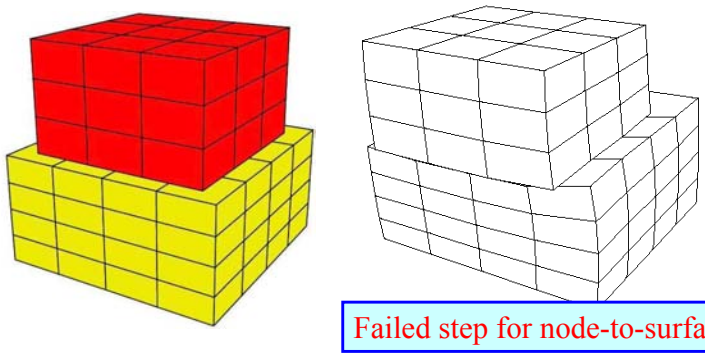
(2) Project slave and master elements onto the surface *p*

(3) Find the intersection of the projected polygons

(4) Divide into triangles to perform numerical integration

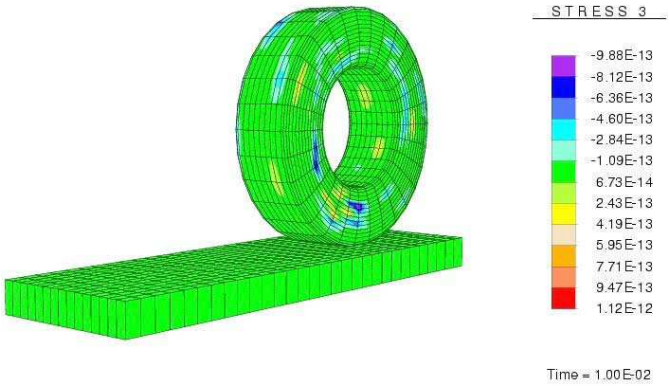
Some Numerical Examples

This simple 3D problem demonstrates robustness when nodes leave contact, resulting from nonlocal constraint definitions



- Sliding and pressing the upper block, part of the upper block slides out of the lower block
- The node-to-surface fails at t=0.29

An Industrial "Toy" problem for Michelin (self-contact occurs inside the tire also)



Current State of the Art

In computational contact mechanics, the past few years have seen several advances

- Energetically consistent algorithms for contact mechanics
- Some incorporation of new constitutive models for friction (including thermomechanical contact)
- New classes of algorithms giving much greater numerical accuracy and robustness (mortar methods)
  - As D. Segalman mentioned yesterday, this is not to be taken for granted (physics issues aside)

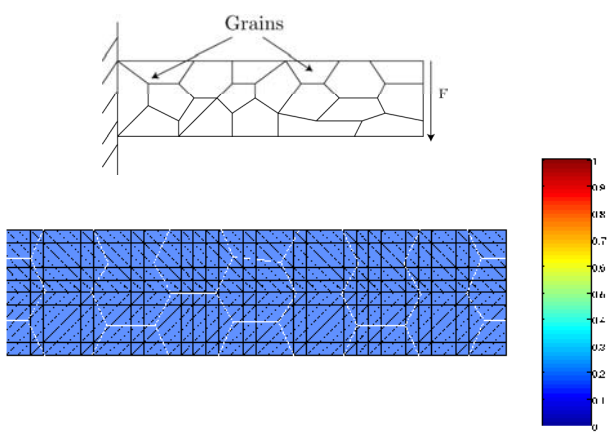
Challenges:

- Mesh density requirements for good resolution of lubrication, dry friction damping
- Multiscale
- Is there a chance of avoiding explicit gridding of interfaces altogether?

We Think Yes

A New Direction (joint with J. Dolbow): XFEM treatment of interfaces

Polycrystalline elastic beam bending:



D.2.3 Slide Presentation of Muzio Gola, Politecnico di Torino

Politecnico di Torino

AERMEC Laboratory, Politecnico di Torino
Dept. of Mechanical Engineering – the team

Co-ordinator

M. M. **Gola**

Assistant professors

T. **Berruti**

D. **Botto**

S. **Zucca**

Research Assistants

C. **Firrone**

M. **Lavella**

A. **Campagna**

D.**Zanello**

V. **Maschio**

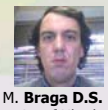
PhD students

S. **Pavone**

P. **Vargiu**

L. **Tong (CHINA)**

Visiting researchers



M. **Braga D.S.**
University of Uberlandia
BRASIL

C. **Siewert**
University of Hannover
GERMANY

(end 2008)

Current main contracts and research programs

European Union Projects

DREAM - ValiDation of Radical Engine Architecture systeMs (2008-2011)
FUTURE - Flutter-free turbomachinery blades (2008-2011)
PREMECY- Subcontract for high mean value fatigue test (2007-2010)
VITAL – EnVIronmenTALly Friendly Aero Engine (2005-2008)
VERDI – Virtual Engineering for Robust Manufacturing with Design Integration (2005-2008)

Italian government Research Grants

GREAT 2020 – Green Engine for Ait Traffic (2009-2011)
CORALE - low environment impact aeroengine (2007-2010)
PRIN Design criteria for mistuned turbomachinery (2006-2009)

Research Contracts with AVIO Group

High temperature tribology for turbine materials (2007-2009)
Design of damper rings for aerospace application (2008-2010).
Study of turbine disk vibrations with MISTUNING (2007-2008)

Research lines

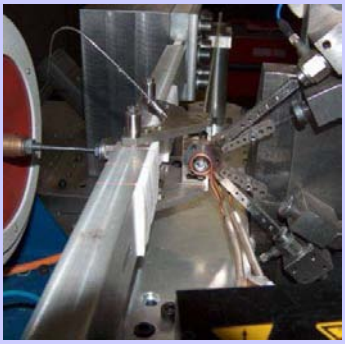
- Contact mechanics & contact modelling
- Modelling damping components (underplatform, shroud, blade root)
- Dynamic response of turbine discs

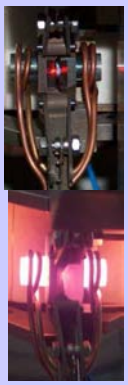
Complementary activities

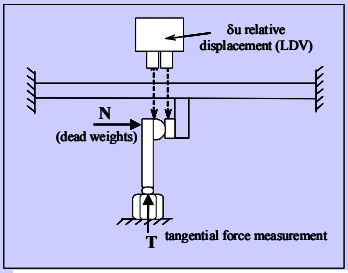
- Tribology, wear measurement
- Spin-test rig (work in progress)
- Dynamics of turbine disks with asymmetrical effects (MISTUNING)
- Real time evaluation of temperature and thermal stresses at critical locations of turbine disc (disc lifing)
- X Ray evaluation of residual stress in turbine components

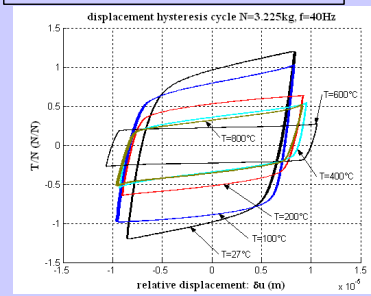
Contact mechanics - 1

High Temperature Test Rig









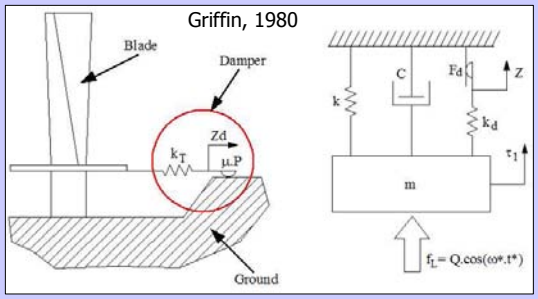
Working range:

- Displacement: 0,1 mm - 100mm
- Normal loading: 1kg a 10kg
- Operating frequency: 1 - 100Hz
- Induction heating 20 – 80 kHz
- Temperature : 20 - 800°C

Contact mechanics - 2

State-of-the-Art

Turbine blades vibration: friction damping



Griffin, 1980:
amplitude of resonant response of an airfoil with blade-to-ground friction damper

Assumptions: Coulomb friction law (no microslip), damper as a mass-less spring of stiffness k_T, use of the Ritz method to found the phase and amplitude.

- Srinivasan & Cutts, 1983: damping due to shrouds
- Menq & Griffin, 1985: use of HBM and FEM
- Menq, 1986: variable normal loading with Coulomb friction, HBM
- Cameron & Griffin, 1989: steady-state response with frequency domain method
- Sanliturk & Ewins, 1999: 2D motion and microslip
- Swedowicz, 2003: determination of contact stiffness of a friction damper
- Koh & Griffin, 2006: model of friction damper with spherical heads

Contact mechanics - 3

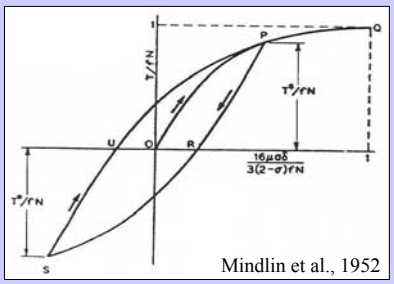
State-of-the-Art

Cattaneo-Mindlin contact model

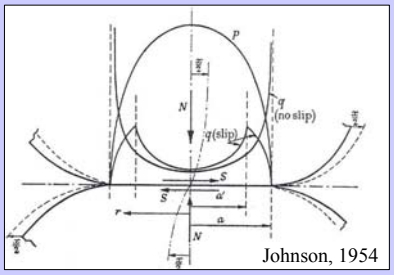
Hertzian theory extended to the case of tangential loading

Hypothesis: non-conform contact type, absence of asperities (smooth surfaces), elastic materials, Coulomb law at local level: t =m·p

Experimental validation: Mindlin, Mason, Osmer, Deresiewicz -1951; Goodman, Bowie -1961; Goodman, Brown –1962, Johnson –1955 e 1962



Mindlin et al., 1952



Johnson, 1954

There is an extension to conform contacts (1990, Ciavarella, Farris, Hills&Nowell, ...)

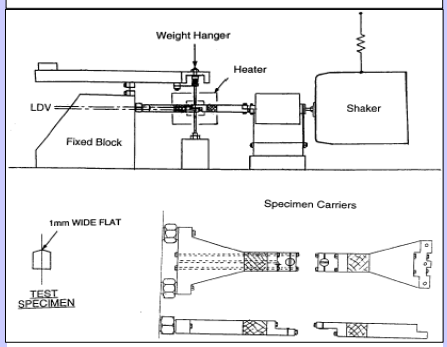
Other models consider another factors, e. g. roughness (Bowden & Tabor, Greenwood & Williamson, Archard, O'Connor & Johnson), velocity, etc.

Contact mechanics - 4

State-of-the-Art

The test rig at Imperial College 1999

Flat-on-flat contact type, measurements after wearing-off the surfaces



1999, A B Stanbridge, K Y Sanliturk & D J Ewins, "Measurement and Analysis of High-Temperature Friction Damper Properties" - Imperial College

Friction behaviour associated with fretting fatigue

Reference to Murthy et al. (2002), Murthy and Farris (2003), Matlik and Farris (2003), who investigated fretting fatigue as a function of temperature in advanced materials utilized in turbine engine components.

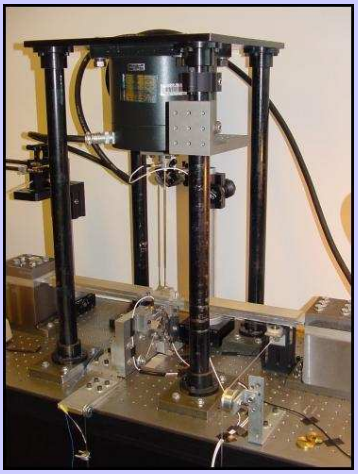
Tests at temperature up to 610°C in the contact region demonstrated that the friction coefficient increased with the wear of contact surfaces, and that the friction coefficient is dependent on the contact history.

Contact mechanics - 5

the Pittsburgh-Polito 1D test rig – room temperature

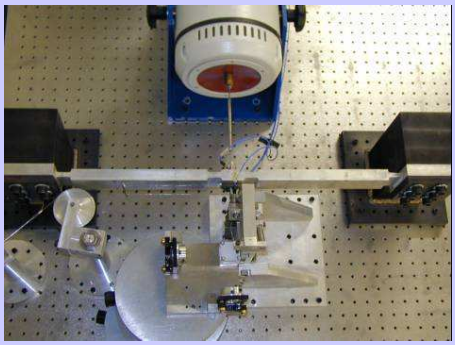
in co-operation with Prof. A. Akay

Carnegie Mellon University



Filippi, S., Akay, A., Gola, M. M., 2004, "Measurement of Tangential Contact Hysteresis During Microslip", ASME Journal of Tribology, v. 126-3 July, pp. 482-489.

Koh, K-H., Griffin, J.H., Filippi, S., Akay, A., 2004, "Characterization of Turbine Blade Friction Dampers", Proceedings of ASME Turbo Expo 2004, June 14-17, Vienna Austria, GT2004-53278.

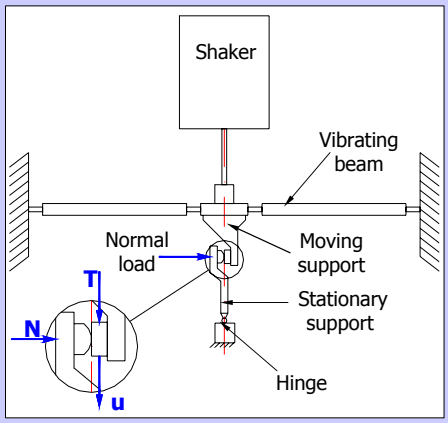


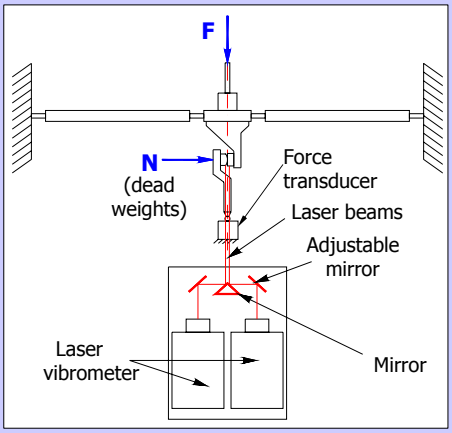
Contact mechanics - 6

the Pittsburgh-Polito 1D test rig – room temperature

Design Concepts

Working principles: a shaker excites the vibrating beam, and so the moving specimen (flat). The other is stationary.





Measurement system: force transducers measure the tangential force. Relative displacements by two LDV beams.

Contact mechanics - 7

the Pittsburgh-Polito 1D test rig – room temperature

Design Requirements at room temperature

- **Force measurements** as close as possible to the contact
- Constant normal loading, to avoid **dynamic effects**
- **Unidirectional motion**, cyclic
- **Rotations** of the friction pair must be avoided
- For one of the contact surfaces: **negligible stiffness** in the direction normal to the contact and **small mass**
- **Replaceable** contact surfaces;
- Measurements at a **wide range** of normal loading, relative displacement and frequency excitation;

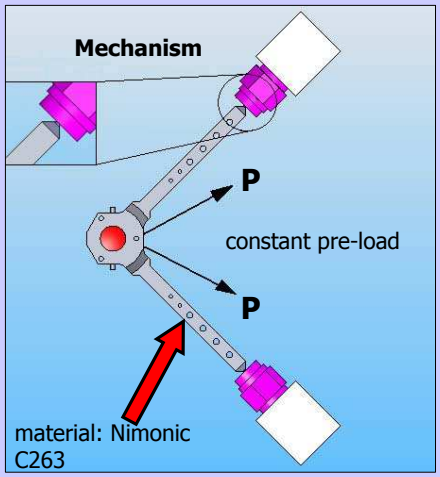
For high temperatures, there are additional requirements

- **Non-contact** and **localised** heating
- Measurement system **compatible** with the temperatures

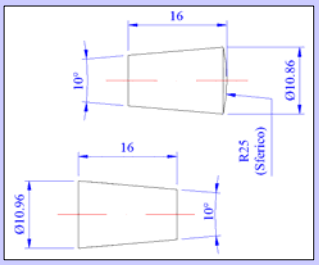
Contact mechanics - 8

the Pittsburgh-Polito 1D test rig – room temperature

Details of the rig



Conical Specimens



Contact mechanics - 9

data acquisition & control high temperature

2 Vibrometri laser Polytec single-point: controller OFV -5000, sensor head OFV-505

Differential laser Vibrometer Polytech: controller OFV-3001, sensor heads OFV-512: resolution 2nm, max displacement 82 mm.

Acquisiition SignalCalc Mobilyzer II, 32 channels, up to 8 sources
8 tachometer channels
120 - 150 dB dynamic range
49 kHz analysis bandwidth

Shaker Tira TV52122-M: force 220N, max acceleration 102 g, max displacement 25 mm, max frequency 5 kHz.

Induction heating machine: MTC-6, power 6 kW, operating frequency: 20 to 80 kHz.

Contact mechanics - 10

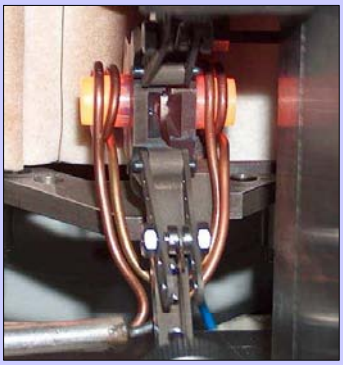
the Polito 1D test rig – high temperature

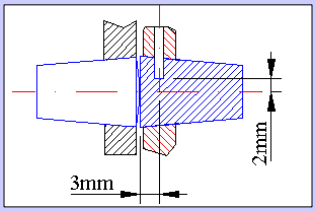
The heating system

- **Electromagnetic induction system:**
  - No-contact
  - Large power density
  - Easy control of temperature
  - Acceptable costs

  MTC-6 Induction Machine:
  - Nominal power: 6kW;
  - Working frequency: 20-80 kHz
- Temperature measurements: **k-type thermocouple**, placed near the contact
- Temperature control: **NI-card + Labview**
- **Error** on temperature measurements: estimated by FEM thermal analysis

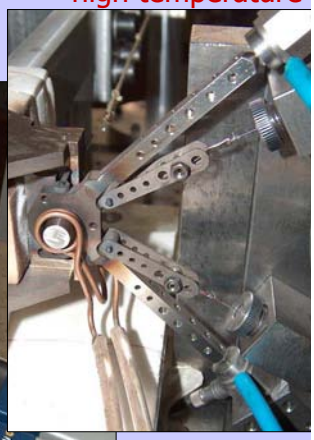
$$Temp_T = Temp_M \cdot (1 \pm 1{,}5\%) \pm 8°C$$





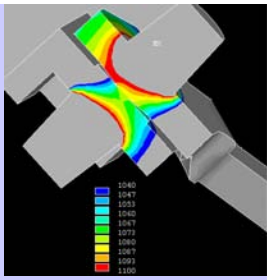
Contact mechanics - 11

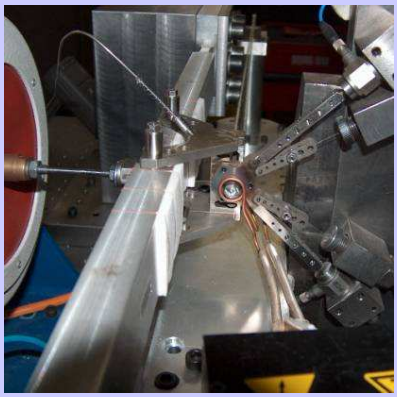
the Polito 1D test rig – high temperature

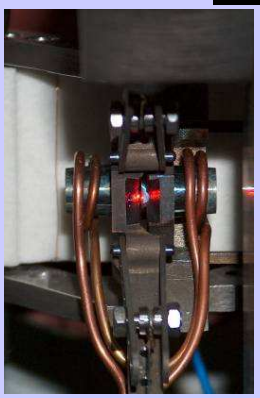


Contact mechanics - 12

the Polito 1D test rig
– high temperature







Contact mechanics - 13

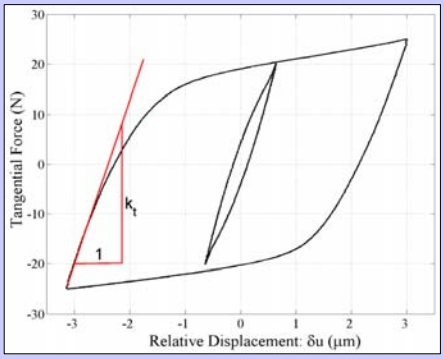
the Polito 1D test rig
– high temperature

Determination of contact parameters from hysteresis cycles

Dissipated energy:
⇒ It is the area of the hysteresis cycle.

Contact Stiffness:
⇒ Slope of the curve after reversal of motion

Friction Coefficient:
⇒ Since it varies in gross-slip phase, it is calculated with Mindlin's theory, in terms of dissipated energy, normal loading and contact stiffness.
⇒ Calculation gives the "average" value.
⇒ Only for gross-slip cycles



$$E = E_{microslip} + E_{gross-slip}$$

$$E = 4 \cdot (u \cdot \mu \cdot N) - \frac{24}{5} \cdot \frac{(\mu \cdot N)^2}{k_T}$$

Contact mechanics - 14

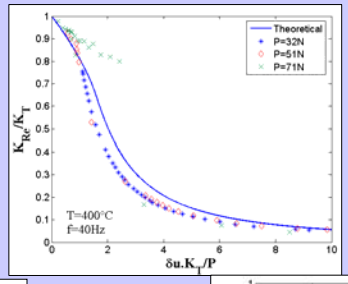
HBM real and imaginary parts

Behaviour of K_{Re} and K_{Im} of the cycles at high temperature

Extraction of K_{Re} and K_{Im} from real hysteresis cycles:

Characteristic length: $X_0 = \frac{\mu.P}{k_T}$

Dimensionless amplitude: $\tilde{X} = \frac{X}{X_0}$

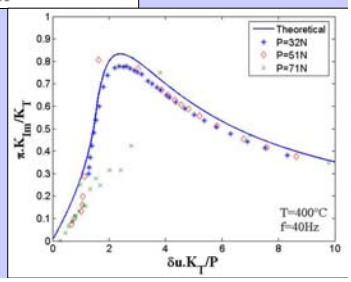


Dimensionless equiv. contact stiffness:

$$\tilde{k}_e(\tilde{X}) = K_{Re}(\tilde{X}) = \frac{1}{\pi \tilde{X} \mu P} \int_0^{2\pi} q \cos(\theta) d\theta$$

Hysteretic effective contact damping:

$$\tilde{c}_e(\tilde{X}) = K_{Im}(\tilde{X}) = \frac{1}{\tilde{X} \mu P} \int_0^{2\pi} q \,\mathrm{sen}(\theta) d\theta$$



Contact mechanics - 15

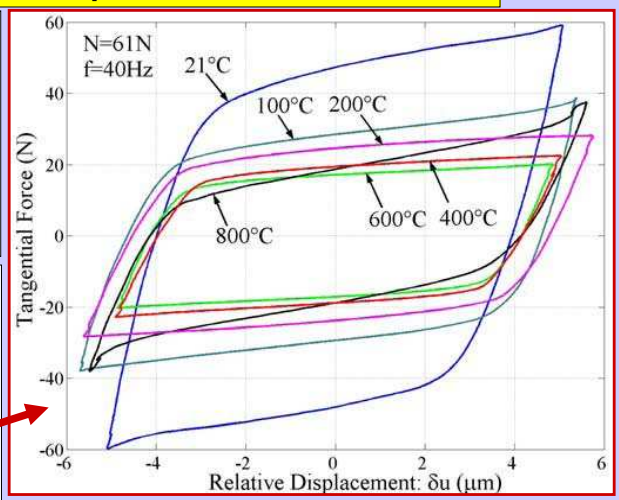
hysteresis cycles at high temperature

First set of experiments: specimens of Inconel 100

Experimental Procedure:
One couple of specimens for each normal load, measurements for increasing and decreasing temperatures

Performed tests:
- Microslip for N=32N
- Microslip for N=61N
- Gross-slip for N=32N
- Gross-slip for N=61N

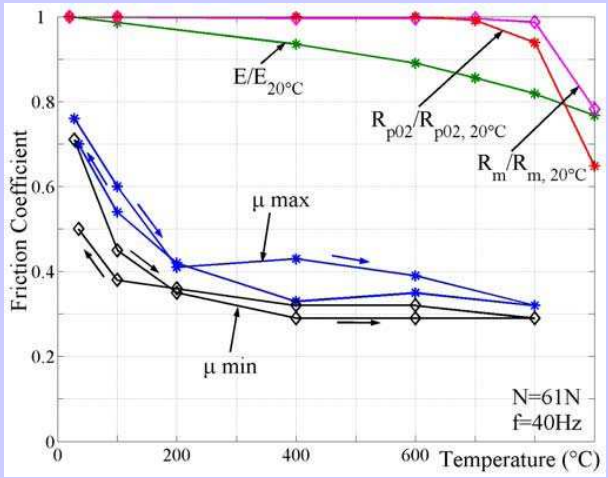
Temperature Range: T=20°C up to 800°C



Hysteresis cycles measured for different temperatures (Normal load: 61N)

Contact mechanics - 16

hysteresis cycles
at high temperature



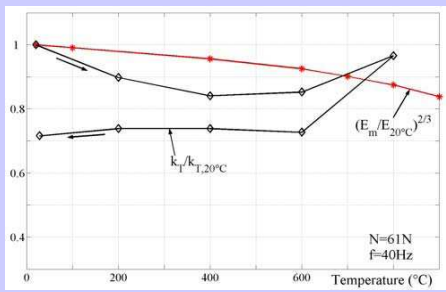
Friction coefficient & material parameters vs. temperature
(Normal load: 61N)

for each value of temperature, the maximum and the minimum values among the friction coefficients calculated from different measurements are reported in the figure

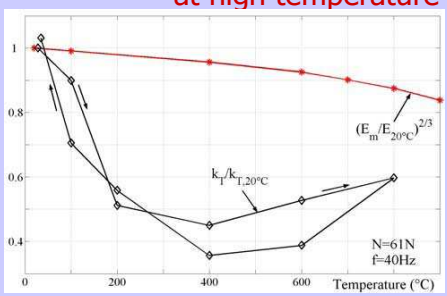
While the friction coefficient decreases rather sharply from room temperature up to 200°C and then becomes almost stable, yield and ultimate strength are practically constant up to almost 800°C and the Young modulus decreases almost linearly with temperature in the same temperature range. Therefore no simple relationship seems to exist between the friction coefficient and the basic material properties.

Contact mechanics - 18

hysteresis cycles
at high temperature



Tangential stiffness & Young modulus vs. temperature
(Normal load: 61N – micro-slip conditions)



Tangential stiffness & Young modulus vs. temperature
(Normal load: 61N – gross-slip conditions)

If the contact followed the Cattaneo-Mindlin model, the stiffness would be proportional to the Young modulus of the specimens raised to the power of 2/3.
The behavior of the Young's modulus with temperature is also reported in the diagrams. But it can be noted that the variation of Young's modulus with temperature does not explain the variation of contact stiffness, and so far no explanation has been found for the behavior of the stiffness with temperature.

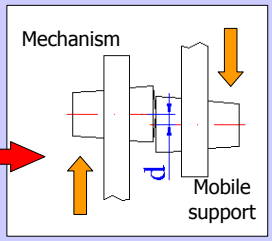
Contact mechanics - 17

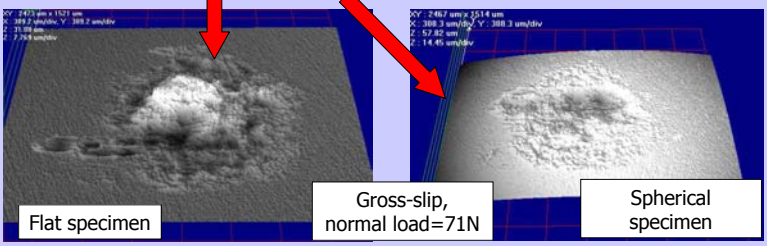
problems & developments

More problems found in the Test Procedure

- Thermal expansions => contact point moves with temperature
  - Expansion of the mechanism
  - Expansion of the mobile support
  - Bending of the vibrating beam ???

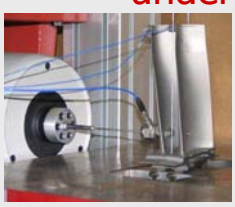
  20-800°C: d>1mm
- Wear changes the contact properties

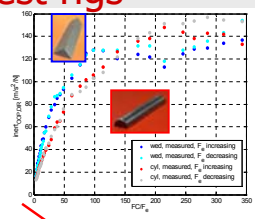




Materials under test now (2008/2009 – AVIO restricted access to data): RENE 77, 80, 108, 125; CMSX-4; Inconel 718 – with and without T800 coating (high roughness and hardness)

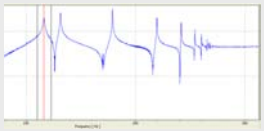
Modelling damping components -1 underplatform dampers – test rigs

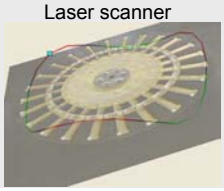




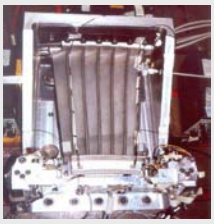
Laser scanner

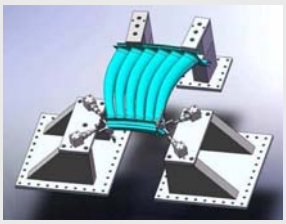
Non contact electromagnetic excitation

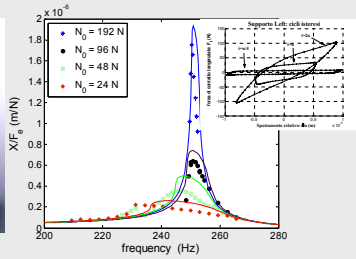


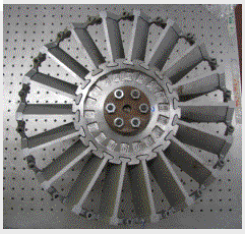


Modelling damping components -2
vane segments and shrouded blades – test rigs

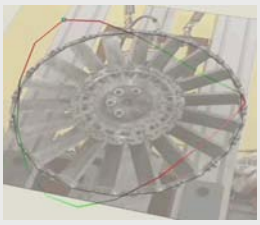








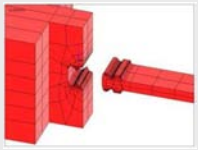
Laser scanner measurement

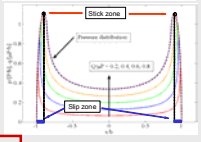


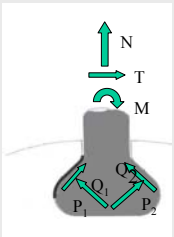
Modelling damping components -3
blade root damping characterization

The Test Rig

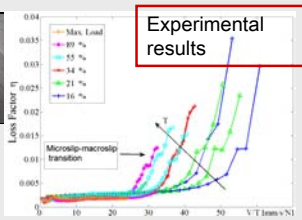
Numerical Contact model based on contact mechanics principles



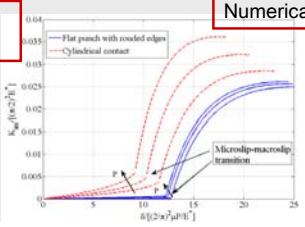




Experimental results

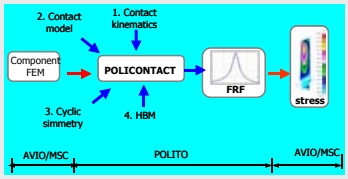


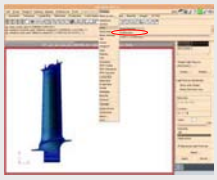
Numerical results



Dynamic response of turbine discs - 1

Non linear forced response calculation with friction damping POLI.contact software

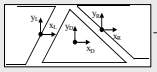




POLI.CONTACT in MSC.Patran and MSC.Nastran

underplatform damper

Rigid damper

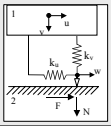


Kinematic assumptions

Elastic damper

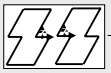
No kinematic assumptions

contact Model



shrouded blades and vane segments

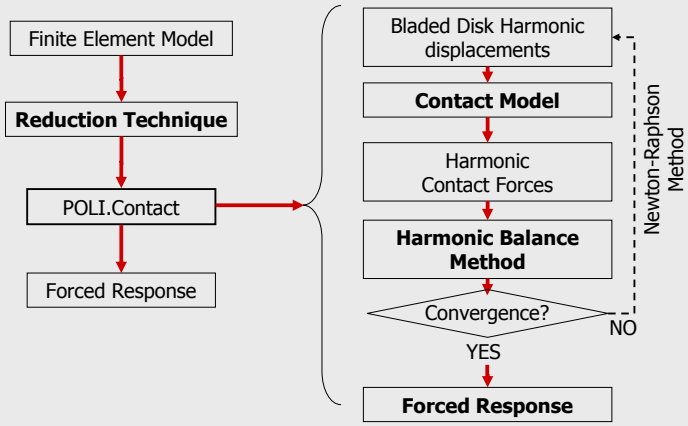
Direct blade-to-blade contact → simple contact kinematics



contact model

Dynamic response of turbine discs - 2

Flow-chart of the numerical code developed for the forced response calculation of bladed disks with shrouds.



Dynamic response of turbine discs - 3

Reduction technique to compute the forced response of frictionally damped bladed disks.

Reduction Matrix

N Physical dofs $\leftarrow \{X\} = [\Psi] \cdot \{q\} \rightarrow$ **n** generalized dofs

$N >> n$

$\longrightarrow$

Reduced model

$$[K_R] = [\Psi]^T \cdot [K] \cdot [\Psi]$$
$$[M_R] = [\Psi]^T \cdot [M] \cdot [\Psi]$$
$$\{F_R\} = [\Psi]^T \cdot \{F\}$$

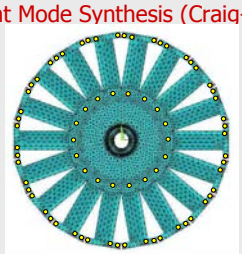
- Component Mode Synthesis (Craig-Bampton).

Physical dofs of master nodes

$$\{q\} = \begin{Bmatrix} X_M \\ \eta_S \end{Bmatrix}$$

Modal dofs of slave nodes

=



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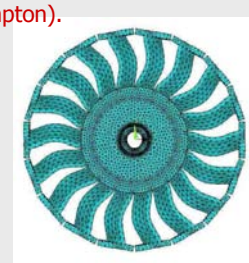
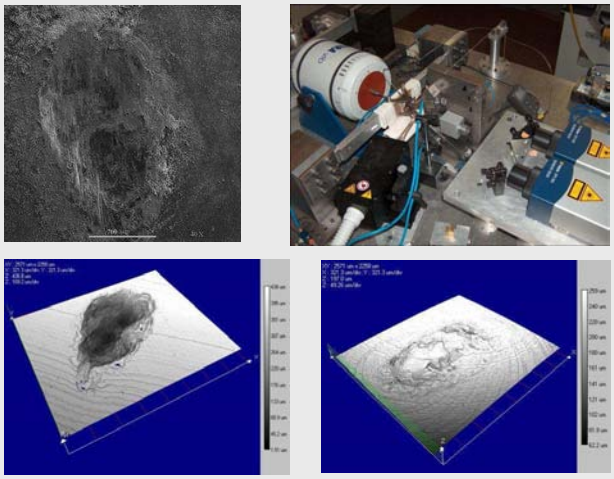


Complementary activities

- Tribology, wear measurement
- Spin-test rig (work in progress)
- Dynamics of turbine disks with asymmetrical effects (MISTUNING)
- Real time evaluation of temperature and thermal stresses at critical locations of turbine disc (disc lifing)
- X Ray evaluation of residual stress in turbine components

Tribology, wear measurement

Measurement of wear on contact surfaces
Validation of theoretical wear models



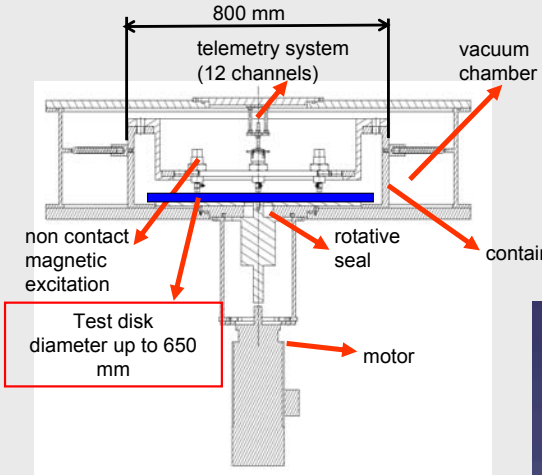
Point contact test

$10 \cdot 10^6$ cycles

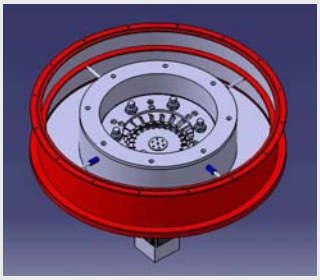
amplitude 30 μm

T = 900 °C

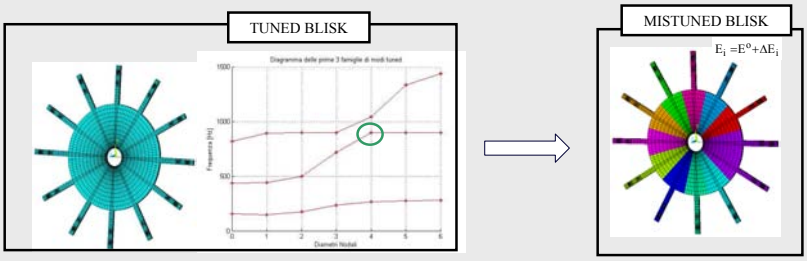
Spin-test rig (work in progress)



Rotation speed
up to 4000 rpm



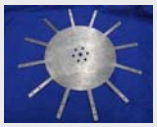
Dynamics of turbine disks with asymmetrical effects (MISTUNING)

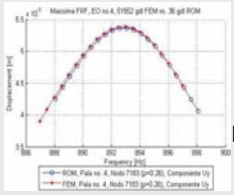


Identification model of mistunnig

Comparison of different reduction techniques and improvement

Experimemtal validation by means of dummy disks

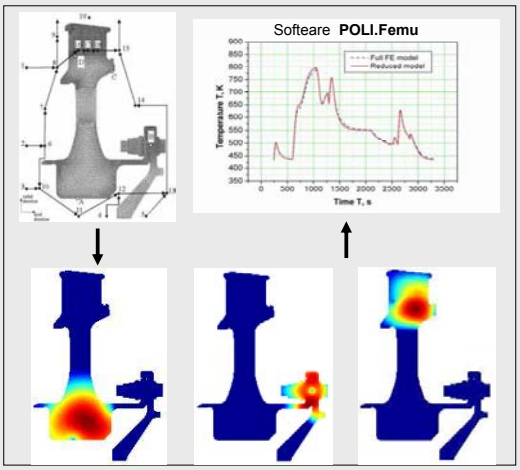




Magnification Factor: 1,42

Real time evaluation of temperature and thermal stresses at critical locations of turbine disc (disc lifing)

Original methodologies for temperature and thermal stress monitoring based on the modal reduction techniques and on the Green's function theory for both linear and non-linear applications



X Ray evaluation of residual stress in turbine components

Residual stress measurements by means of

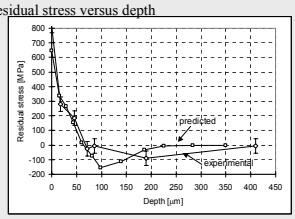
X-ray diffractometer

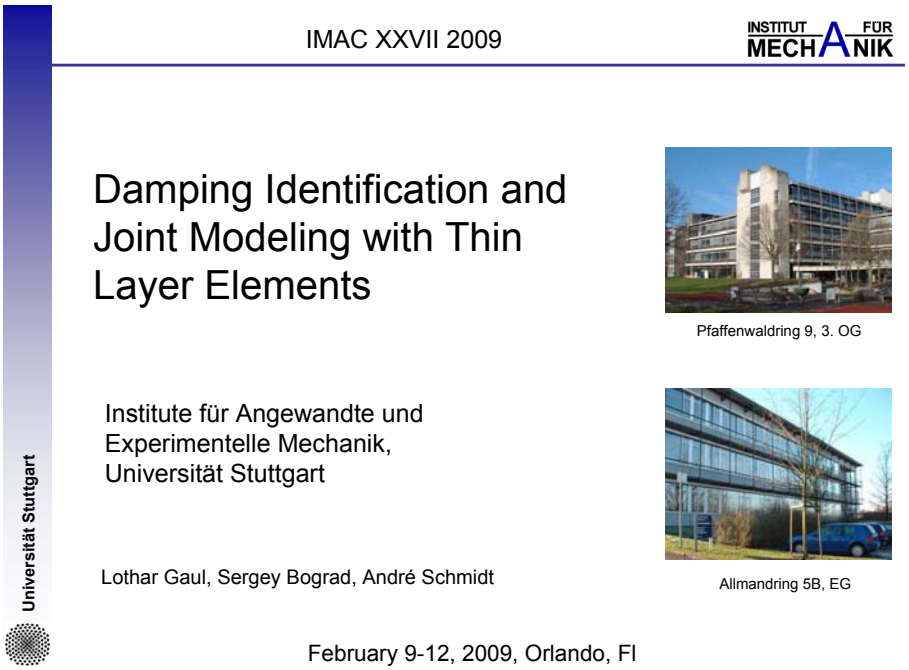
(Siemens D5005)

VERDI EU Project (6th FWP 2005-2008)

Validation of numerical model for mechanical working (milling and turning) simulation

Residual stress versus depth





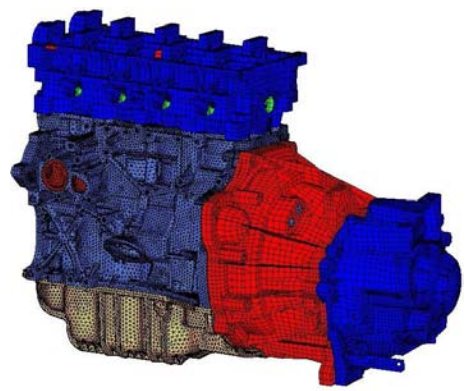
D.2.4 Slide Presentation of Lothar Gaul, Universitat Stuttgart: *Damping Identification and Joint Modeling with Thin Layer Elements*

Overview

- Motivation
- Joint damping parameters
- Test structure
- FE – model description
- Comparison between FE simulation and experiment

Motivation

- Prediction of damping in a structure before the prototype is available
- Estimation of a structure independent joint parameters
- Constant hysteretic damping
- Application of damping locally at the joint interface



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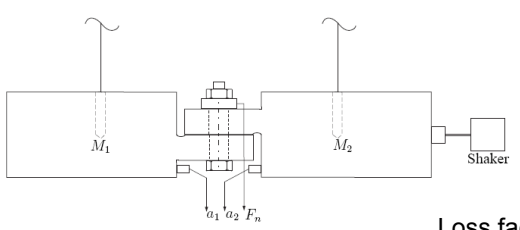
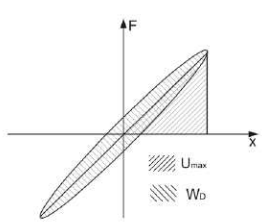
Joint patch damping – measurement set-up



$$\Delta x = \iint a_1 dtdt - \iint a_2 dtdt$$

$$F_t = M_1 a_1$$



Loss factor and stiffness determination from hysteresis diagram

$$\chi = \frac{W_D}{2\pi U_{\max}}$$

$$F_t = c\Delta x$$

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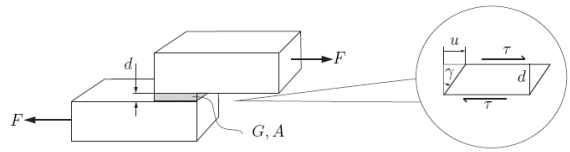
Stiffness of the generic joint

Calculation of shear modulus from the experiment



$$\tau = G\gamma \approx G\frac{u}{d}$$

$$\tau = \frac{F}{A}$$

Experimentally determined shear modulus

$F \approx$ $u =$ $u \Rightarrow G = \frac{cd}{A}$

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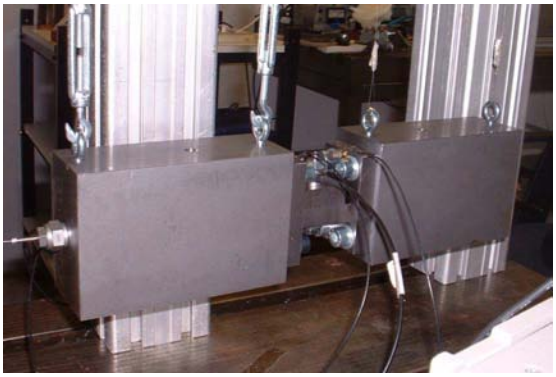
Joint patch damping – experiment

Interchangeable patch samples

Parameter estimation at different frequencies

Careful alignment of the masses is necessary in order to avoid bending in the joint



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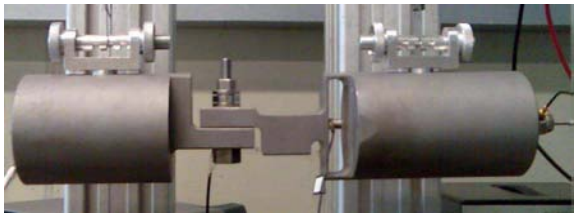
Joint patch damping – experiment with a leaf spring (resonator system)

Allows to achieve good excitation in axial direction; bending in the joint is reduced

Joint parameters can be measured only for one frequency



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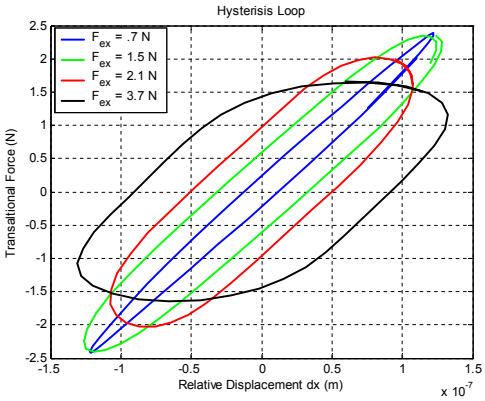
Joint patch damping – resonator system

Measurement of the hysteresis for small contact pressure
Contact pressure – 33 N/cm²

Macro and micro slip behavior

Varied stiffness and dissipation



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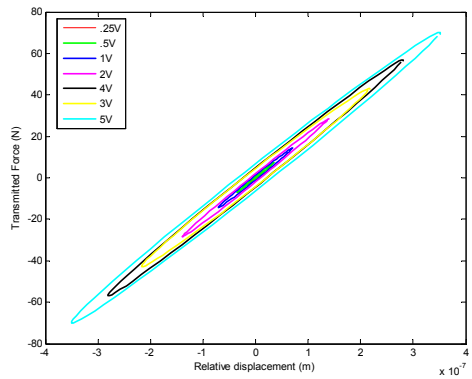
Joint patch damping

Measurement of the hysteresis for high contact pressure
Contact pressure – 1.2 kN/cm^2

No sliding occurs – only micro slip behavior

Constant stiffness and dissipation



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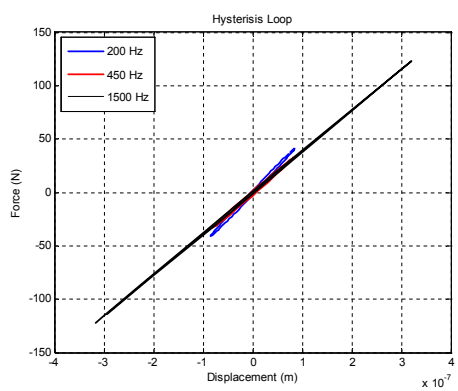
Joint patch damping

Measurement of Hysteresis at variable frequencies for high contact pressure
Contact pressure – 2 kN/cm^2

Stiffness and damping are nearly frequency independent in the measurment range

$$\chi \approx 0.06$$
$$c \approx 490\, kN/mm$$



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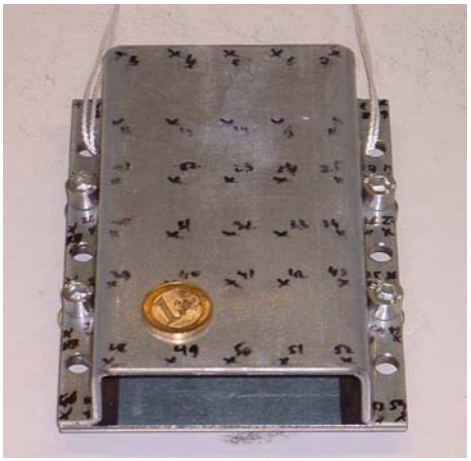
Experimental modal analysis – test structure 1

Mounting torque:
14 Nm

Roughness of the joint surface:
Rz 6.3

Boundary conditions:
free-free



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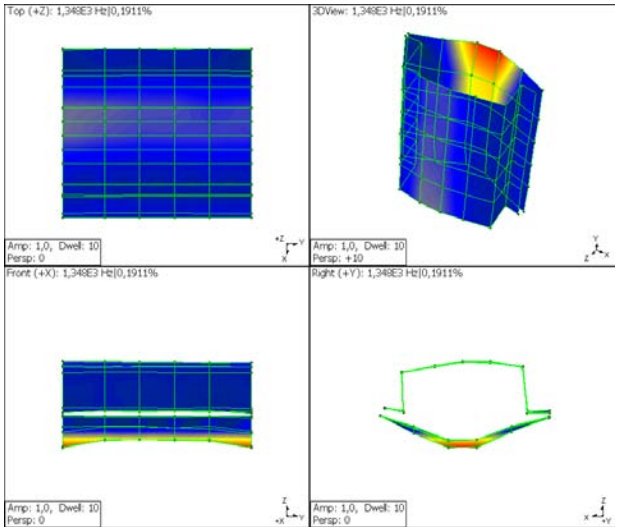
Experimental modal analysis – test structure

Mode with the highest measured damping



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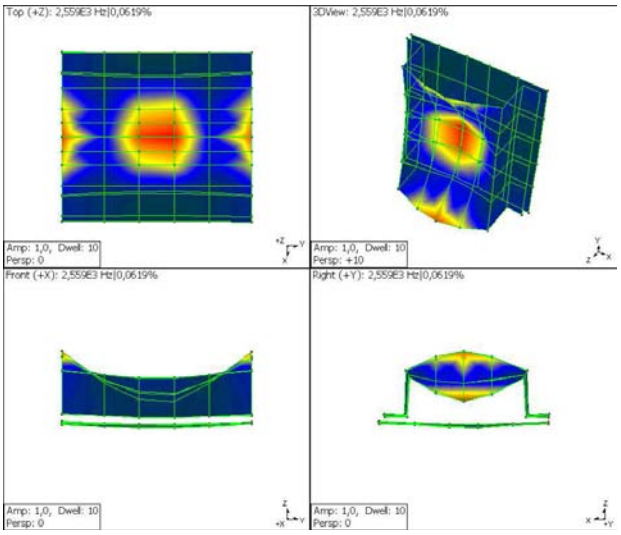
Experimental modal analysis – test structure

Mode with the lowest measured damping



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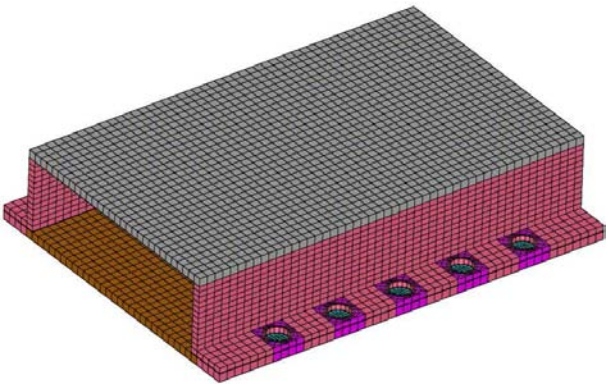
Implementation of the local damping modeling in the FE-simulation

Modeling of damping with the thin layer elements



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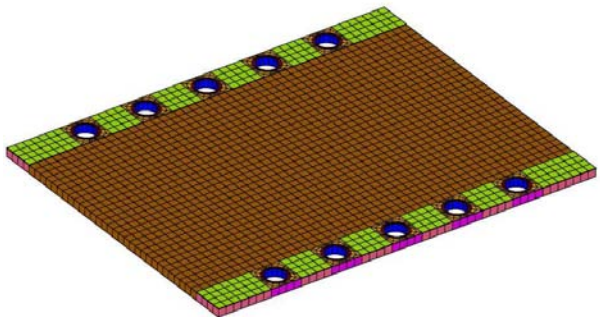
Implementation of the local damping modeling in the FE-simulation

Modeling of damping with the thin layer elements



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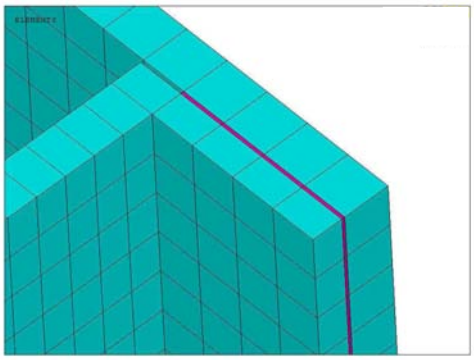
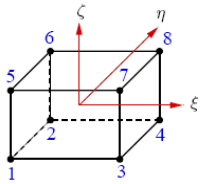
Implementation of the local damping modeling in the FE-simulation – thin layer elements





Brick or penta elements with up to 1:1000 thickness to length ratio

Implementation of the local damping modeling in the FE-simulation – orthotropic material behavior in the joint

MAT9 Solid Element Anisotropic Material Property Definition

$$\begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{zz} \\ \sigma_{xy} \\ \sigma_{yz} \\ \sigma_{zx} \end{bmatrix} = \begin{bmatrix} E_1 & E_{12} & E_{13} & 0 & 0 & 0 \\ & E_2 & E_{23} & 0 & 0 & 0 \\ & & E_3 & 0 & 0 & 0 \\ & & & E_4 & 0 & 0 \\ & & & & E_5 & 0 \\ & & & & & E_6 \end{bmatrix} \begin{bmatrix} \varepsilon_{xx} \\ \varepsilon_{yy} \\ \varepsilon_{zz} \\ \varepsilon_{xy} \\ \varepsilon_{yz} \\ \varepsilon_{zx} \end{bmatrix}$$

E_3 – Normal stiffness

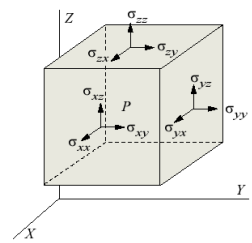
E_5, E_6 – Tangential stiffness

Other matrix elements are ignored

GE Structural element damping coefficient. (Real)

To obtain the damping coefficient GE, multiply the critical damping ratio C/C_0, by 2.0.

MSC.Nastran 2005, Quick Reference Guide



Nastran Material Parameter GE = Loss factor χ

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Implementation of the local damping modeling in the FE-simulation – comparison between experiment and simulation

Mode Nr	Experimental Freq (Hz)	Simulated Freq (Hz)	Difference (%)	Experimental Damping (%)	Simulated Damping (%)	Difference (%)
1	1063	1057	-0,5	0,110	0,107	-2,5
2	1348	1339	-0,7	0,191	0,204	6,9
3	1441	1406	-2,4	0,107	0,114	7,1
4	1558	1567	0,6	0,147	0,178	21,6
5	2149	2155	0,3	0,143	0,179	25,1
6	2307	2244	-2,7	0,077	0,072	-6,1
7	2447	2428	-0,8	0,086	0,065	-24,9
8	2559	2531	-1,1	0,062	0,026	-58,0
9	3372	3363	-0,3	0,116	0,110	-5,3
10	3713	3742	0,8	0,076	0,009	-87,7

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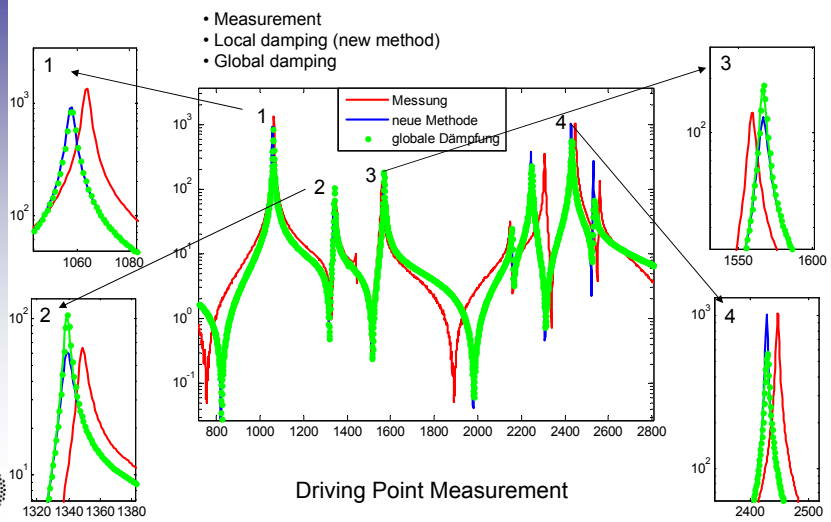
Implementation of the local damping modeling in the FE-simulation – comparison between experiment and simulation



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Simulation of Cylinder Block with Oilpan

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Cylinder Block – Oilpan

Meshing of the contact surfaces with conformed FE-Mesh

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Implementation of the local damping modeling in the FE-simulation – comparison between experiment and simulation

Mode Nr	Experimental Freq (Hz)	Simulated Freq (Hz)	Difference (%)	Experimental Damping (%)	Simulated Damping (%)	Difference (%)
1	1011	1008	-0,29	0,157	0,152	-3,2
2	1287	1317	**2,34**	0,214	0,117	-45,4
3	1305	1276	**-2,26**	0,049	0,060	22,1
4	1399	1403	0,25	0,143	0,130	-9,0
5	1558	1574	1,02	0,197	0,068	**-65,4**
6	1667	1674	0,45	0,191	0,099	-48,3
7	1849	1859	0,56	0,258	0,110	**-57,3**
8	1874	1900	1,38	0,196	0,083	**-57,8**
9	1910	1953	**2,26**	0,116	0,063	-45,5
10	1998	2059	**3,09**	0,174	0,125	-27,8
11	2052	2058	0,33	0,094	0,096	1,3
12	2226	2211	-0,65	0,096	0,126	30,3
13	2320	2300	-0,89	0,200	0,169	-15,5
14	2389	2415	1,08	0,198	0,127	-36,1
15	2493	2476	-0,67	0,128	0,091	-28,8

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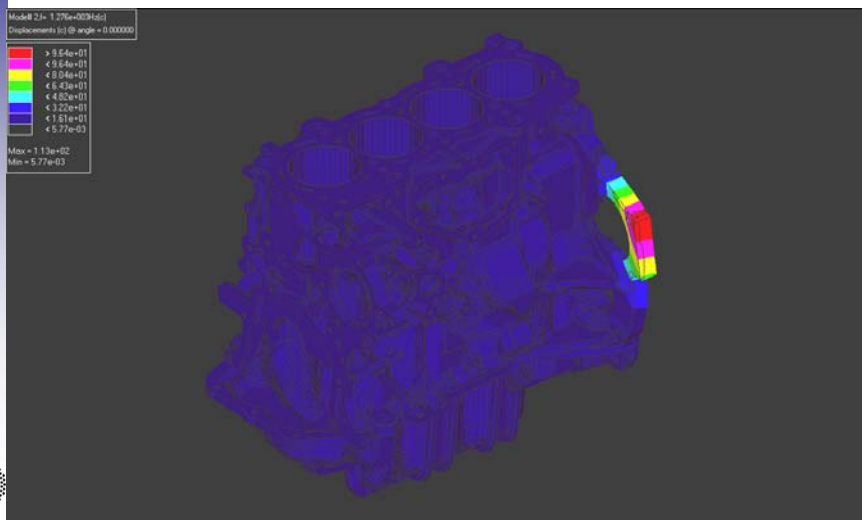
Simulation of Cylinder Block with Oilpan



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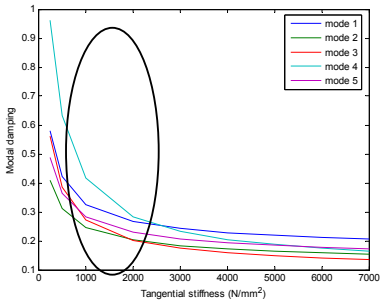
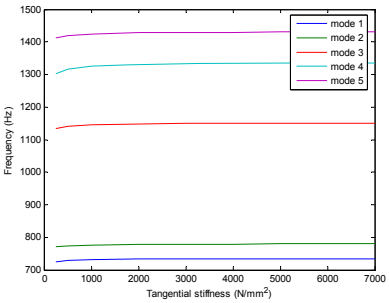
Sensitivity analysis

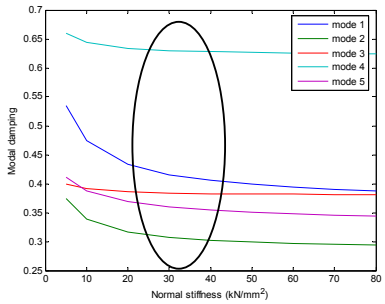
Sensitivity of the damping and eigenfrequencies due to the changes in the tangential stiffness of the thin layer elements

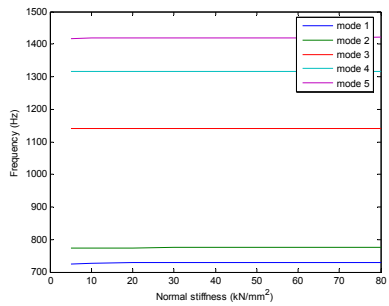




Sensitivity analysis

Sensitivity of the damping and eigenfrequencies due to the changes in the normal stiffness of the thin layer elements





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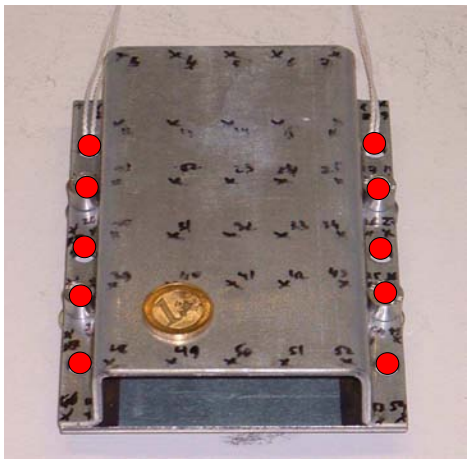
Experimental Modal Analysis of the structure with variable number of bolts

Three measurements

10 bolts

6 bolts

4 bolts



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Experimental Modal Analysis of the structure with variable number of bolts

Bolts #	10		6			4		
Mode Nr.	Freq (Hz)	Damping (%)	Freq (Hz)	Damping (%)	Difference (% Damping)	Freq (Hz)	Damping (%)	Difference (% Damping)
1	1063	0,1099	1060	0,127	16	1030	0,219	99
2	1348	0,1911	1320	0,266	39	1100	1,69	784
3	1441	0,1066	1430	0,147	38	1260	0,691	548
4	1558	0,1466	1520	0,189	29	1380	0,167	14
5	2149	0,1428	2100	0,17	19	1800	1,53	971
6	2307	0,0766	2320	0,0966	26	2280	0,341	345
7	2447	0,0863	2450	0,0974	13	2410	0,138	60
8	2559	0,0619	2550	0,0653	5	2550	0,161	160
9	3372	0,1162	3300	0,137	18	2680	0,644	454

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Conclusions

- Joint patch damping shows only small frequency dependence, which allows the use of the constant hysteresis method
- FE-simulation with the thin layer elements containing orthotropic material properties shows good correlation with experimental results
- Method works for the joints with regularly distributed contact pressure; objective classification of the pressure distribution in the joints and applicability of the method should be investigated

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Experiments and Modeling at the Microscale

Andreas A. Polycarpou
Microtribodynamics Laboratory
Mechanical Science & Engineering
University of Illinois at Urbana-Champaign

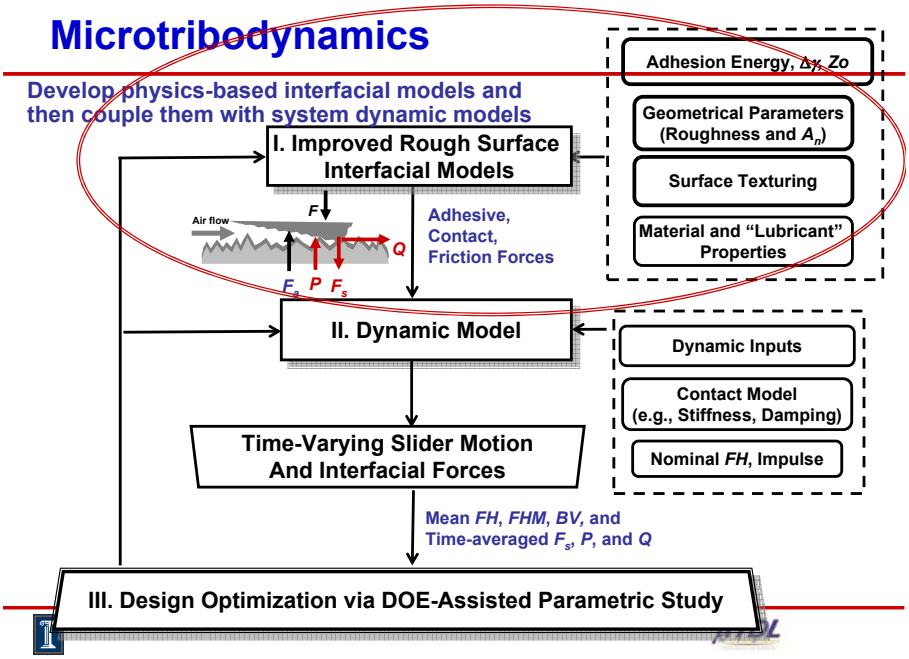
Acknowledgements
National Science Foundation-CMMI

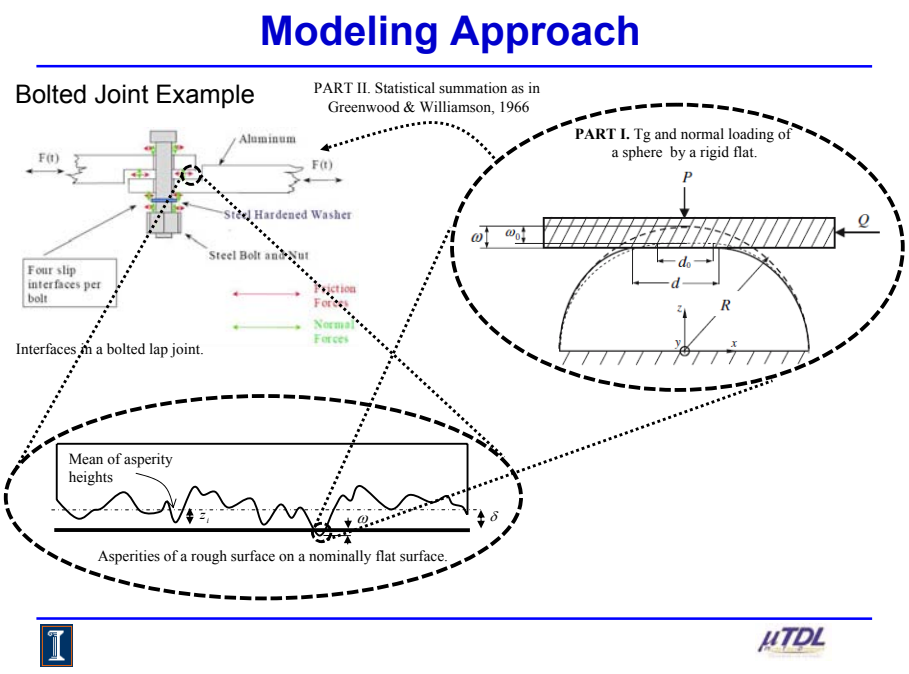
2nd Workshop on Joints Modeling, Dartington, UK, April 27-29, 2009

D.2.5 Slide Presentation of Andreas Polycarpou, University of Illinois at Urbana-Champaign: *Experiments and Modeling at the Microscale*

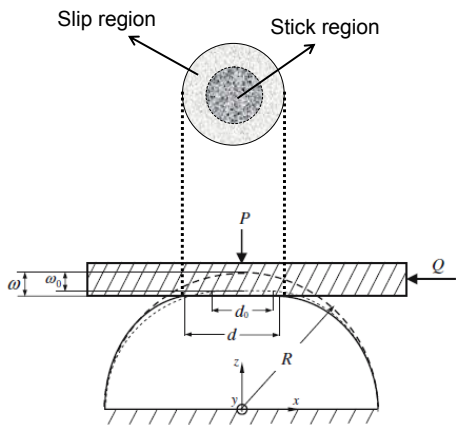
Outline

- Continuum modeling approach for contacts with friction from "sub-micron" to the "cm" length scales
- The significance of roughness (measurement and statistical modeling)
- Successful predictions of normal contact stiffness and friction
- Unsuccessful predictions - Challenges
- Microslip/partial slip experimental studies
- Summary/Challenges


Microtribodynamics
Develop physics-based interfacial models and then couple them with system dynamic models
I. Improved Rough Surface Interfacial Models
Adhesion Energy, Δγ, Zo
Geometrical Parameters (Roughness and A_n)
Surface Texturing
Material and "Lubricant" Properties
Air flow
F
Q
F_s P F_s
Adhesive, Contact, Friction Forces
II. Dynamic Model
Dynamic Inputs
Contact Model (e.g., Stiffness, Damping)
Nominal FH, Impulse
Time-Varying Slider Motion And Interfacial Forces
Mean FH, FHM, BV, and Time-averaged F_s, P, and Q
III. Design Optimization via DOE-Assisted Parametric Study


Modeling Approach
Bolted Joint Example
PART II. Statistical summation as in Greenwood & Williamson, 1966
PART I. Tg and normal loading of a sphere by a rigid flat.
Aluminum
F(t)
F(t)
Steel Hardened Washer
Steel Bolt and Nut
Four slip interfaces per bolt
Friction Forces
Normal Forces
Interfaces in a bolted lap joint.
P
Q
ω
ω_0
d_0
d
R
z
y
x
Mean of asperity heights
z_i
ω_i
δ
Asperities of a rough surface on a nominally flat surface.
μTDL

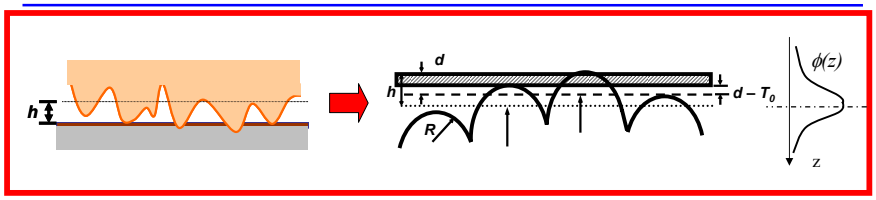
Single Asperity



- Deformable sphere of radius R in contact with a rigid flat
- Loading starts with normal preload (clamping force), P
- Tangential load, Q is applied while keeping P constant
- Interference due to preload, ω_0 < interference after tg. loading, ω
- Contact diameter due to preload, d_0 < contact diameter after tg. loading, d (junction growth)
- Contact region contains both stick and slip regions before gross sliding occurs (partial slip)

With assumptions one can do, full slip, full stick, elastic plastic

Rough Surface Statistical Model



$$F_s = f(\underbrace{h, \sigma, R, \eta, \phi, A_n, T_0}_{\text{Micro contact Roughness/Geometrical}}, \underbrace{E, H, \nu, \Delta\gamma}_{\text{Material}})$$

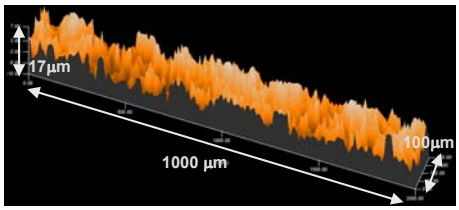
$$P, Q = f(\underbrace{h, \sigma, R, \eta, \phi, A_n}_{\text{Micro contact Roughness/Geometrical}}, \underbrace{E, H, \nu}_{\text{Material}})$$

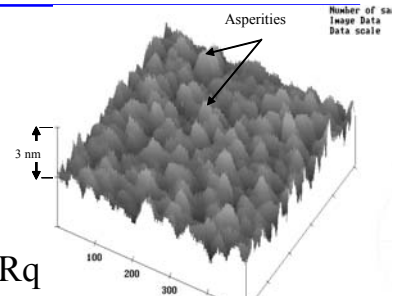
Despite the known limitations of the GW model (e.g., scale dependence of some of its parameters, asperities act independently, constant R etc, it gives good results in some engineering situations

Note that in this work, models include elastic/plastic contact, may have asymmetric asperity distribution and may contain a trace of thin lubricant on the surface

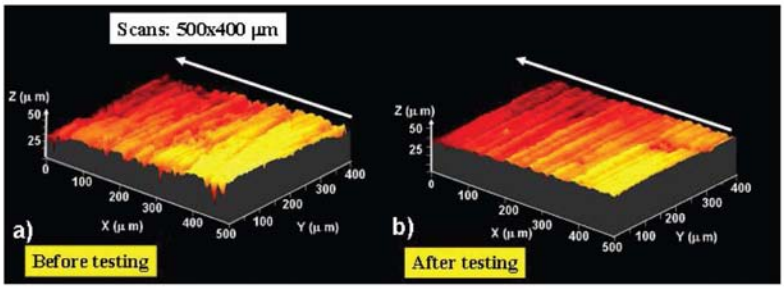
Do asperities exist?

Surface Roughness at Different Length Scales



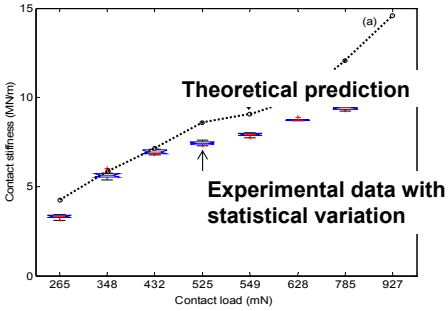


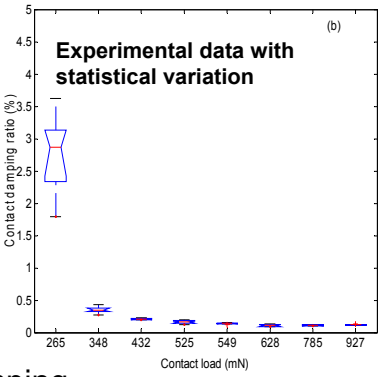
Joint-type surfaces of about 1 µm Rq



Comparisons with Experiments

	Individual surface parameters					Combined interface parameters			
	E (GPa)	H (GPa)	R_q (μm)	R_s (μm)	$\eta(/\mu m^2)$	E^* (GPa)	R_s (μm)	$\eta(/\mu m^2)$	R_q (μm)
Sample 1	192.92	2.96	0.167	3.555	0.122	105.3	2.402	0.1256	0.189
Sample 2	192.92	2.96	0.088	7.409	0.144				



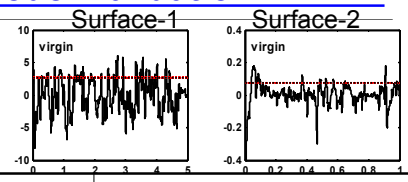


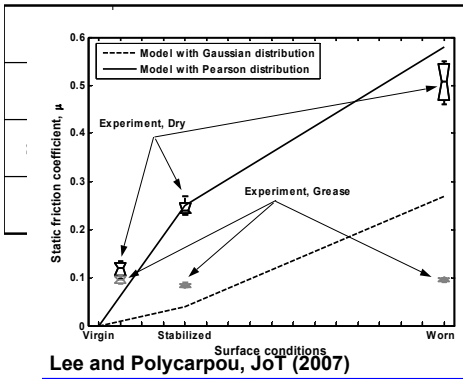
Normal contact stiffness and damping

Shi and Polycarpou, JVA, 2004

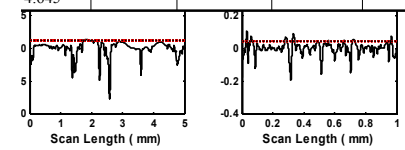
Material, Roughness Parameters (Physical Parameters) and Model Validation

Parts	Type	Young's modulus (GPa)
Surface-1	4121-H Steel	206.85
Surface-2	52100 Steel	206.85





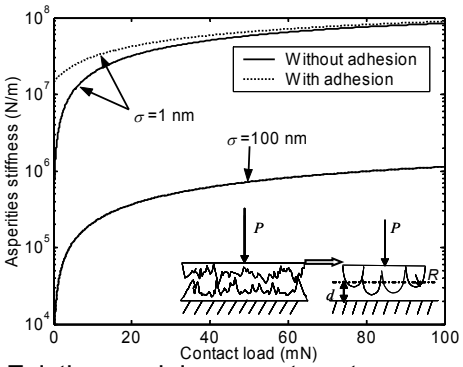
Combined interface parameters				
(/ μ m^2)	R (μ m)	η (/ μm^2)	R_q (μm)	Ψ
0.373 / 12.213	37.49	0.500	2.661	4.55
0.143 / 4.794	97.54	0.290	1.771	2.30
0.151 / 4.645	156.11	0.335	1.241	1.52



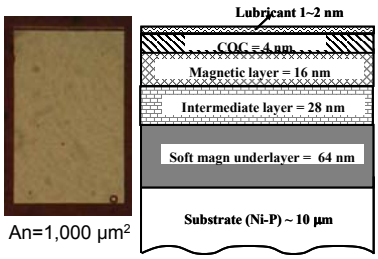
Lee and Polycarpou, JoT (2007)

Model Validation (no surface layers, asperity interactions)

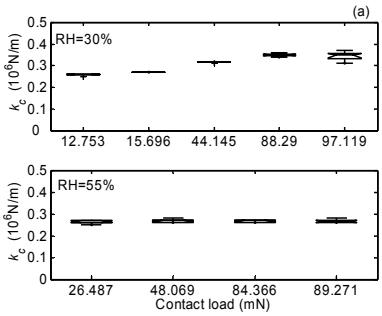
More Comparisons with Experiments



Existing models can not capture the much lower contact stiffness values, possibly due to (1) substrate deformation, (2) asperity interactions, (3) finite nominal area effects and uncertainty



An=1,000 μm²

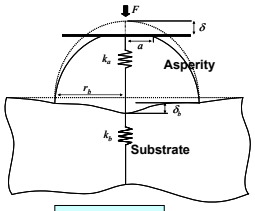


Shi and Polycarpou, JoT, 2008

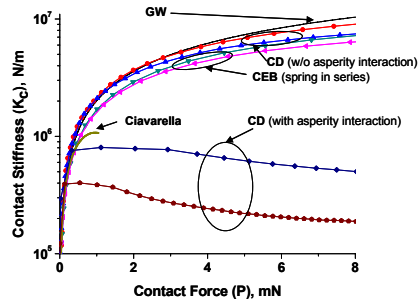
Improved Contact Models-Needed

• Improved contact models to account for the **effects of bulk substrate** and **asperity interactions**

Single asperity



Hertz theory



- Asperity stiffness $k_a(\delta) = 2\sqrt{R}E_a\delta_a^{1/2}$
- Bulk stiffness $k_b = \frac{4}{3}r_bE_b$
- Combined effective stiffness

$$k(\delta) = \left[\frac{1}{k_a(\delta)} + \frac{1}{k_b}\right]^{-1}$$

- Contact force of asperities

$$F_{ci} = \frac{4}{3}E^*R^{1/2}h_i^{3/2} \qquad (h_i = z_{si} - d - u_{zi})$$

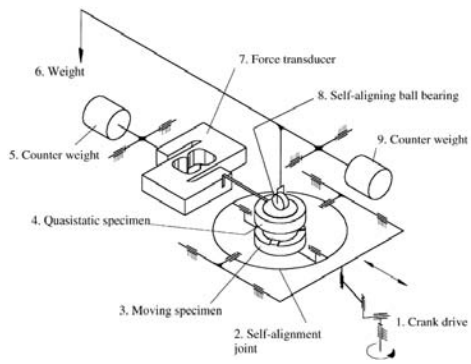
- Displacement of the neighboring asperities (Johnson, 1985)

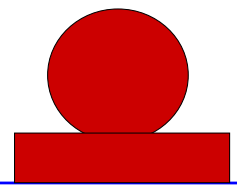
$$U_z = \frac{p_0}{2r_bE^*}\left[\left(2r_b^2 - r^2\right)sin^{-1}\left(r_b/r\right) + r^2\left(r_b/r\right)\sqrt{\left(1 - r_b^2/r^2\right)}\right]$$

Having shown that the contact and static friction models (with simplifying assumptions) work well in several cases, can we apply them to the case of Joints?

Spherical Elastic Plastic Model with Slip

- Experiments: Varenberg et al.2004
- 5 mm diameter steel ball is fretted on a flat steel specimen
- Each surface has rms roughness of 40-50 nm (reasonable to assume Hertzian spherical contact)
- Oscillation frequency 16.5 Hz
- 2 sets of experiments with loads: 23, 35 N and imposed tangential displacements of 10 and 1.5 µm
- Only physical parameters input to the model: hardness or yield strength (of the softer material), Young's modulus, Poisson's ratio and radius of the sphere [i.e., no friction coefficient]

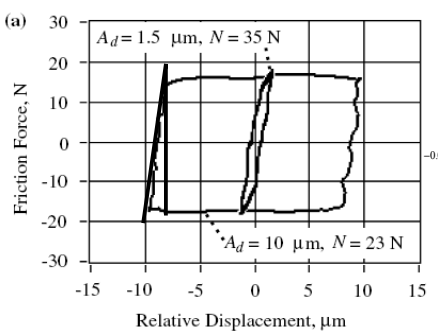


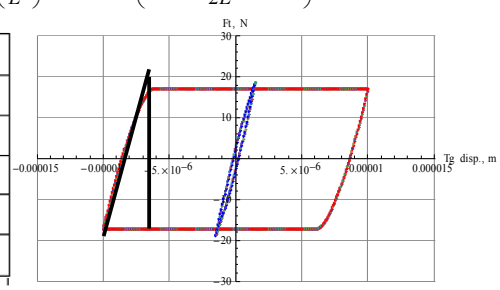


Spherical Elastic Plastic Model with Slip

• Kogut & Etsion contact model applied to partial slip
• Transition from elastic to elastic-plastic regime is designated by the critical normal load and interference values

$$P_c = \frac{(\pi(0.454+0.41\nu))^3}{6}\left(\frac{H}{E^*}\right)^2 HR^2, \omega_c = \left(\frac{\pi(0.454+0.41\nu)H}{2E^*}\right)^2 R$$





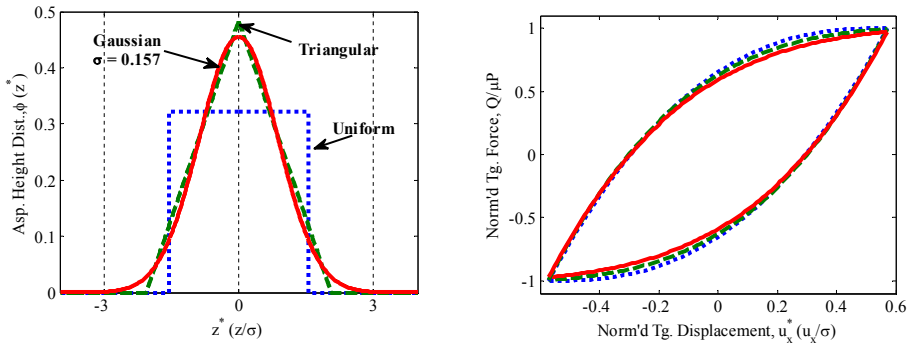
Stiffness 20 (test) Vs 12 MN/m (mod)

Magnitude of friction is captured well but not the stiffness

Roughness Model with Partial Slip

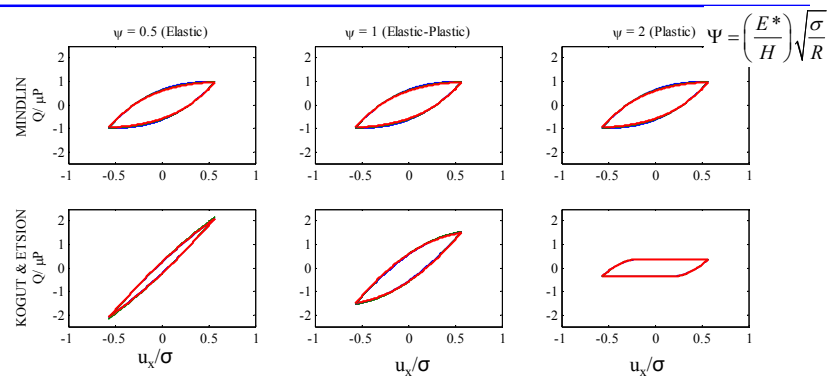
- Mindlin constant friction (μ=0.3) + Hertzian normal loading + Mindlin tg. Loading/unloading/reloading + GW statistical summation = Björklund (1997)
- Doesn't account for plasticity
- Type of asperity height dist. has almost no effect on microslip behavior (interesting and "opposite" to the gross sliding friction predictions)

No direct experimental comparison, ongoing



Based on physical parameters PLUS Friction Coefficient

Going Forward: Nominally Flat Contact



Friction loop predictions with only physical parameters of material properties, surface roughness parameters, and nominal contact area

Need to compare with experiments

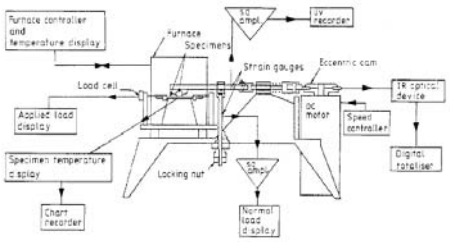
There are some model assumptions that could be challenged e.g., tangential loading doesn't affect the tractions obtained by frictionless normal loading

Asperity interactions, film effects, substrate effects, temperature effects etc

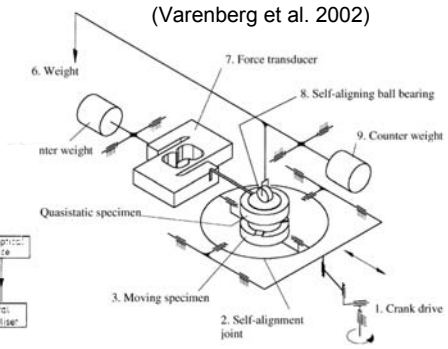
Experiments (Partial Slip)

Fretting parameters:
- Normal loading
- Slip magnitude
- Fretting freq. & duration
- Joint conditions + lubrication

(Kuno et al. 1988)

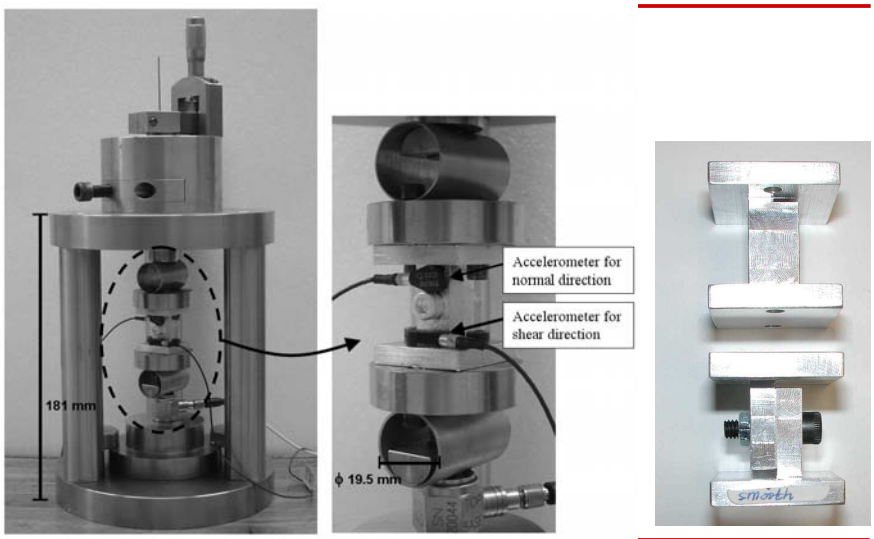


(Varenberg et al. 2002)



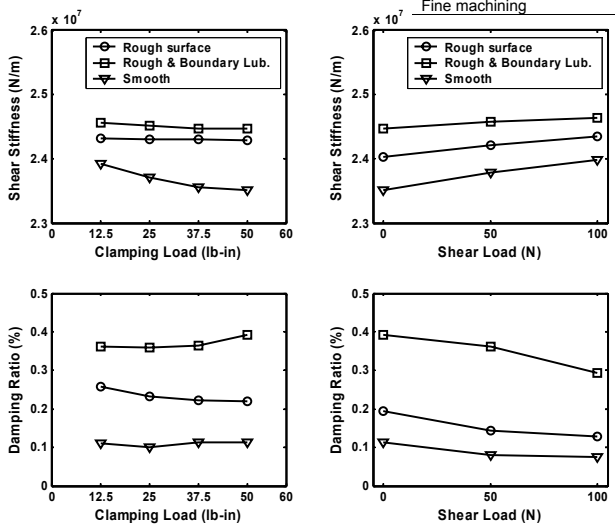
Testing machines use the following actuation mechanisms
- Servo-hydraulic → High stress fretting studies
- Electromagnetic → Flexibility in slip amplitude and frequency
- Piezoelectric → Displacements of small amplitude and high frequency (better for micro-slip)
- Rotational to linear motion mechanical devices (DC motor, eccentric cam, crank drive, etc.) → Easy to build and robust.

Stiffness and Damping of Shear Lap Joints



Experimental Results

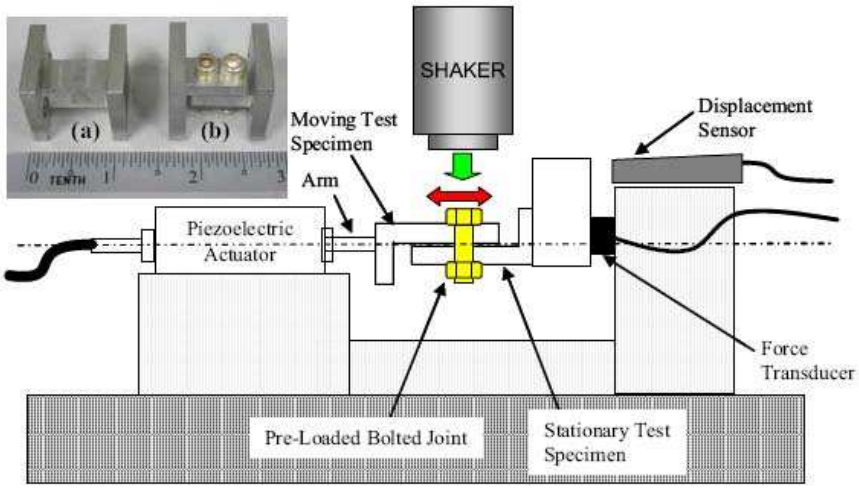
	E (GPa)	H (GPa)	R_a (μm)	R_q (μm)
Sample 1 (Smooth) Superfinishing	72	1.15	0.061	0.081
Sample 2 (Rough) Fine machining	72	1.15	1.154	1.628



No friction loop measurements possible with this set up

Need to investigate the coupling between normal and shear interfacial and dynamic interactions

"Rigid" Partial Slip Tester for Joints



Summary/Challenges

Micro scale parameters can be measured and used in continuum-based interfacial models to predict contact and friction, including friction loops encountered in joints

Some challenges: (1) Contact mechanics assumptions in the analysis; (2) identifying their range of applicability; (3) Improvements in the contact mechanics and roughness models; (4) correlation of testing methodologies and results

Experiments and Modeling at Mesoscale

April 27th, 2009

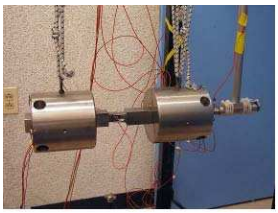
Sarah K. Leming
Sandia National Laboratories
Albuquerque, NM, USA
sleming@sandia.gov

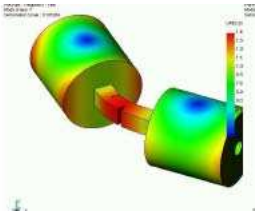
Sandia is a multiprogram laboratory operated by Sandia Corporation, a Lockheed Martin Company, for the United States Department of Energy's National Nuclear Security Administration under contract DE-AC04-94AL85000.

D.2.6 Slide Presentation of Sarah Leming, Sandia National Laboratories: *Experiments and Modeling at Meso-scale*

Ultimate Goal

- **To develop predictive simulations of full-up systems, including energy dissipating mechanisms, such as bolted joints**
  - Meaningful predictions of system response to realistic loads
  - “Before-built” predictions to guide design decisions
  - Predict response and margin to un-testable environments





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Ultimate Goal Cont.

- Discovery and validation experiments
  - Give intuition to the modeler
- Long-term: Compile a database of experimental data to use in future model development and validation
  - Make use of experimental data and theories to explain that data.
  - Primary focus thus far: Interfaces with bolted joints
    - Majority of work with single joints in shear

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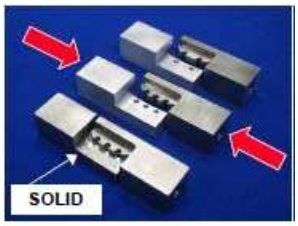
Where Do We Go From Here?

- Many energy-dissipating mechanisms
  - Bolted, threaded, tape joints…
  - Foam
  - Interfaces
- Considered the major contributors to response of systems
  - Determined that bolted joints were a major contributor
  - Expand our understanding in more complex geometries and environments
  - Also determined that "top-down" or "hybrid" approaches may lead to more predictive models



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How Do We Get There?

- **To gain insight into the behavior of real bolted joints in full-up systems during realistic environments:**
  - **Variety of inputs**
    - **Complex shock waveform, random vibration**
  - **Multiple joints**
    - **Bolt rings, flange joints, etc.**
  - **Combination Loading**
    - **Past work has focused on joints in shear. Real loading includes bending, etc.**
  - **Explore "top-down" and "hybrid" approaches**
    - **Empirical approaches to subsystem damping**
  - **Variability and Uncertainty**

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Predictive Capabilities

- **Repeatability and Variability Experiments (Mike Starr)**
  - A phenomenologically-based analysis to capture variability intrinsic to the assembled lap joint structure due to interface surface features and interface misalignment.
  - Examine sample-to-sample variation and assembly-to-assembly variation of the same sample
  - Experimental results will be compared to computational ones (Adagio FE model)
  - Random and shock inputs will also be included for future model validation

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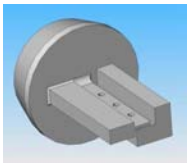
Predictive Capabilities

- **Non-dimensional relationship for predictive modeling (Mike Starr)**
  - Non-dimensional parameter study indicates a relationship between ratios of key joint characteristics (i.e. cyclic load amplitude/clamping load) and dissipation
  - Tailored variability study to consider the cyclic load/clamping load ratio
  - Will expand to vary thickness and contact area of the lap specimens
  - If verified experimentally, would develop set of key parameters to populate a model for unique joints without the need for extensive experimentation or simulation

Multiple Joints/Shock Loading

- **Bolt Rings, Multi-Joint (Mike Jew)**
  - Study dissipation in shock
  - Several different configurations,
    - Multiple bolts in one lap joint (Tab)
    - Multiple single lap joints in one assembly (“layer cake”)
    - Bolt ring configuration (24 bolts around circumference
    - Different materials-steel, aluminum
    - Top-down modeling



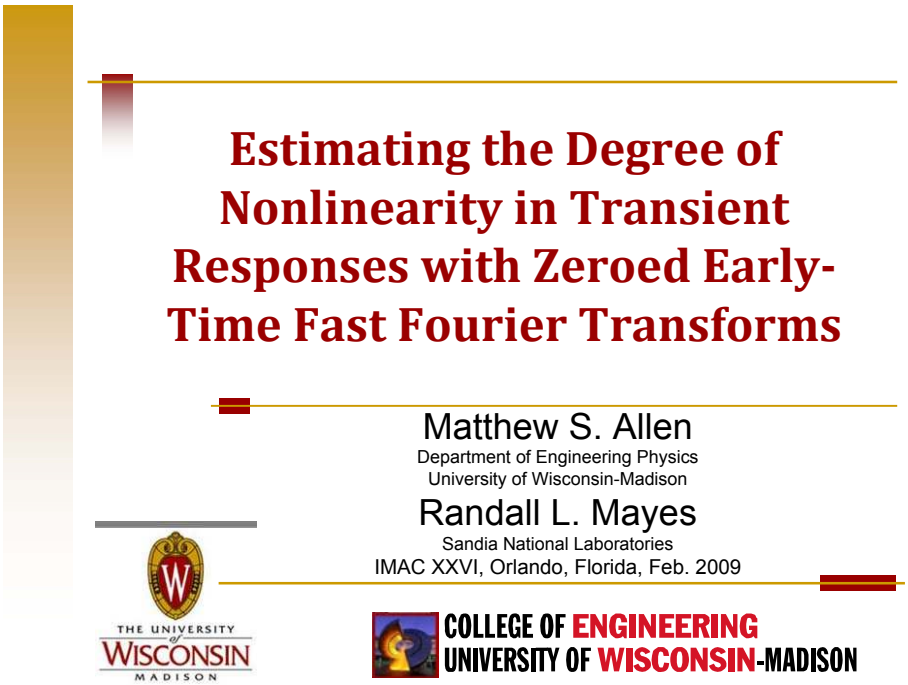
Tab (serial)

Layer Cake

Combination Loading

- **Previously, the majority of work has focused on joints in shear**
  - Actual environments include more complex loading (bending, torsion, etc.)
  - Choosing from the following
    - Big Mass Device with offset mass to provide moment with axial shear
    - Big Mass Device with a static moment from a cable
    - 6-DOF system to provide combined loads
    - Use piezo-washers to provide time-varying bolt loads to lap joint

Sandia National Laboratories



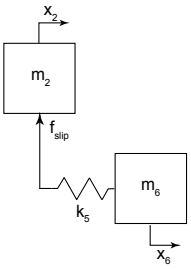
D.2.7 Slide Presentation of Randall L. Mayes, Sandia National Laboratories: *Estimating the Degree of Nonlinearity in Transient Responses with Zeroed Early Time Fast Fourier Transforms*

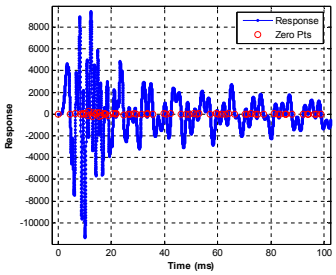
Introduction

- Challenge
  - Detect and characterize short duration nonlinearity in transient response data from relatively high order systems.
- Proposed Tools and Theory
  - Zeroed Early-time FFT (ZEFFT)
  - Backwards Extrapolation for Nonlinearity Detection (BEND)

Challenge

- Some systems with bolted joints respond to impulses nonlinearly in the first few cycles followed by a linear decay.

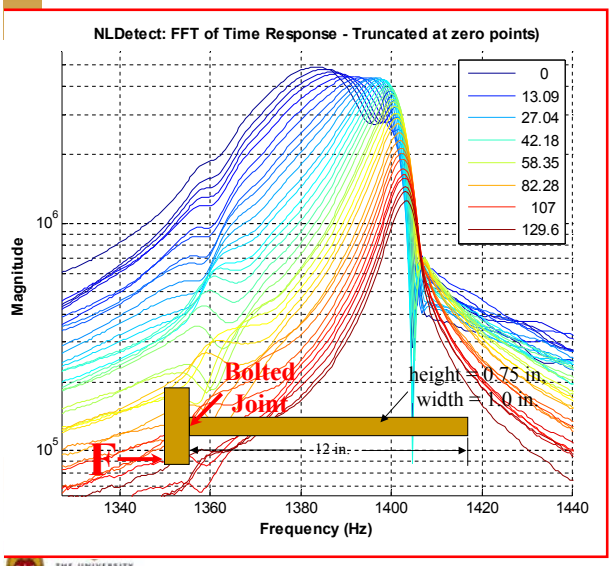




Challenge (2)

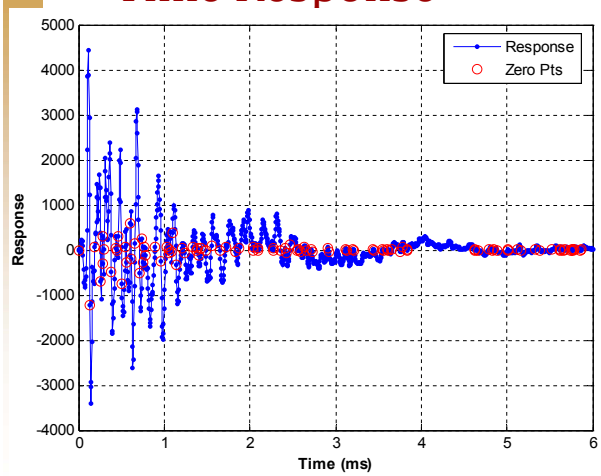
- Full nonlinear system identification of structures with joints is extremely difficult.
  - Moderate to High order systems are currently beyond the reach of state of the art nonlinear system identification algorithms
  - Time-frequency methods may not have sufficient resolution for the responses of interest. (i.e. Spectrogram, Wavelet, Choi-Williams, Hilbert-Huang…)
- What can we do then?

Zeroed Early-time FFT



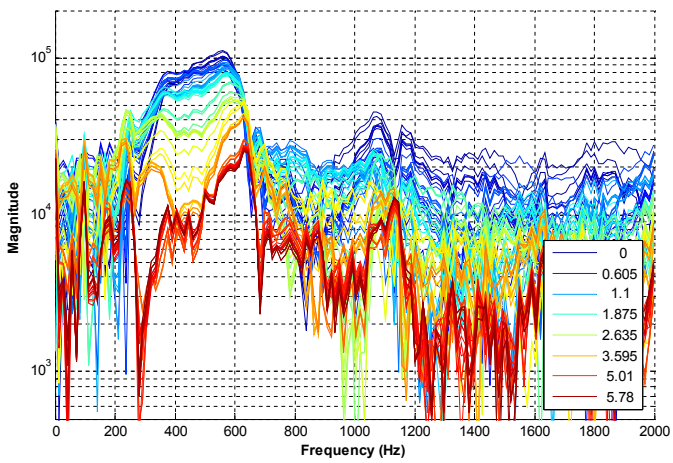
- Nonlinearity is assumed to be active at high amplitudes and inactive at lower amplitudes
- The response then becomes more linear as more of the initial nonlinear response is nullified.
- Impulse responses with initial segments of varying length set to zero are compared in the frequency domain.
- The nonzero portion of each impulse response begins at a point in which the response is near zero.

Experimental Shock Data: Time Response



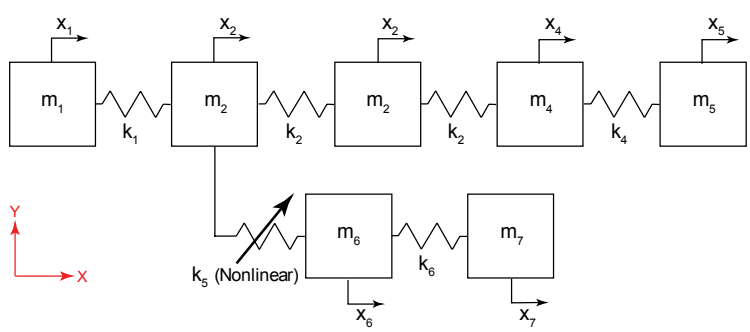
- Response of a complex structure with a bolted joint to an impulsive force.

Shock Data: ZEFFTs



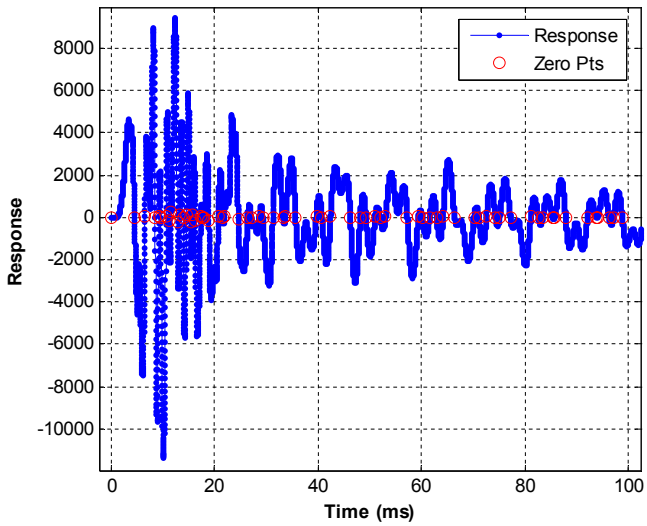
- Response contains a large, broad peak between 375 and 600 Hz at early times.
- Response appears to be very noisy above 800 Hz. This could be due in part to harmonics of the lower frequency modes.

Analytical Example



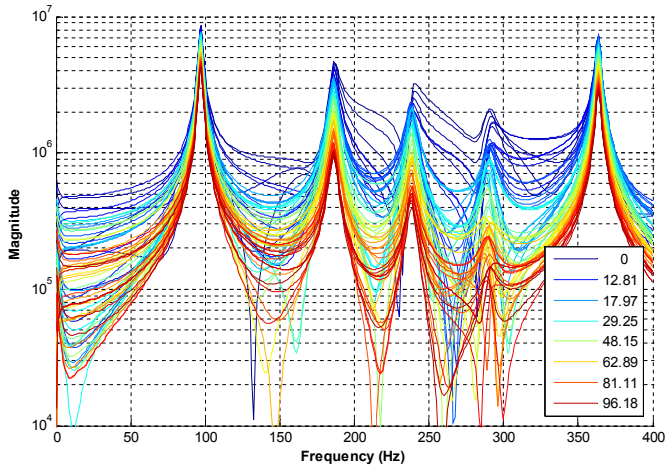
- 7-DOF system above used to test the methods.

Example #1 – Slip: Time Response



- Zero crossings in time response identified.
- It is difficult to discern if the system is nonlinear by simply inspecting the acceleration time history.

Example #1 – Slip: ZEFFTs



- ZEFFTs show extra frequency content between 180 and 350 Hz for zero times less than 15 ms.

Truncated Analytical Impulse Response

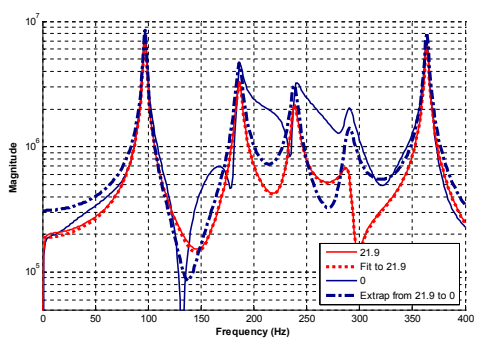
- Free response of an LTI system in Frequency Domain:

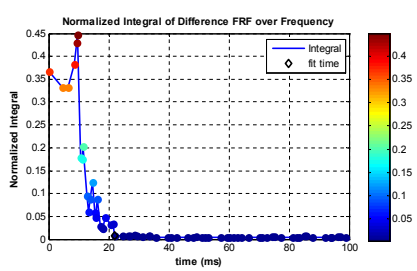
$$H(\omega)=\boxed{\sum_{r=1}^{2N}\frac{A_r}{i\omega-\lambda_r}}=\sum_{r=1}^{N}\left(\frac{A_r}{i\omega-\lambda_r}+\frac{A_r^*}{i\omega-\lambda_r^*}\right)=\sum_{r=1}^{N}\left(\frac{i\omega B_1+B_0}{\omega_r^2-\omega^2+2i\zeta_r\omega_r\omega}\right)$$

$$H_{t_z}(\omega)=\boxed{\sum_{r=1}^{2N}\frac{A_r e^{\lambda_r t_z}}{i\omega-\lambda_r}e^{i\omega t_z}}$$

Free response truncated at time t_z has the same form as the impulse response except for the $e^{i\omega t}$ factor.

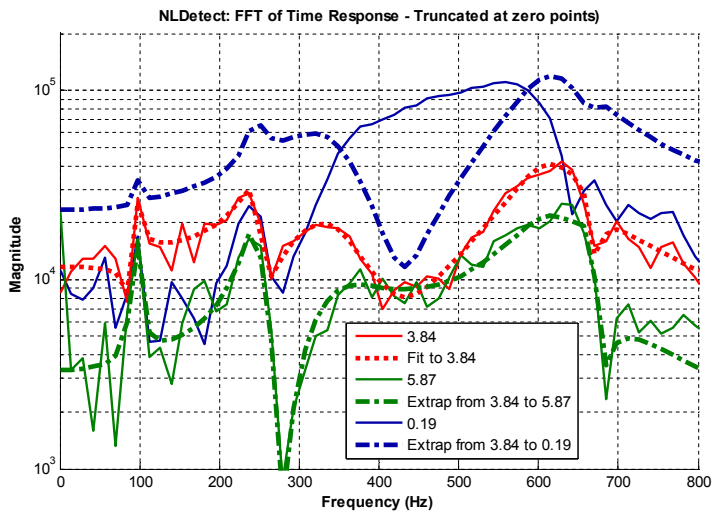
Example #1 – Slip: BEND & IBEND





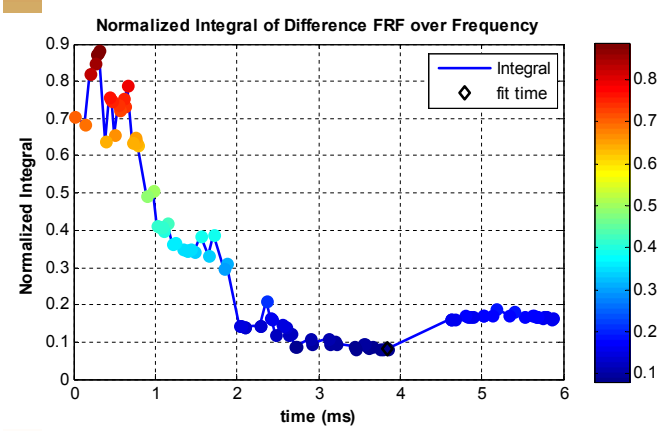
- Response at 21.9 ms was curve fit using AMI algorithm. Agreement is excellent at 21.9 ms and at all later times.
- Backwards extrapolation of 21.9 ms response to time 0 does not agree well suggesting that the system behaves nonlinearly some time before 21.9 ms.
- IBEND suggests that the system is linear for t > 10-15ms.

Shock Data: BEND



- Fit the response at 3.84ms and extrapolated backwards to 0.19ms and forwards to 5.87ms.
- Forward extrapolation agrees with the data about as well as the fit suggesting that the curve fit is accurate.
- Backward extrapolation does not show the broad peak between 375 and 600 Hz, suggesting that this is indeed due to nonlinearity.

Shock Data: IBEND



- IBEND suggests that the system behaves linearly after 2ms.
- The agreement is not perfect from 4-6ms.
- This may be due to the increasing relative importance of measurement noise.

Conclusions

- Zeroed Early-time FFTs (ZEFFTs) and Backwards Extrapolation (BEND) provide insight into the response of a nonlinear system to shock loading.
- BEND and IBEND can be used to provide quantitative information and to develop insight.
  - Care must be taken when interpreting the results of linear system identification.
  - Even if SYSID fails, direct inspection of the ZEFFTs may still yield useful information.

Multiscale Modeling of Interfaces

Arif Masud
Department of Civil and Environmental Engineering
University of Illinois at Urbana-Champaign

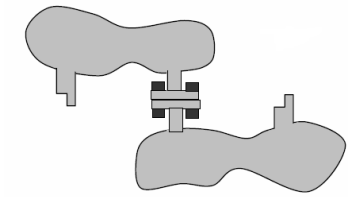
Second Workshop on Joints Modeling,
Dartington, April 27-29, 2009

D.2.8 Slide Presentation of Arif Masud, University of Illinois at Urbana-Champaign: *Multi-scale Modeling of Interfaces*

Outline

1. Conclusions from NSF-Sandia Workshop, October 2006
   - Merging the top-down and bottom-up approaches
2. Issues involved in the Modeling of Joint Interfaces
3. Modeling of Multiscale Structural Response
4. Merging CG and DG Methods for Joint Interfaces
5. Error Estimation

Merging the Top-Down and Bottom-Up Approaches (NSF-Sandia Conf. Oct 2006)



Segalman *et al. Sandia Report*

1. Consider the interface as an entity that can be represented at various scales through different types of "constitutive relations".
2. Multiscale Material Modeling: Hierarchical models that have micro / nano information built in.
3. Multiscale Variational Frameworks: Sub-scale modeling concepts.

Technical Issues involved in Multiscale Modeling of Joints

Material models with inherent scale effects

Geometric modeling of joint interfaces induce scale effects, both linear and nonlinear

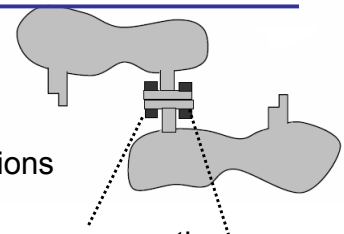
Response functions with embedded scale effects

Sensitivity with respect to the variation in the values of the parameters

Predictive capability for the modeling of multiscale effects requires multiscale error estimators

Heterogeneous multiscale phenomena with different PDE's appearing at different scales

Geometric Description of the Joint Interface



Consider the body as one domain

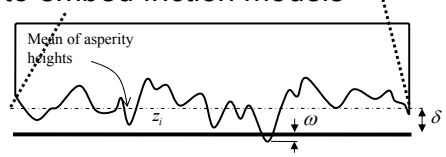
Employ DG ideas ⇨ discontinuous functions

Physical fields may or may not be continuous across the strong discontinuity ⇨ relative slip is permitted

Flux terms weakly enforce the continuity of the fields

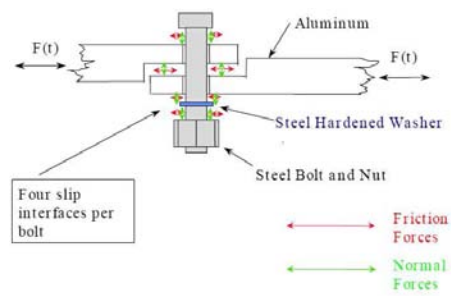
Flux terms provide a mechanism to embed friction models

Schematic diagram of interface:
Can be defined at various levels



Asperities of a rough surface on a nominally flat surface.

Modeling of Multiscale Structural Response



- Model Problem

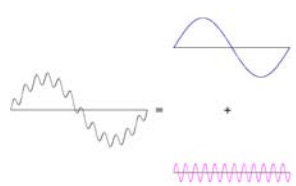
$$\mathcal{L}\boldsymbol{u} = \boldsymbol{f}$$

- Weak Form

$$(\boldsymbol{w}, \mathcal{L}\,\boldsymbol{u}) = (\boldsymbol{w}, \boldsymbol{f})$$

- Scale Decomposition

$$\boldsymbol{u} = \overline{\boldsymbol{u}} + \boldsymbol{u}' \qquad \boldsymbol{w} = \overline{\boldsymbol{w}} + \boldsymbol{w}'$$



- Multiscale Weak Form

$$(\overline{\boldsymbol{w}} + \boldsymbol{w}', \mathcal{L}\,(\overline{\boldsymbol{u}} + \boldsymbol{u}')) = (\overline{\boldsymbol{w}} + \boldsymbol{w}', \boldsymbol{f})$$

Multiscale Structural Response (cont'd)

- Coarse Scale Problem

$$(\bar{\boldsymbol{w}}, \mathcal{L}(\bar{\boldsymbol{u}} + \boldsymbol{u}')) = (\bar{\boldsymbol{w}}, \boldsymbol{f})$$

- Fine Scale Problem

$$(\boldsymbol{w}', \mathcal{L}(\bar{\boldsymbol{u}} + \boldsymbol{u}')) = (\boldsymbol{w}', \boldsymbol{f})$$

- Fine Scale Solution

$$\boldsymbol{u}' = -\boldsymbol{\tau}\boldsymbol{r} \qquad \text{where} \qquad \boldsymbol{r} = \mathcal{L}\bar{\boldsymbol{u}} - \boldsymbol{f}$$

- Modified Coarse Problem

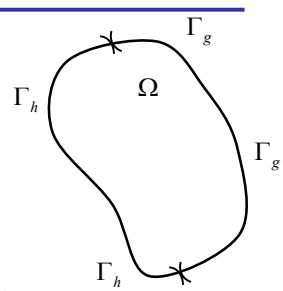
$$(\bar{\boldsymbol{w}}, \mathcal{L}\bar{\boldsymbol{u}}) + (\mathcal{L}^*\bar{\boldsymbol{w}}, \boldsymbol{\tau}\mathcal{L}\bar{\boldsymbol{u}}) = (\bar{\boldsymbol{w}}, \boldsymbol{f}) + (\mathcal{L}^*\bar{\boldsymbol{w}}, \boldsymbol{\tau}\boldsymbol{f})$$

Continuous and Discontinuous Galerkin Methods

- Standard "Continuous" Weak From

$$\int_\Omega [\mathrm{div}\boldsymbol{\sigma} + \boldsymbol{f}] \cdot \boldsymbol{w}\, d\Omega + \int_{\Gamma_h} [\boldsymbol{h} - \boldsymbol{\sigma}\boldsymbol{n}] \cdot \boldsymbol{w}\, d\Gamma = 0$$

$$\sum_e \int_{\Omega^e} \left[-\boldsymbol{\sigma} \cdot \nabla \boldsymbol{w}^h + \boldsymbol{f} \cdot \boldsymbol{w}^h\right] d\Omega + \sum_e \int_{\Gamma_e} \boldsymbol{\sigma}\boldsymbol{n} \cdot \boldsymbol{w}^h d\Gamma$$
$$+ \sum_e \int_{\Gamma_h \cap \Gamma_e} [\boldsymbol{h} - \boldsymbol{\sigma}\boldsymbol{n}] \cdot \boldsymbol{w}^h d\Gamma = 0$$

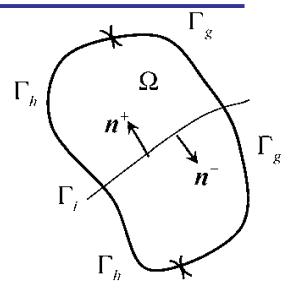


Continuous and Discontinuous Galerkin Methods

- Standard "Continuous" Weak From

$$\int_\Omega \left[\operatorname{div}\boldsymbol{\sigma} + \boldsymbol{f}\right]\cdot \boldsymbol{w}\, d\Omega + \int_{\Gamma_h}\left[\boldsymbol{h} - \boldsymbol{\sigma n}\right]\cdot \boldsymbol{w}\, d\Gamma = 0$$

$$\sum_e \int_{\Omega^e}\left[-\boldsymbol{\sigma}\cdot\nabla \boldsymbol{w}^h + \boldsymbol{f}\cdot \boldsymbol{w}^h\right] d\Omega + \sum_e \int_{\Gamma_e} \boldsymbol{\sigma n}\cdot \boldsymbol{w}^h d\Gamma$$
$$+ \sum_e \int_{\Gamma_h \cap \Gamma_e}\left[\boldsymbol{h} - \boldsymbol{\sigma n}\right]\cdot \boldsymbol{w}^h d\Gamma = 0$$



$\Gamma_{eh} = \Gamma_e \cap \Gamma_h$
$\Gamma_{ei} = \Gamma_e \cap \Gamma_i$
$\Gamma_{eh} \cap \Gamma_{ei} = \phi$

- "Discontinuous at Interface" ⇨ Jump Terms

$$-\sum_e \int_{\Omega^e} \boldsymbol{\sigma}\cdot\nabla \boldsymbol{w}^h\, d\Omega + \sum_e \int_{\Omega^e} \boldsymbol{f}\cdot \boldsymbol{w}^h\, d\Omega + \sum_e \int_{\Gamma_{eh}} \boldsymbol{h}\cdot \boldsymbol{w}^h d\Gamma$$
$$+ \sum_e \int_{\Gamma_{ei}} \boldsymbol{\sigma n}\cdot \boldsymbol{w}^h d\Gamma = 0 \qquad \forall\ \boldsymbol{w}^h \in V^h$$

Interface Stabilization

$$-\sum_e \int_{\Omega^e} \boldsymbol{\sigma}\cdot\nabla \boldsymbol{w}^h\, d\Omega + \sum_e \int_{\Omega^e} \boldsymbol{f}\cdot \boldsymbol{w}^h\, d\Omega + \sum_e \int_{\Gamma_{eh}} \boldsymbol{h}\cdot \boldsymbol{w}^h d\Gamma$$
$$+ \sum_e \int_{\Gamma_i} \boldsymbol{\sigma n}\cdot \boldsymbol{w}^h d\Gamma - \sum_i \int_{\Gamma_i}\left[\boldsymbol{\sigma n}^+ + \boldsymbol{\sigma n}^-\right]\cdot \bar{\boldsymbol{w}}^{h-} d\Gamma = 0$$

- $\sum_e \int_{\Gamma_{ei}} \boldsymbol{\sigma n}\cdot \boldsymbol{w}^h d\Gamma = \sum_i \int_{\Gamma_i^+} \boldsymbol{\sigma n}^+ \cdot \boldsymbol{w}^{h+} d\Gamma + \sum_i \int_{\Gamma_i^-} \boldsymbol{\sigma n}^- \cdot \boldsymbol{w}^{h-} d\Gamma \equiv$ jump term
- $\bar{\boldsymbol{w}}^h \equiv \left(\boldsymbol{w}^{h+} + \boldsymbol{w}^{h-}\right)/2 \equiv$ average of variational displacement along Γ_i

Interface Stabilized Form

$$-\sum_e \int_{\Omega^e} \boldsymbol{\sigma}\cdot\nabla \boldsymbol{w}^h\, d\Omega + \sum_e \int_{\Omega^e} \boldsymbol{f}\cdot \boldsymbol{w}^h\, d\Omega + \sum_e \int_{\Gamma_{et}} \boldsymbol{h}\cdot \boldsymbol{w}^h d\Gamma$$
$$+ \frac{1}{2}\sum_i \int_{\Gamma_i^+} \boldsymbol{\sigma n}^+ \cdot\left[\boldsymbol{w}^{h+} - \boldsymbol{w}^{h-}\right] d\Gamma + \frac{1}{2}\sum_i \int_{\Gamma_i^-} \boldsymbol{\sigma n}^- \cdot\left[\boldsymbol{w}^{h-} - \boldsymbol{w}^{h+}\right] d\Gamma = 0$$

Error Estimation

- Explicit Error Estimation

  Fine Scale Solution: $\boldsymbol{u}' = -\tau \boldsymbol{r} = -\tau[\mathcal{L}\bar{\boldsymbol{u}} - \boldsymbol{f}]$ $\quad (*)$

  Solve coarse problem ⇨ Substitute the nodal values for $\bar{\boldsymbol{u}}$ in $(*)$

- Implicit Error Estimation

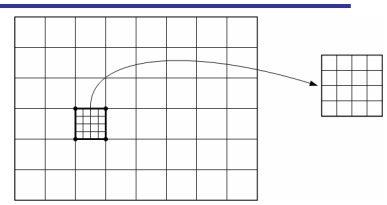
  Fine Scale Problem: $(\boldsymbol{w}', \mathcal{L}(\boldsymbol{u}')) = (\boldsymbol{w}', \boldsymbol{f} - \mathcal{L}(\bar{\boldsymbol{u}}))$ $\quad (**)$

  Substitute the nodal values for $\bar{\boldsymbol{u}}$ in $(**)$ and solve it.

  Construct patches to solve local / element level problems
  - Element based approach
  - Node based approach

Error Estimation (cont'd)

Element Based Approach



$$(\psi \boldsymbol{w}', \mathcal{L}(\boldsymbol{u}')) = (\psi \boldsymbol{w}', \boldsymbol{f} - \mathcal{L}(\bar{\boldsymbol{u}}))$$

- Need flux boundary conditions

$\psi = 1$ for given element e; $\psi = 0$ outside e

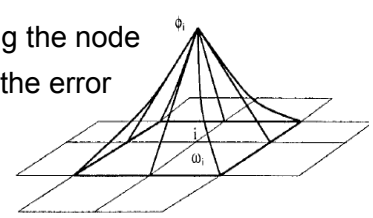
Node Based Approach

$$(\psi \boldsymbol{w}', \mathcal{L}(\boldsymbol{u}')) = (\psi \boldsymbol{w}', \boldsymbol{f} - \mathcal{L}(\bar{\boldsymbol{u}}))$$

- Need to determine size of patch feeding the node
  - ⇨ Elements within the radius feed into the error associated with node

+ Better accuracy

− Computationally expensive



$\psi = 1$ for node n; $\psi = 0$ at distance $\geq r$

Explicit Error Estimation for Elasticity

Standard L2 error norm

$$L_2 \boldsymbol{e} = \left[\int_\Omega \left(u^h - u^{\text{exact}} \right)^{\mathrm{T}} \left(u^h - u^{\text{exact}} \right) d\Omega \right]^{1/2}$$

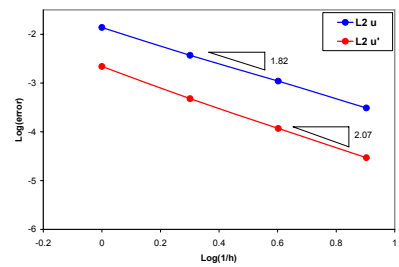
L2 error norm from fine-scales

$$\boldsymbol{e} \cong \boldsymbol{u}' = -\tau \left[\boldsymbol{f} - \mathcal{L}\, \overline{\boldsymbol{u}} \left(\boldsymbol{u}' \right) \right]$$

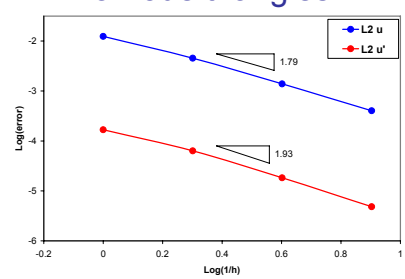
$$L_2 \boldsymbol{e} = \frac{1}{2} \sqrt{\left(\boldsymbol{e} \cdot \boldsymbol{e} \right)}$$

$$H^1 \boldsymbol{e} = \frac{1}{2} \sqrt{\left(\nabla \boldsymbol{e} \cdot \nabla \boldsymbol{e} \right)}$$

4-node quads



3-node triangles



Concluding Remarks

The meso-to-micro scale response of the interface is mathematically nested into the meso-to-macro scale response of the system.

A-posteriori error estimator is naturally built into the framework.

Modeling interface as a discontinuity blends ideas from CG and DG methods ⇨ provides a variationally consistent method for embedding interface models.

Points for Discussion:

Significance of the “Interface” Nonlinear Dynamic Effects.

Should Verification and Validation be an integral part of the analysis strategy?

Do we have the right mathematical framework for the analysis of systems that are multiphysics and multiscale in nature?

Shakedown at frictional contact

M. Ciavarella

•A. Klarbring, M. Ciavarella, J.R. Barber, Shakedown in elastic contact problems with Coulomb friction, International Journal of Solids and Structures, Volume 44, Issues 25-26, 15 December 2007, Pages 8355-8365

•James R. Barber, Anders Klarbring, Michele Ciavarella, Shakedown in frictional contact problems for the continuum, Comptes Rendus Mécanique, Volume 336, Issues 1-2, January-February 2008, Pages 34-41

•Young Ju Ahn, Enrico Bertocchi, J.R. Barber, Shakedown of coupled two-dimensional discrete frictional systems, Journal of the Mechanics and Physics of Solids, Volume 56, Issue 12, December 2008, Pages 3433-3440

D.2.9 Slide Presentation of M. Ciavarella: *Shakedown at Frictional Contact*

We consider a system of contacting elastic bodies, discretized using the finite element method with incremental Coulomb friction boundary conditions.

The external loading

$$\boldsymbol{F}(t) = \boldsymbol{F}_0 + \lambda \boldsymbol{F}_1(t)$$

comprises a constant 'mean load' $\boldsymbol{F}_0$ and a periodic time-varying load $\lambda \boldsymbol{F}_1(t)$ where λ is a scalar load factor.

We assume that the loading is never sufficient to cause 'gross slip' (sliding) in which all the contact nodes slip at the same time. However, at any given time a subset of the contact nodes may slip - a state known as 'microslip'.

After an initial transient, the long-time steady state might involve:-

- *Shakedown*: Slip displacements during the transient phase lead to a state of residual stress that prevents all slip in the steady state.
- *Cyclic slip*: The slip scenario is exactly repeated during each loading period and the total accumulated slip at each node during one period is zero.
- *Ratcheting*: The slip scenario is exactly repeated during each loading period, but a constant total slip is accumulated at each node during one period each period. This is possible only if the system has a rigid-body mode.

This behaviour is closely analogous with that of elastic-plastic systems under periodic loading.

However, there are important differences:-

1) The 'failure' (i.e. slip) condition for Coulomb friction is a dimensionless coefficient of friction, rather than a yield stress.
One consequence is that if the complete loading scenario $\boldsymbol{F}(t)$ is multiplied by a positive scalar factor, the long term behaviour will be qualitatively unchanged and the nodal forces and displacements will all be increased by the same factor.

2) The flow rule for frictional slip is non-associative.

For elastic-plastic deformation with an associative flow rule, Melan's theorem applies.

Many frictional systems appear to obey a form of Melan's theorem.

However, the proof of Melan's theorem for plasticity depends on the flow rule being associative and this is not satisfied by the Coulomb friction law.

We have recently proved that Melan's theorem applies to elastic systems with Coulomb friction contact conditions if and only if there is no coupling between tangential (slip) displacements and normal contact reactions.

Counter examples can be found for all coupled systems in that the final steady state can depend upon the initial conditions.

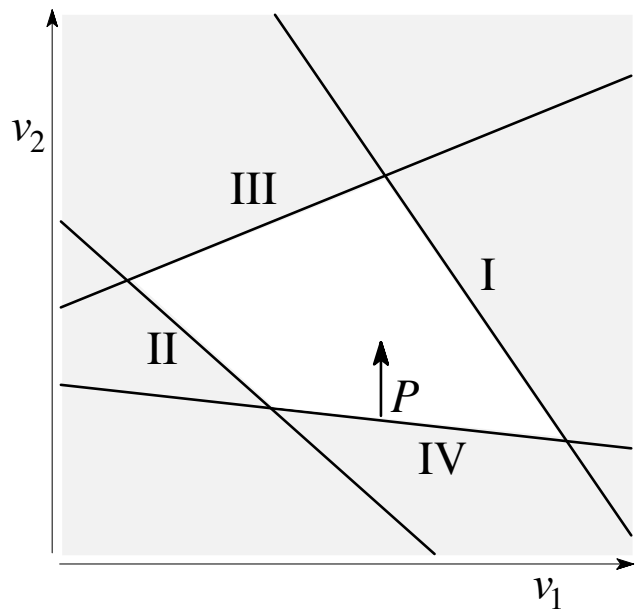
We assume all nodes remain in contact ($w_i = 0$) and define the instantaneous state as a point P in the space v_i of slip displacements.

The frictional constraints (incipient forward or backward slip) at node j are defined by

$$\begin{aligned}(A_{ji} - fB_{ji})v_i &\leq fp_j^w - q_j^w \\ (A_{ji} + fB_{ji})v_i &\geq -fp_j^w - q_j^w \ ,\end{aligned}$$

Each of these $2N$ constraints (2 for each node) is a directional hyperplane in v_i-space.

We illustrate this for a simple two-node system.



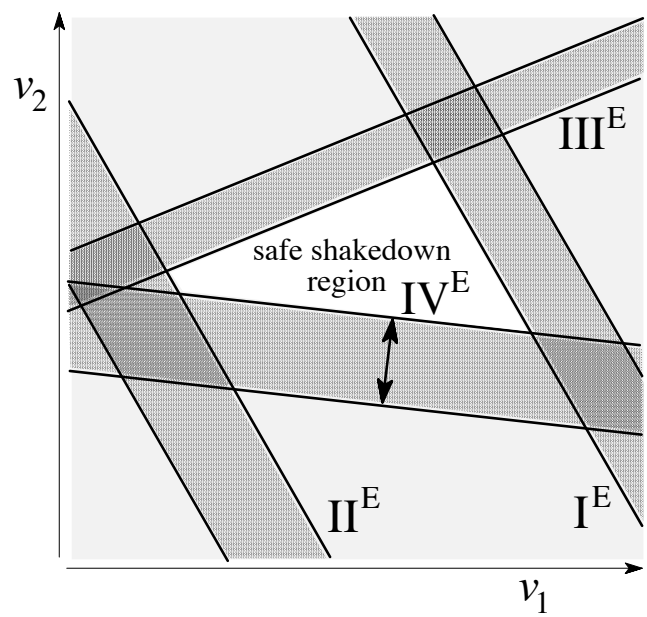
The four constraints exclude the (shaded) region of $v_1 v_2$-space on one side of the lines I,II,III,IV respectively.

The instatansous state of the system must lie in the unshaded region.

If the external loads cause IV to move to exclude more space, P is 'pushed' upwards ($\dot{v}_2 > 0$).

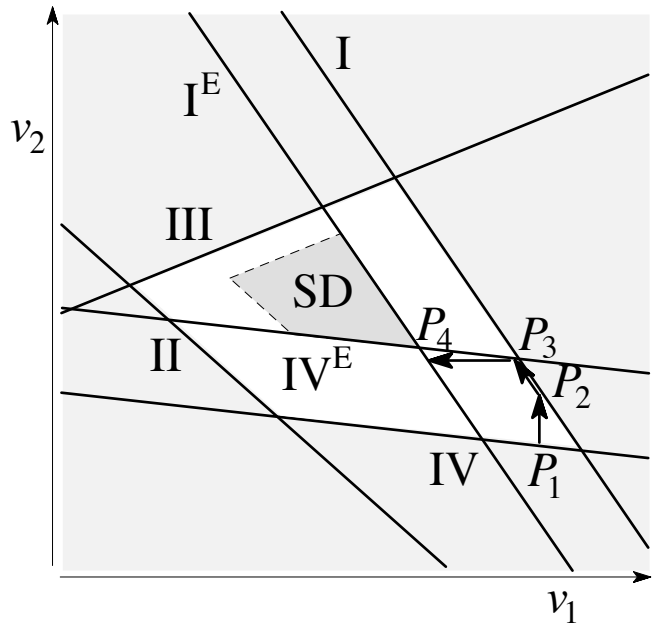
Now imagine the external loads changing periodically, so that the lines I,II,III,IV move back and forth as shown.

We can identify the extreme positions I^E etc. where each constraint excludes the maximum space.



Shakedown is possible if and only if these extreme constraints leave a non-null safe shakedown region.

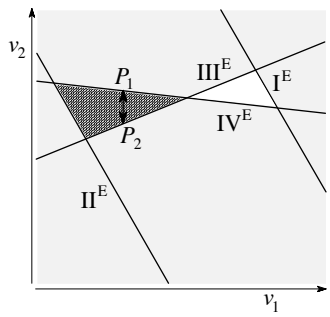
Suppose IV advances to IVE and then recedes, after which I advances to I^E and recedes.



The point P is moved monotonically towards the safe shakedown region SD.

This is always true if the safe shakedown region is a quadrilateral.

If the safe shakedown region is a triangle, the steady state can be either cyclic slip or shakedown, depending on the initial condition (position of P).



In the case illustrated, P could end up cycling between P_1 and P_2.

If the scalar load factor λ is increased from zero, the safe shakedown region is initially quadrilateral. At some critical value λ_L it becomes triangular, and then at a higher critical value λ_U it becomes null.

- For $\lambda < \lambda_L$ we have shakedown for all initial conditions.
- For $\lambda_L < \lambda < \lambda_U$ we may have shakedown or cyclic slip depending on the initial conditions.
- For $\lambda > \lambda_U$ shakedown is impossible and we have cyclic slip for all initial conditions.

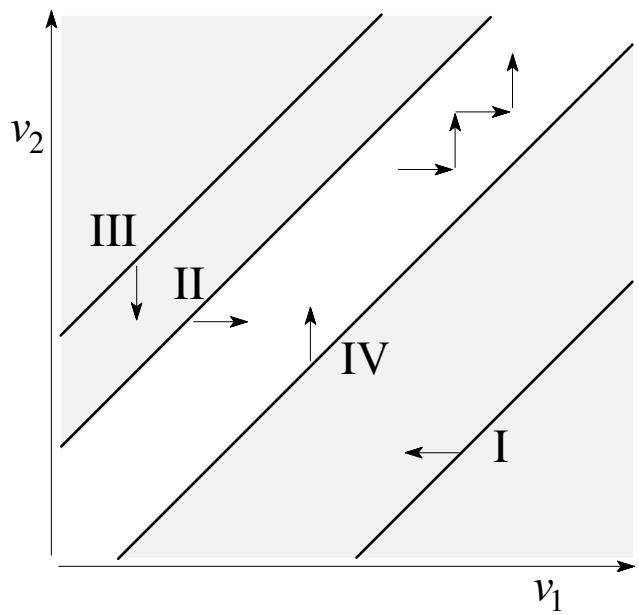
If the system is uncoupled, the constraints are parallel in pairs and the safe region is a parallelogram. It can therefore never reduce to a triangle and $\lambda_L = \lambda_U$.

This confirms that Melan's theorem applies when the system is uncoupled.

Ratcheting

If the system has a rigid-body mode, the stiffness matrix is singular and the constraints are all parallel to a common line.

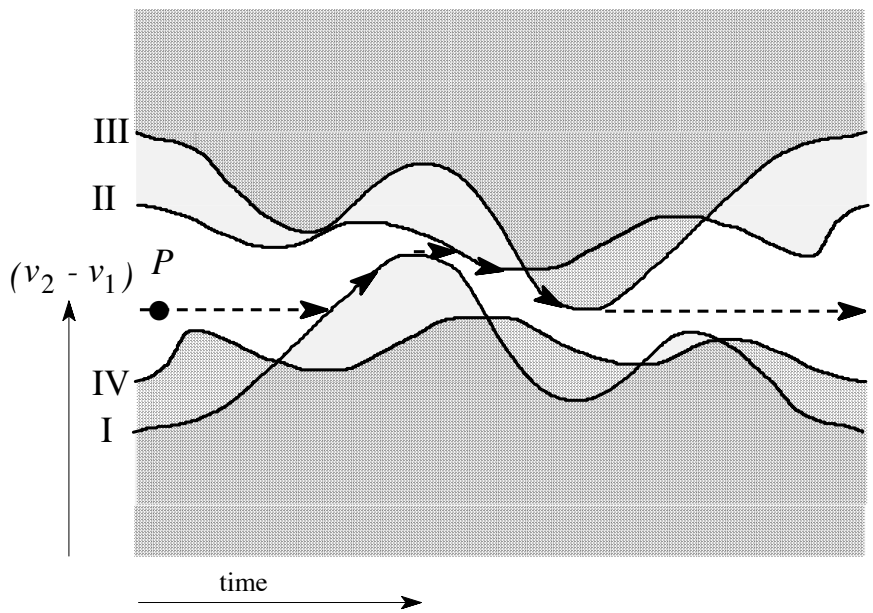
For the two-node system with possible rigid-body motion $v_1 = v_2$, they are at 45°.



With appropriate motion of the constraints, we can obtain the ratcheting motion illustrated.

However, if there is a safe shakedown region (a strip in this case), the system must shakedown.

Motion of P depends on which of the constraints is most advanced at a given time. We can project these positions onto the line orthogonal to the rigid-body motion and plot them as a function of time.

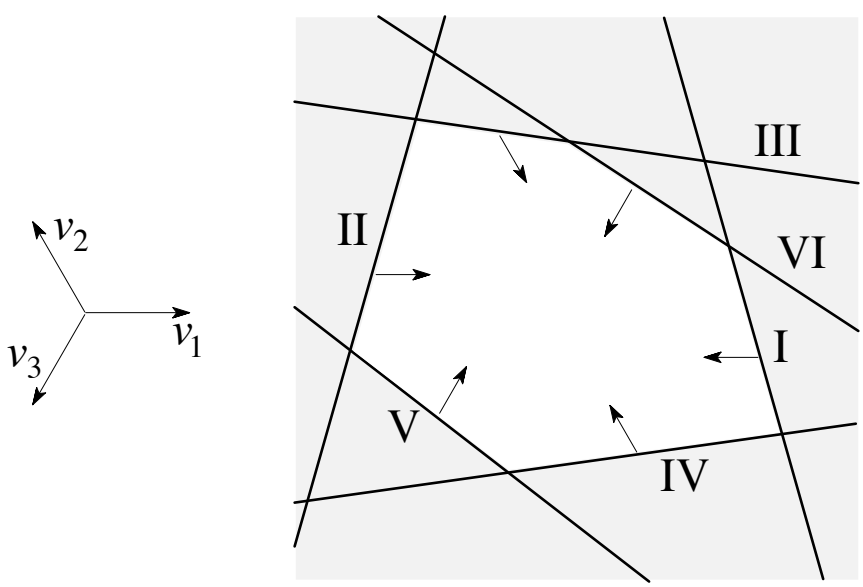


The motion of P is equivalent to the trajectory of a ball dropped through the space between the constraints in the figure.

Either cyclic slip or ratcheting is possible depending on the sequence of constraint motions.

Depending on the exact form of the periodic load $\boldsymbol{F}_1(t)$, increasing λ can cause a transition to either ratcheting or cyclic slip and further transitions between these states can occur at higher values of λ.

> For a three-node system with a rigid-body mode, the constraints will be planes all parallel to a common line $v_1 = v_2 = v_3$.
>
> Looking along this line, we can track the deformation of the structure in response to the (now six) moving constraints.



Conclusions

The motion of the point $P(v_1, v_2, ...)$ in v_i-space provides useful information about the kinds of behaviour to be expected under time-varying loads.

The lower bound λ_L below which shakedown occurs for all initial conditions can be found by solving subsets of the frictional constraints as equalities and checking the remaining constraints as inequalities.

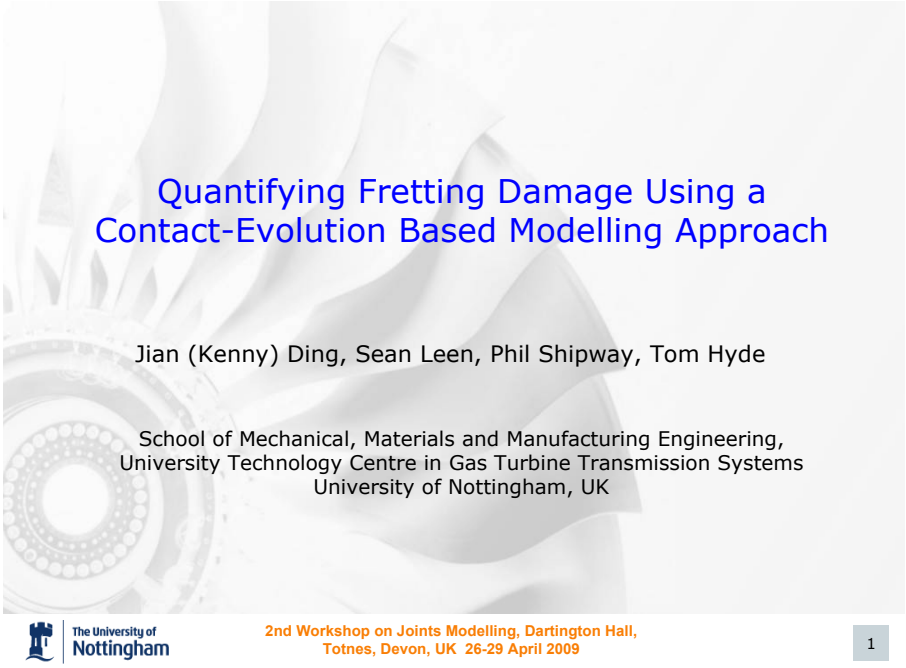
The upper bound λ_U above which shakedown is impossible can also be found this way, or alternatively by using a constrained optimization technique.

In systems with a rigid body mode, all the constraints are parallel to a given line and evolution of the system can be tracked by projecting the space onto a hyperplane perpendicular to this line.

•A. Klarbring, M. Ciavarella, J.R. Barber, Shakedown in elastic contact problems with Coulomb friction, International Journal of Solids and Structures, Volume 44, Issues 25-26, 15 December 2007, Pages 8355-8365

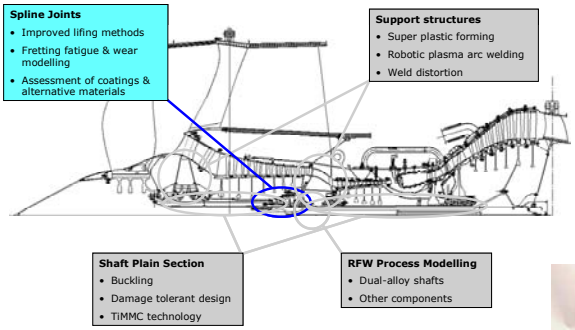
•James R. Barber, Anders Klarbring, Michele Ciavarella, Shakedown in frictional contact problems for the continuum, Comptes Rendus Mécanique, Volume 336, Issues 1-2, January-February 2008, Pages 34-41

•Young Ju Ahn, Enrico Bertocchi, J.R. Barber, Shakedown of coupled two-dimensional discrete frictional systems, Journal of the Mechanics and Physics of Solids, Volume 56, Issue 12, December 2008, Pages 3433-3440



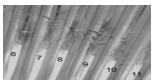
D.2.10 Slide Presentation of Jian Ding, University of Nottingham: *Quantifying Fretting Damage Using a Contact-Evolution Based Modelling Approach*

Nottingham UTC in Gas Turbine Transmission Systems



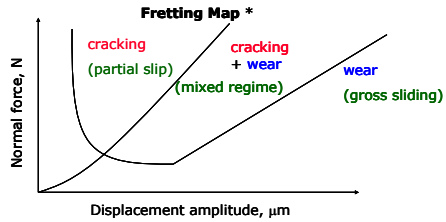
Study of Spline Couplings

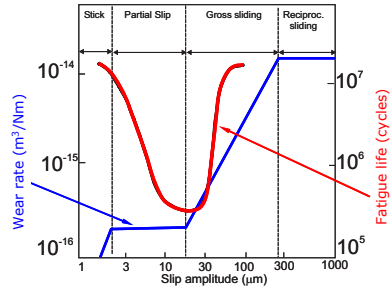
- Experimentally characterise the fretting behaviour of splines using scaled-down spline and/or representative specimens
- Develop lifing methodologies for spline against fretting



Spline fretting failure

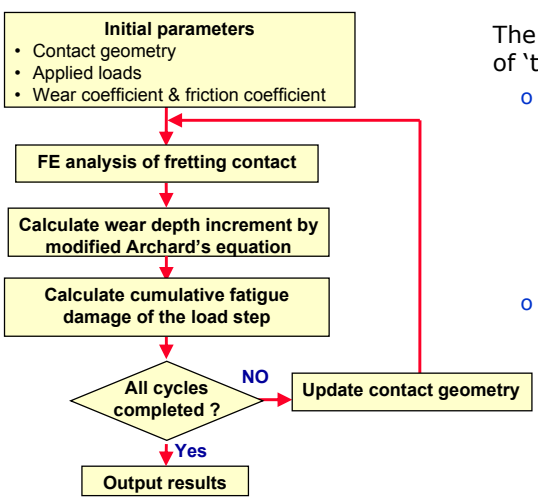
Contact-evolution based lifing methodology





- An approach that considers transient interaction between wear and fatigue under fretting, especially the effect of wear on fatigue life.
- Contact-evolution based lifing approach comprises:
  - A finite element wear simulation tool to determine the evolution of contact geometry.
  - Damage Accumulation approach for crack nucleation.
- Ongoing EPSEC project in collaboration with Oxford (total grant ~0.6 Millions)

Modelling Framework



The approach integrates a number of 'tools'.

- Fretting wear tool is central, which predicts the extent of wear damage and the concomitant change of contact geometry

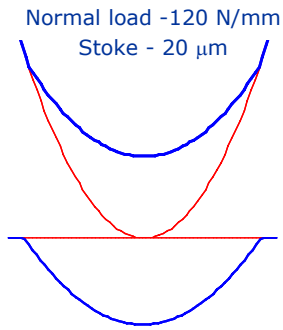
  Archard's Wear equation:

$$dh_i(x,t) = k_i \times p(x,t) \times ds(x,t)$$

- For each wear step n, accumulated fatigue damage is calculated by fatigue parameter Smith-Woston-Topper; thus, total accumulated damage is given by

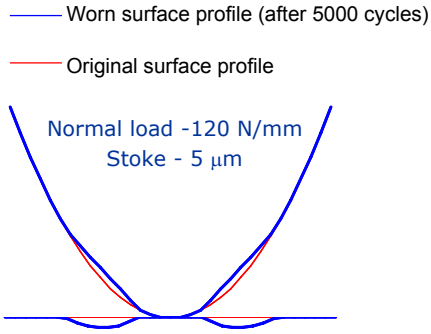
$$_i\varpi = \sum_{n=1}^{n=\frac{N_T}{\Delta N}} \frac{1}{N_{i,n}}$$

Fretting Wear Modelling



Contact width increase markedly from Hertz prediction

Gross slip case



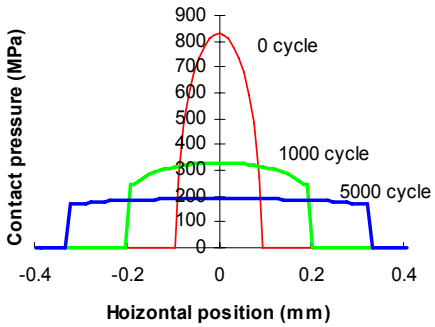
Little change of contact size, wear occurring at slip zone

Partial slip case

(Ding et al, *Int J of Fatigue*, 2004)

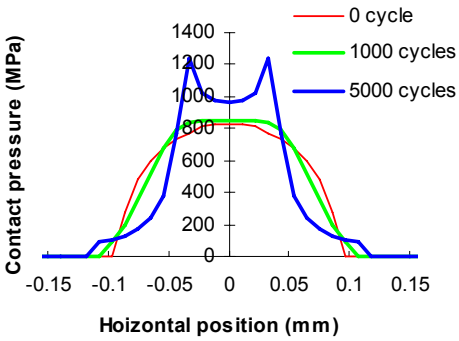
Fretting Wear Modelling

Normal load -120 N/mm
Stoke - 20 μm



Gross slip case

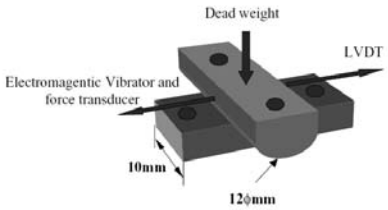
Normal load -120 N/mm
Stoke - 5 μm

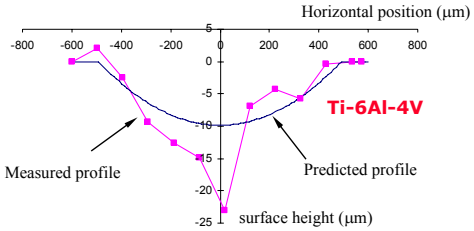


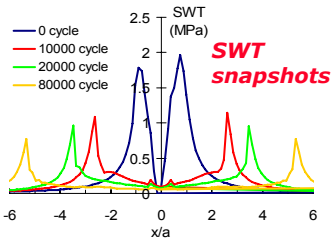
Partial slip case

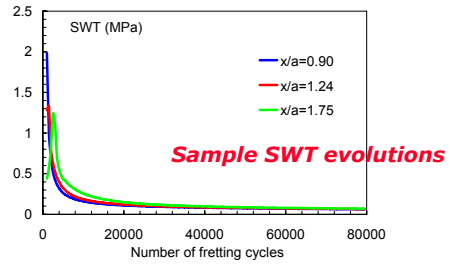
Evolution of contact pressure

Contact-evolution based prediction of crack nucleation (I) gross sliding

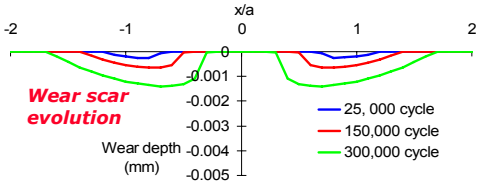


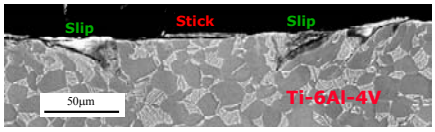


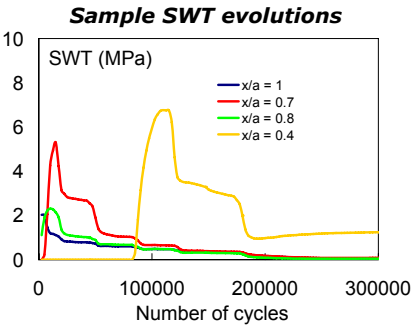




Contact-evolution based prediction of crack nucleation (II) partial slip

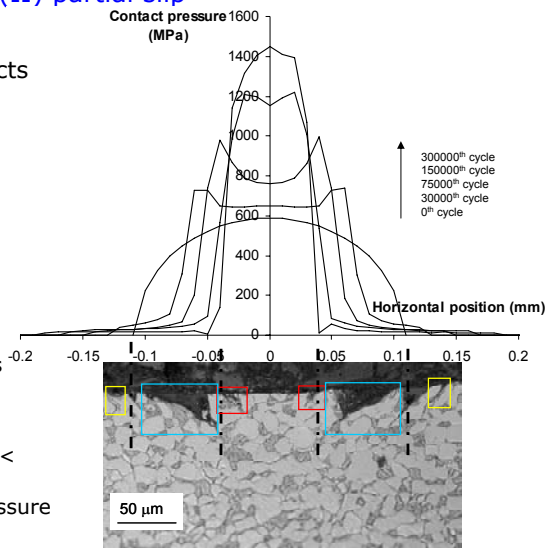




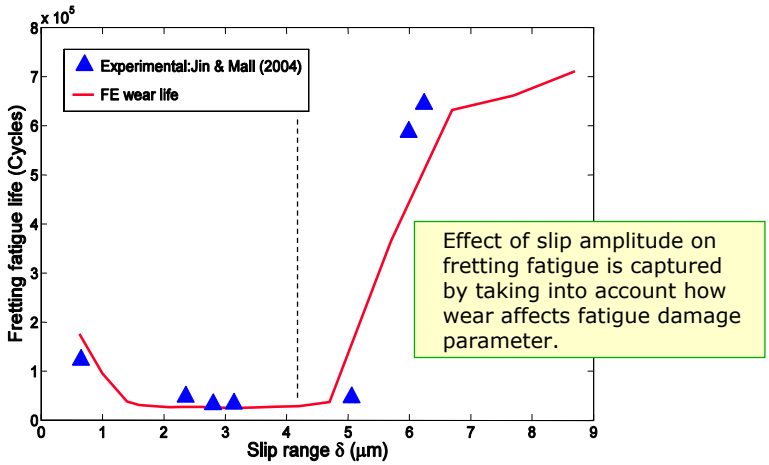


Contact-evolution based prediction of crack nucleation (II) partial slip

- Complex pressure evolution predicted due to plasticity effects
- Multiple cracking locations
  - $x \approx \pm 0.11$ mm (light blue)
  - $x \approx \pm 0.04$ mm (red)
  - $x \approx \pm 0.13$ mm (yellow)
- $x \approx \pm 0.13$ mm
  - Initial Hertzian contact edge
  - early cycles, low COF: gross sliding (N < 3k)
- $x \approx \pm 0.04$ mm
  - ~ initial stick-slip boundaries
  - late cycles ($N \approx$ 150k-300k)
- $x \approx \pm 0.11$ mm
  - intermediate cycles (3k < N < 150k)
  - due to flat indenter type pressure peaks



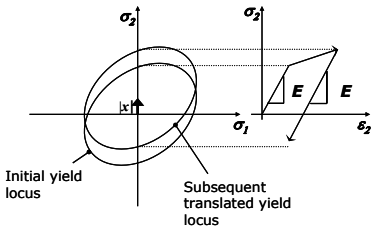
Effect of slip amplitude on fretting fatigue

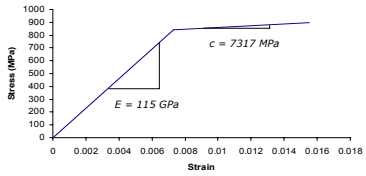


Effect of slip amplitude on fretting fatigue is captured by taking into account how wear affects fatigue damage parameter.

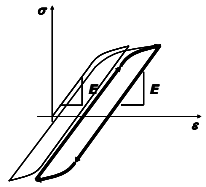
Prediction vs. tests (Madge et al, 2007)

Cyclic Plasticity in Fretting

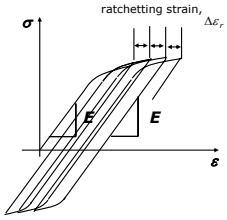




- Prager linear kinematic hardening
- Material: Ti-6Al-4V (α + β)
- Coefficient of friction – 0.9



Plastic shakedown
steady reversed cyclic plastic strains



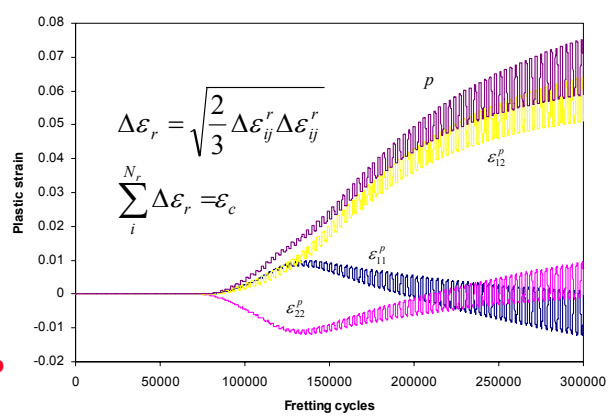
Ratchetting
plastic strain magnitude increases continually with load cycling

Cyclic Plasticity in Partial Slip

- Nominal Hertzian geometry ➔ elastic
- Wear simulation with plasticity ➔ ratchetting phenomenon
- Possibility of damage/cracking due to ductility exhaustion

$$\Delta\varepsilon_r = \sqrt{\frac{2}{3}\Delta\varepsilon_{ij}^r \Delta\varepsilon_{ij}^r}$$

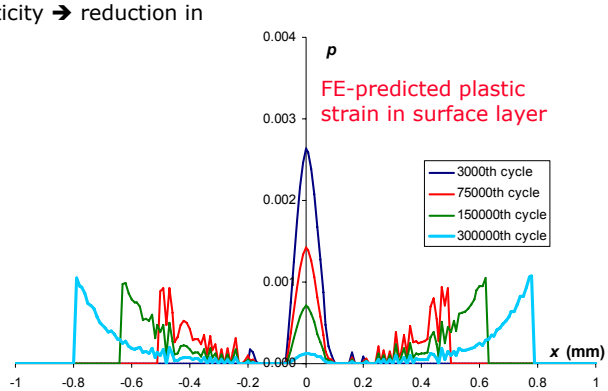
$$\sum_i^{N_r} \Delta\varepsilon_r = \varepsilon_c$$



Predicted wear-induced evolution of plastic strains at final stick-slip interface

Cyclic Plasticity in Gross Sliding

- Gross sliding: shear-dominant plasticity
- g.s. plasticity take a W-shape
- Wearing away of plasticity ➔ reduction in equivalent plastic strain



Conclusions and Future Challenges

- Contact-evolution based fretting lifing methodology provides
  - an integrated solution for fretting wear and fatigue prediction.
  - a convincing explanation about the effects of slip amplitude on fretting fatigue
- Future challenges:
  - Incorporate near-surface effects into fretting fatigue prediction, such as asperity, oxidation, plasticity and debris accumulation. How important are they for fretting crack nucleation?
  - Fretting contact mechanics under micro or nano scales.

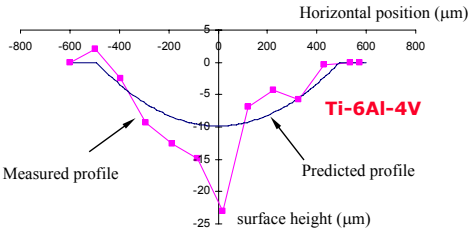
Contact-evolution based prediction of crack nucleation (I) gross sliding

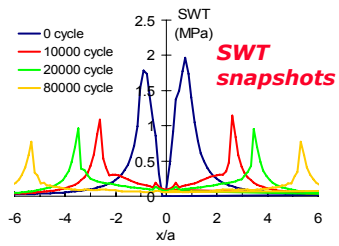
Crack nucleation defined to occur at material point *i* when accumulated damage ω reaches value of 1, where ω is defined as :

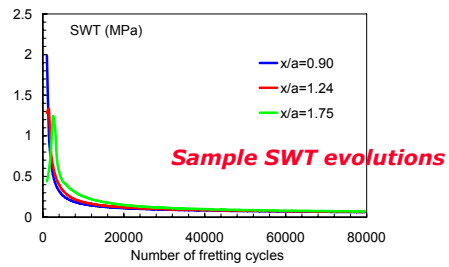
$$\varpi = \sum_{n=1}^{n=\frac{N_T}{\Delta N}} \frac{\Delta N}{N_{i,n}}$$

Each $N_{i,n}$ is calculated based on a critical-plane fatigue damage parameter Smith-Watson-Topper (SWT).

$$(\sigma_{\max}\Delta\varepsilon_a)_{i,n} = \frac{(\sigma_f')^2}{E}(2N_{i,n})^{2b} + \sigma_f'\varepsilon_f'(2N_{i,n})^{b+c}$$







Modeling Interfaces in Structural Dynamics Analysis

Enriching our joints models and capabilities

2nd Workshop on Joints Modelling
Dartington Hall
Totnes, Devon, UK
26-29 April 2009

Michael Starr, Org. 01526
Applied Mechanics Development
Sandia National Laboratories<

<Sandia is a multiprogram laboratory operated by Sandia Corporation, a Lockheed Martin Company, for the United States Department of Energy's National Nuclear Security Administration under Contract DE-AC04-94-AL85000.

D.2.11 Slide Presentation of Michael Starr, Sandia National Laboratories: *Modeling Interfaces in Structural Dynamics Analysis: Enriching Our Joints Models and Capabilities*

We Wish to Develop New Tools and Capabilities That Retain a Connection to Our Physical Understanding of Interfaces

- Enhanced capabilities in structural dynamics analysis
- Reduced-order interface constitutive models
- Handbook to aid analysts in populating models
  - generalized classes of joints and configurations
  - enriched parameter sets through variability

Historically we have used experimental data to populate models. We would like to move toward numerical "experiments" to derive model parameters, *e.g.* postulate phenomena that manifest as variability observed in experiments and sample from that physical space to generate a statistical representation.

Sandia National Laboratories

Extracting Reduced-Order Joint Parameters from Fine-Mesh Finite Element (FMFE) Models

Parameter Extraction in the Absence of Experimental Data

There are Fundamental Experimental Difficulties that Make Numerical Analysis Appear Attractive

- The physics to be measured all occurs exactly where it cannot be measured directly.
- Kinematics of joint displacements cannot be well-defined in an experimental context.
- Every specimen mounting adds its own features to measurements.
- Specimen compliances drown out joint response except at very high loads.

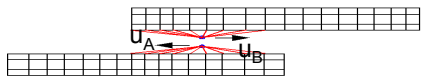
Potential benefit of running FMFE:

> We, of course recognize the importance of verifying the accuracy of any prospective finite element code we intend to use in our analyses

The Development of a Reduced-Order Interface Model Was Driven By Necessity

- FMFE models are not suitable for direct implementation in structural dynamics because of imitations of Courant time step (explicit) conditioning (implicit). This is why we must seek continuum level constitutive models.

- Connect the kinematics of element nodes in the "process zone" to that of a representative node.



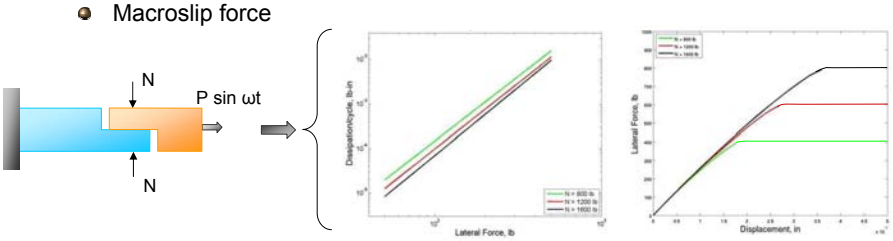
- This approach has weaknesses
  - it does not account for the multi-dimensional nature of loading.
  - it does not account for the true complexity of contact.
  - it relies on the postulated interface law.
  - it ignores micro-level effects that lead to wear-in.

Sandia National Laboratories

Iwan Parameters Can Be Deduced From a Small Set of Numerical Experiments

Calculated quantities required to fully populate the Iwan 4-Parameter constitutive model:

- Dissipation per cycle over a range of loads below macroslip
- Joint stiffness over a range of loads
  - Joint stiffness calculated for a monolithic specimen
  - Elastic compliance of the jointed specimen
- Macroslip force



These quantities could be calculated deterministically to satisfy known distributions, or stochastically using a scheme that reproduces the distributions of known measured quantities.

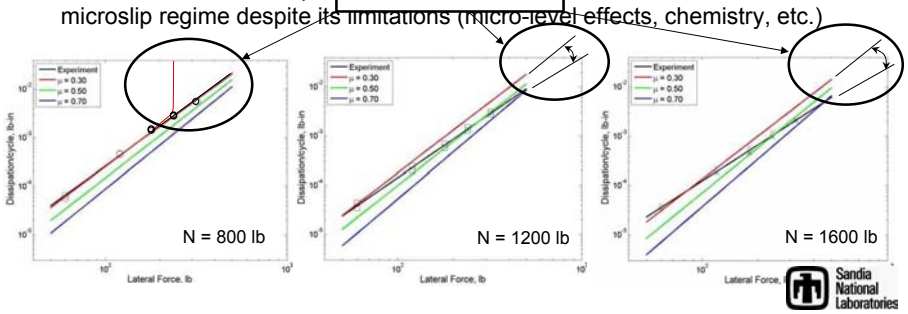
Sandia National Laboratories

Predictions of Dissipative Response For Generalized Lap and Flange Joints

Employing a Coulomb friction interface constitutive model

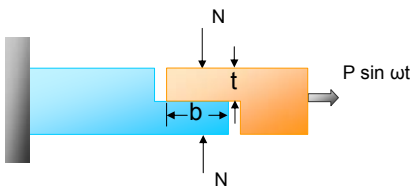
It is Not Efficient for Analysts to Run Full Suites of Simulations to Capture Variability and Uncertainty Using FMFE

- Provide a "Roark's Handbook" of joint response for each important class of joint, providing estimates of the following for arbitrary values of geometry and material parameters
  - Joint dissipation and softening
  - Joint constitutive behavior
  - Variability in the above as a function surface characteristics
- Handbook calculations per "missing" physics friction model in the microslip regime despite its limitations (micro-level effects, chemistry, etc.)

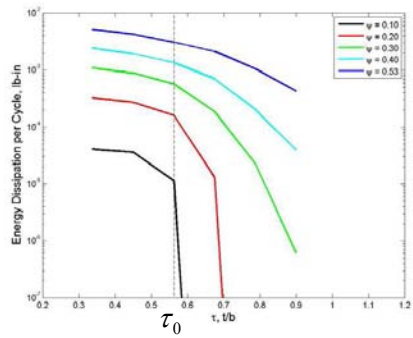


Non-Dimensional Parameter Studies of Generalized Jointed Interfaces were Performed to Better Understand the Contact Mechanics

- Focused on a convenient non-dimensional parameter, ψ = P/μN, which characterizes the fraction of the macroslip load acting across the interface.



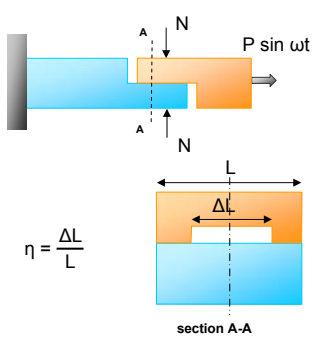
$$\tau = t / b$$

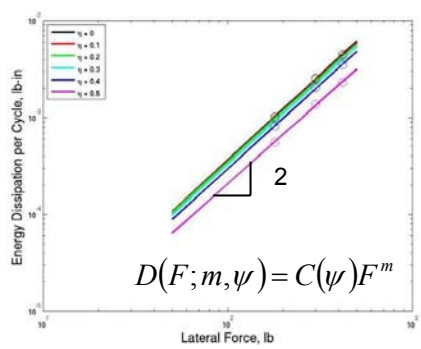


- Tuning to a particular experimental measurement, these results can be used to interpolate or extrapolate to other geometries, stiffnesses, and loads.

Though Coulomb Friction is not Quantitatively Correct, It May Be a Useful Tool For Leveraging Sparse Experimental Data

- FMFE simulations for several cases of loading conforming to ψ = P/μN = constant exhibited a power law response in dissipation, D, with slope $m \approx 2.0$





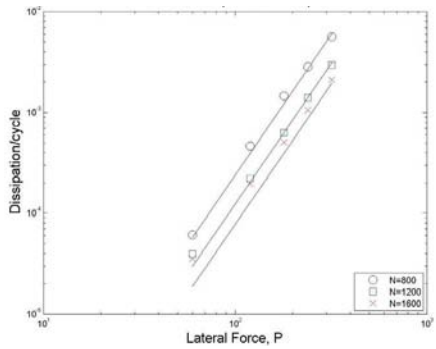
- The power law behavior is also independent of actual contact area for constant ψ.

FMFE Suggests Relationships to Look For in Experimental Data

- Given one small set of experimental data at ψ = constant and another small set at N_0 = constant, a relationship can be derived to predict energy dissipation at other force levels.

$$D(\mu N, P) = C_0 (\mu N_0 \psi)^{\alpha_0 - m} P^m$$

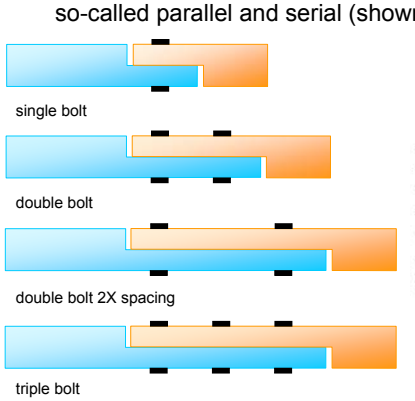
C_0, α_0, m { determined from experimental data



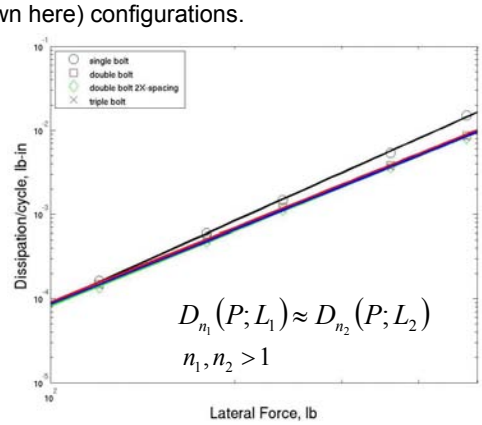
- The Ψ-relationship asserts that the power law dissipation slope is the same for all clamping loads (good assumption for numerical work, bad assumption for experiments).

Coarse Relationships Have Been Developed for "Serially" Bolted Flange Connections

- There are two principal flange orientations that generalize a flange ring, the so-called parallel and serial (shown here) configurations.



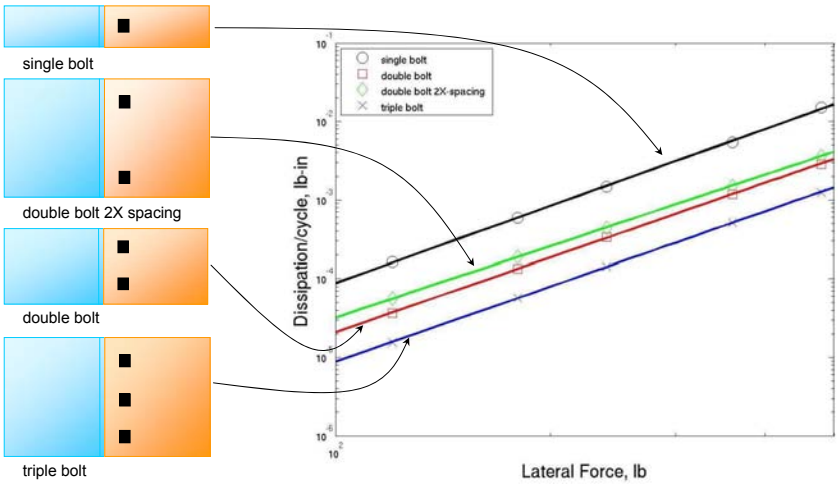
several serial configurations



- The predictions of energy dissipation per cycle are similar between the double bolt and triple bolt cases despite the obvious differences in loading levels and normal stress distributions on the interfaces.

Coarse Relationships Have Also Been Developed for "Parallel" Bolted Flange Connections

- The parallel configuration behaves differently than the serial configuration.

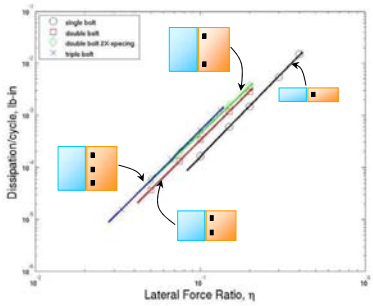


several parallel configurations

- Let's plot the results somewhat differently.

Normalizations Expose Relationships Between Numerical Predictions

- Define lateral force ratio as η = P/nN, where n is the number of bolts in the flange.

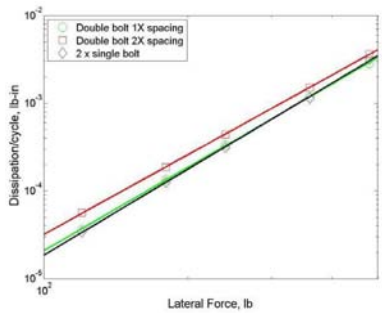


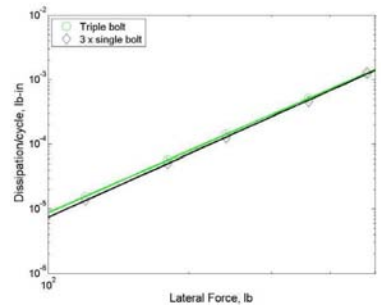
- The double bolt "looks like" the triple bolt at low η, but more interestingly there appears to be a functional relationship between the curves at constant, η.

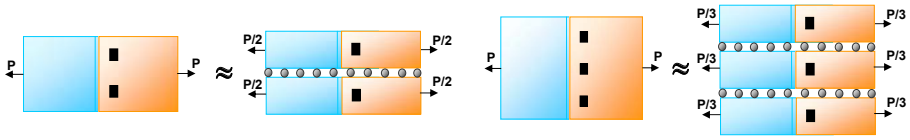
$$D_n(P;n) \approx nD_1\left(\frac{P}{n};1\right) = nC_0\left(\frac{P}{n}\right)^{\alpha} = n^{1-\alpha}D_1(P;1)$$

A Reasonable Approximation for the Parallel Flange Configuration

- The prediction is reasonable for the single-spaced geometries.







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FMFE Predictions Without Incorrect Interface Constitutive Models are not Without Merit

- Phenomenologically motivated numerical simulations have been performed to understand the impact of boundary conditions, joint misfit, and surface features (machining and intrinsic roughness) on predictions of dissipation.
- Although the numerically deduced relationships struggle to extrapolate to experimental data outside of its calibration range, they extrapolate exceptionally well to numerical predictions.
- A very sparse set of numerically calculated data can be used in conjunction with small number of experimental sources to fully populate variability studies from which model parameters can be deduced (such as the four-parameter Iwan model).

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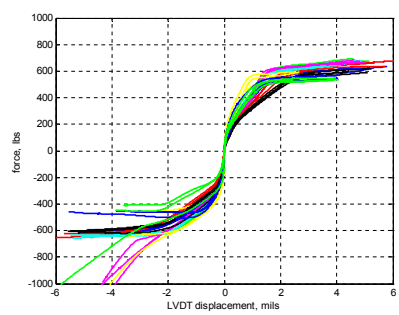
Physical Connections to Observed Variability in Jointed Structures

Continuum Modeling of Causal Variability Phenomena

The Major Source of Variability in Measured Physical Quantities is Part-to-Part Variability

- Even for a simple monotonic, quasistatic pull, with nominally identical clamping load, force saturates in a broad range, $F_S \in (450,634)$ lb

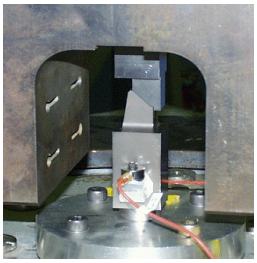
three "tops" and three "bottoms" yield nine unique conformal joint assemblies



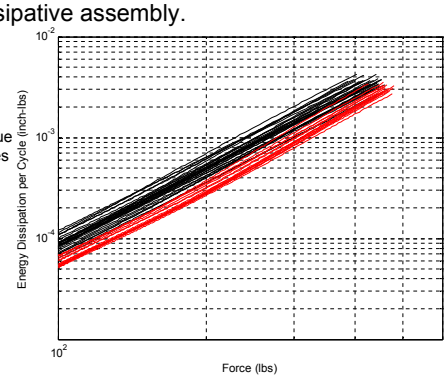
- For a given clamping load (say 1200 lb), we could conclude that the Coulomb friction coefficient is distributed within the range, $\mu \in (0.375,0.528)$, but this doesn't address the underlying phenomena that manifests this distribution.

Observation of Trends in Data Give Clues to Causal Phenomena of Variability

- The most dissipative assembly of nominally identical joint pairs is three times as dissipative as the least dissipative assembly.



three "tops" and three "bottoms" yield nine unique conformal joint assemblies

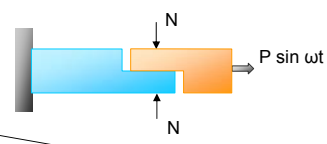


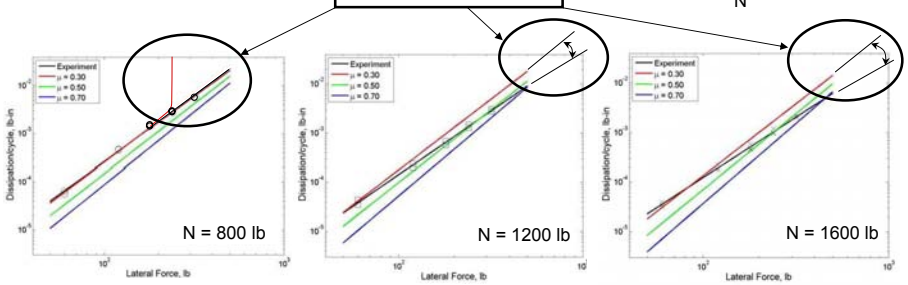
- Are there physical characteristics with length scales on the order of the microslip zone that could lead to the observed spread in energy dissipation?

Coulomb Friction-Based Constitutive Modeling Has Limitations in Microslip Regimes

- Apparently simple dissipation behavior can not be predicted by a simple dry friction model

"missing" physics

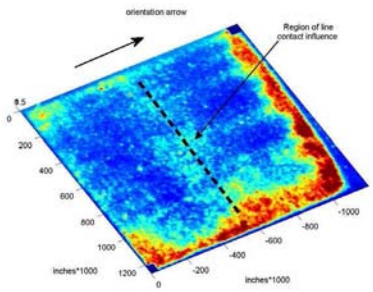




- This systematic deficiency in the Coulomb friction interface constitutive model can not be resolved through mesh refinement.

The Contact Patch Process Zone is Still Poorly Understood

- Assembling a pressure sensitive film into a simple lap joint interface provides a qualitative snapshot of normal pressure on a conformal, self-aligning interface.



- The digitized film shows an apparent assembly misfit, periodic machining marks, and local surface roughness characteristics.
- Let's postulate that these features, all of "continuum" length scale, are the causal phenomena which produce the observed variability in measured quantities.

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We Would Like to Advance Our Understanding of the Physical Phenomena While Providing Enhanced Tools for Analysts

- It is not realistic to expect analysts to model features such as surface roughness, machining characteristics, and misfit in their analyses of engineering structures.

- Use a reduced-order model to replace discrete and distributed interfaces.
  - Iwan model
  - Distributed modal damping

- In a Joints Handbook provide tables of parameters that could be used to populate those reduced-order models.
  - Perform suites of non-dimensional joint studies to allow the selection of nominal parameters consistent with individual geometries and loading conditions.
  - Perform stochastic simulations to create distributions of parameters consistent with those observed experimentally.

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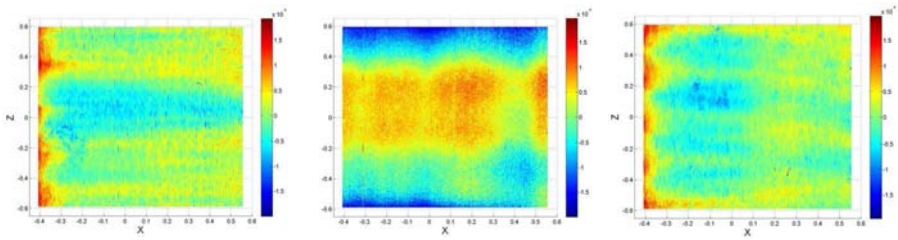
Two Analysis Paths Can Add to Our Qualitative Understanding of Interface Mechanics

The pressure film data is an exceptionally rich set of information. The contributions of feature to overall variability cannot realistically be deconvolved.

- Perform simulations on representative interfaces that account for surface features.

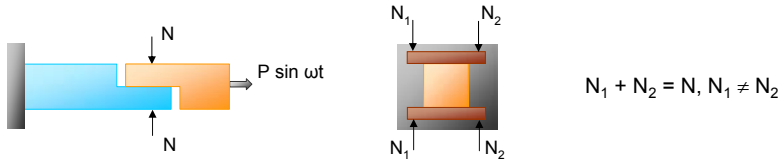


Confocal microscopy scans of flat lap joint surface heights

- Use the surface heights on the contacting surfaces as defined by microscopy and perform fine-mesh finite element simulations to predict mechanical properties of jointed structure.

Numerical Modeling and Predictions can Guide the Design of Experimental Explorations

- Remove effects of surface features from simulations and predict impact of introducing misfit through preferential loading of lap interface.

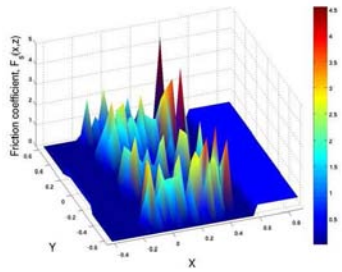


Schematic of preferential loading configuration as proxy for interfacial misfit.

- Compare to experiment and confirm/invalidate predicted trends in mechanical response of jointed interface.

Indirect Modeling Technique for Recovering Quantitative Joint Properties and Variability

- Recognize the simplicity and elegance of Coulomb's friction relationship, but concede that it was never intended to be applied locally. However, the simple, point-wise equilibrium relationship is still valid (Micro-mechanical approach).



Spatial friction map deduced from assembly of rough contacting interfaces, designed to macroslip as a smooth interface with constant friction coefficient.

- Can the "missing" physics be explained through a spatial variation in friction coefficient?
- Is it essential to retain a connection to experimental observation?

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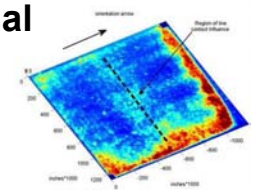
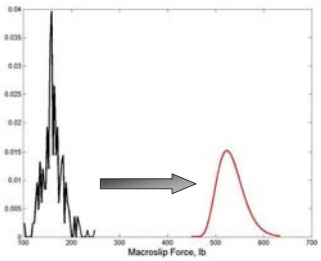
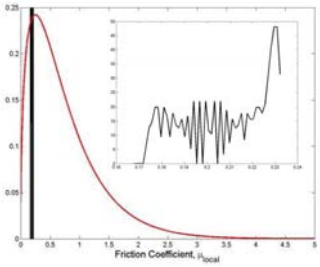
Elements That Must Be Retained if the Spatial Surface Characteristics are the Causal Components of Variability



With regard to the flat lap pressure profile shown profile shown previously:

- The true contact area is some small fraction of apparent contact area.
- An admissible spatial field must achieve macroslip within a predefined range.
- A collection of admissible spatial fields must achieve macroslip in a distribution consistent with a given distribution.
- In order to best recover interface kinematics, there may be an additional requirement on the distribution of nodal friction values.





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A Reduced-Order Model for Two-sided Interfaces

D. Dane Quinn[1] Jason Miller[1]

[1]The University of Akron
Akron, OH 44325–3903

April 27–29 2009

special thanks to Dan Segalman and Sandia National Labs

D.3 Presentations of April 29, 2009 New Ideas for Future Development of Joint Models

D.3.1 Slide Presentation of Dane Quinn, University of Akron: *A Reduced-Order Model for Two-sided Interfaces*

Abstract

Modeling mechanical joints in an accurate and computationally efficient manner is of great importance in the analysis of structural systems, which can be composed of a large number of connected components. This work presents an interface model that can be decomposed into a series-series Iwan model together with an elastic chain, subject to interfacial shear loads. The model is developed and two formulations of the model are considered. Results are then presented as the interface is subject to harmonic loading of varying amplitude. The models presented are able to qualitatively reproduce experimentally observed dissipation scalings. Finally, the interface models are embedded within a larger structural system to illustrate there effectiveness in capturing the structural damping induced by mechanical joints.

- Friction damping in mechanical joints and interfaces contribute a significant fraction of the dissipation in complex engineering structures;

What do we need? **Better Physics, Better Computations**

- Predictive structural models require an accurate representation of the behavior at and near the interface;
- **Small length scales of microslip lead to prohibitivly large computational times**

Existing approach:

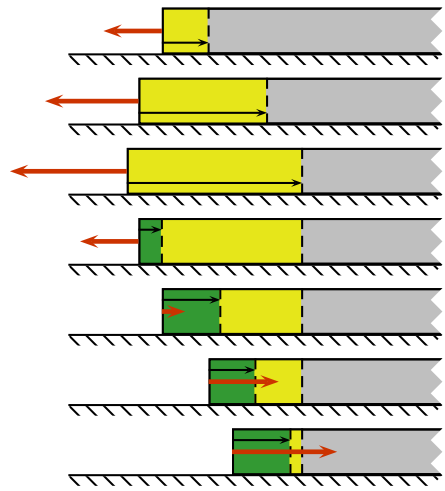
- incorporate the observed dissipation into a linear joint model with effective mass, damping and stiffness parameters

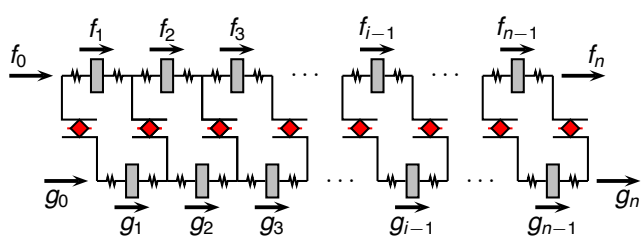
But the tuning is tied to the response of a particular test—the joint model is no longer predictive.

- Menq et al. (1986a,b) develop a continuum model representing the microslip that arises in frictional dampers;
- Segalman (2002) has developed a four parameter Iwan model that is capable of reproducing the qualitative properties of the joint dynamics;
- Song et al. (2004, 2002) have developed an adjusted Iwan beam element (AIBE) based on a parallel-series Iwan model that can be naturally incorporated into an existing finite element framework. With the proper identification of the model parameters, the AIBE can be used to capture experimentally observed profiles for the response of jointed structures.

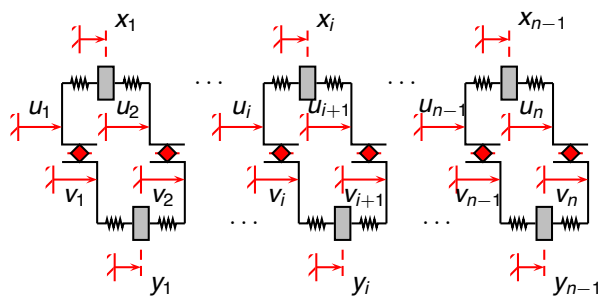
As the loading evolves, multiple slip intervals are developed...

slip interfaces initiate at force reversals and move into the material from the free end





- each element is assumed to be identical, with a mass m, and a stiffness k respectively;
- f_i and g_i, $i = 1, \ldots, n-1$ represent the shear loading applied to the masses;
- f_0 and g_0 (f_n and g_n) describe the forces acting on the left (right) edge of the interface
- each interface is described through the frictional force σ_i.



From the the symmetry of the system, the following coordinates can be identified

$$w_i = \frac{x_i + y_i}{2}, \qquad z_i = \frac{x_i - y_i}{2}, \qquad p_i = \frac{u_i + v_i}{2}, \qquad q_i = \frac{u_i - v_i}{2},$$

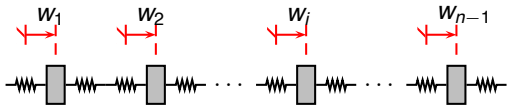
The equations on w_i can be reduced to

$$m\,\ddot{w}_1 + \frac{k}{2}\left(w_1 - w_2\right) = \left(\frac{f_0 + g_0}{2}\right) + \left(\frac{f_1 + g_1}{2}\right),$$

$$m\,\ddot{w}_i + \frac{k}{2}\left(-w_{i-1} + 2\,w_i - w_{i+1}\right) = \left(\frac{f_i + g_1}{2}\right) \qquad i = 2, \ldots, n-2$$

$$m\,\ddot{w}_{n-1} + \frac{k}{2}\left(-w_{n-2} + w_{n-1}\right) = \left(\frac{f_{n-1} + g_{n-1}}{2}\right) + \left(\frac{f_n + g_n}{2}\right).$$

Equivalent to the response of an elastic chain



The dissipation in the system arises solely from the equations on z_i

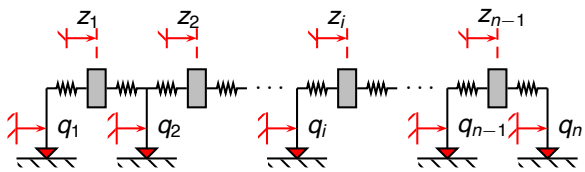
$$m\,\ddot{z}_i + k\left((z_i - q_i) + (z_i - q_{i+1})\right) = \frac{f_i - g_i}{2},$$

with

$$\sigma_1 = -\left(\frac{f_0 - g_0}{2}\right) - k\,(z_1 - q_1),$$

$$\sigma_i = -k\,\left((z_{i-1} - q_i) + (z_i - q_i)\right) \qquad i = 2, \ldots, n-1,$$

$$\sigma_n = -\left(\frac{f_n - g_n}{2}\right) - k\,(z_{n-1} - q_n).$$



If the overall static stiffness and total mass of the chain are held fixed as $K_{\rm eq}$ and $M_{\rm eq}$ respectively, then the inter-element stiffness and mass can be represented as

$$k = (n-1)\,K_{\rm eq}, \qquad\qquad m = \frac{M_{\rm eq}}{2\,(n-1)}.$$

If the lowest characteristic frequency of the interface scales as $\omega_{\rm c} = \sqrt{K_{\rm eq}/M_{\rm eq}}$, then the largest characteristic frequency scales as

$$\omega_{\rm max} = \sqrt{\frac{2\,k}{m}} = 2\,(n-1)\sqrt{\frac{K_{\rm eq}}{M_{\rm eq}}} = 2\,(n-1)\,\omega_{\rm c}.$$

Approximate the response of the elastic chain with a Galerkin method

- keep only the linear vibrational modes whose characteristic times are comparable to the timescales of the surrounding structure;
- depends on the structure as well as the forcing—longer timescales require, in general, fewer modes.

Retaining only the lowest s linear modes for the elastic chain, denoted as ϕ_i, $i = 1, \ldots, s$, the response of this component is then given as

$$w_i(t) = \sum_{j=1}^{s} W_j(t)\,[\phi_j]_i,$$

where

$$\hat{\boldsymbol{M}}\,\ddot{\boldsymbol{W}} + \hat{\boldsymbol{K}}\,\boldsymbol{W} = \hat{\boldsymbol{f}}(t),$$

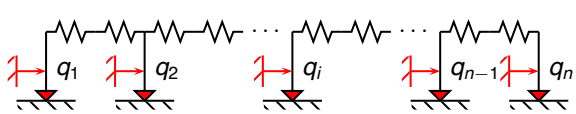
$$[\hat{\boldsymbol{M}}]_{jk} = \phi_j^T \boldsymbol{M} \phi_k, \qquad [\hat{\boldsymbol{K}}]_{jk} = \phi_j^T \boldsymbol{K} \phi_k, \qquad [\hat{\boldsymbol{f}}(t)]_j = \phi_j^T \boldsymbol{f}(t).$$

In the series-series Iwan chain, the dissipation can be accurately captured by neglecting the mass in each Iwan element—effectively solving for the quasistatic response (Kim and Kwak, 1996; Berger et al., 2000; Cocu et al., 1996)

$$k\,(q_1 - q_2) = (f_0 - g_0) + \left(\frac{f_1 - g_1}{2}\right) + 2\,\sigma_1,$$

$$k\,(-q_{i-1} + 2\,q_i - q_{i+1}) = \left(\frac{f_{i-1} - g_{i-1}}{2}\right) + \left(\frac{f_i - g_i}{2}\right) + 2\,\sigma_i \qquad i = 2, \ldots$$

$$k\,(-q_{n-1} + q_n) = (f_n - g_n) + \left(\frac{f_{n-1} - g_{n-1}}{2}\right) + 2\,\sigma_n,$$



Direct time-dependent loading

$$f_0(t) = 0, \qquad f_n(t) = F_0 \ \sin(\omega\, t),$$
$$g_0(t) = -F_0 \ \sin(\omega\, t), \qquad g_n(t) = 0.$$

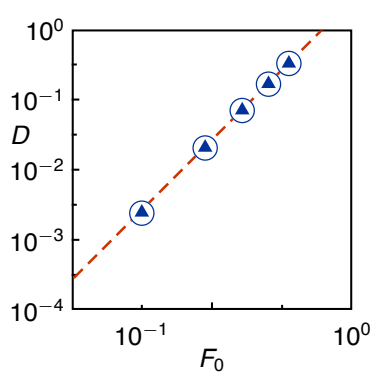
The relative displacement across the interface

$$\Delta_1 = u_n - v_1 = \left(w_{n-1} - w_1\right) + \left(q_n + q_1\right) + \left[\frac{(f_n + g_n) - (f_0 + g_0)}{2\ k}\right],$$

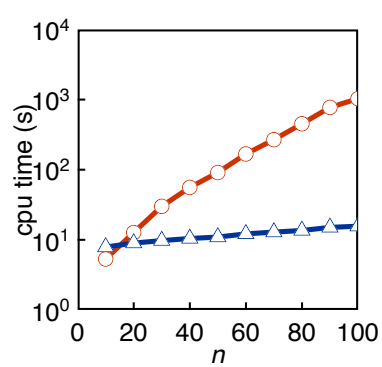
The mass and stiffness are chosen as

$$k = (n-1), \qquad m = \frac{1}{2\,(n-1)},$$

so that the total mass and overall equivalent stiffness for the interface model are unity, that $K_{\text{eq}} = 1$ and $M_{\text{eq}} = 1$.

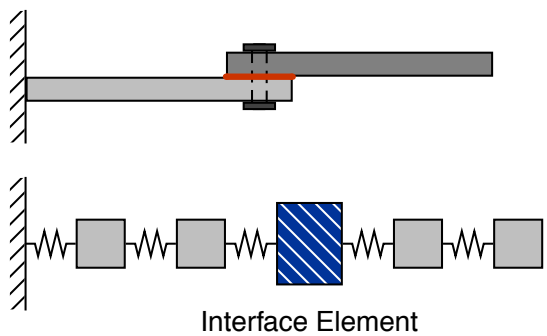


$n = 40$, $\omega = \pi/15$
circles: original
triangles: reduced
dashed: continuum
$D = 8/3\,(F_0)^3$



$F_0 = 0.30$, $\omega = \pi/15$
circles: original
triangles: reduced

Incorporate the interface model into an elastic beam



Interface Element

The ℓth mass is replaced by an Iwan interface element (Song et al., 2004).

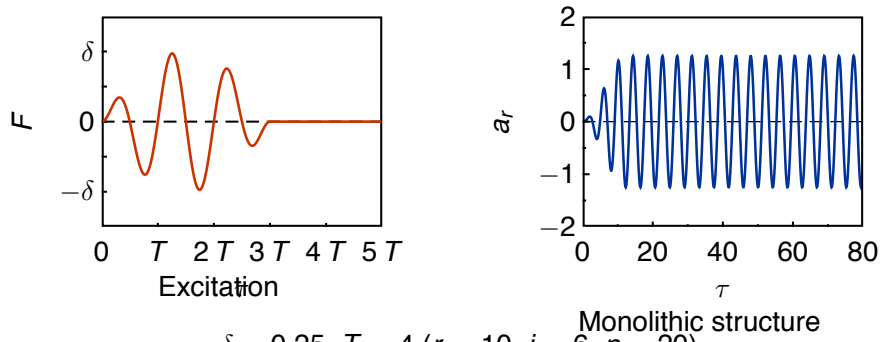
Near the interface the equations governing $a_{\ell-1}$ and $a_{\ell+1}$ become

$$\begin{aligned} \frac{M}{r}\,\ddot{a}_{\ell-1} + K\,r\,(a_{\ell-1} - a_{\ell-2}) &= -g_0(t), \\ \frac{M}{r}\,\ddot{a}_{\ell+1} + K\,r\,(-a_{\ell+2} + a_{\ell+1}) &= -f_n(t), \end{aligned}$$

where $g_0(t)$ and $f_n(t)$ represent the coupling between the Iwan interface element and the surrounding chain. These forces are described as

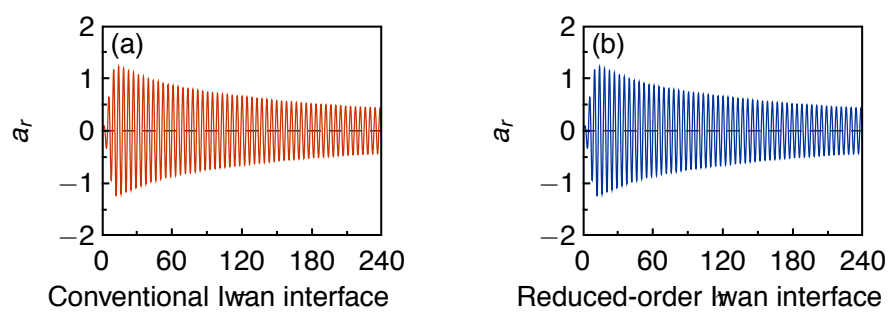
$$\begin{aligned} g_0(t) &= 2\,K\,r\,(a_{\ell-1} - v_1) = \frac{2\,K\,r}{\left(1 + \frac{1}{n-1}\right)}\,\Big(a_{\ell-1} - (w_1 - q_1)\Big)\,, \\ f_n(t) &= 2\,K\,r\,(a_{\ell+1} - u_n) = \frac{2\,K\,r}{\left(1 + \frac{1}{n-1}\right)}\,\Big(a_{\ell-1} - (w_1 + q_n)\Big)\,, \end{aligned}$$

and $f_0(t) = g_n(t) \equiv 0$. The quantities w_1, q_1, w_{n-1}, and q_n represent the elastic and dissipative coordinates used within the interface model described above, and are valid when considering either the conventional, or the massless Iwan interface.



Excitation Monolithic structure

$\delta = 0.25,\ T = 4\ (r = 10,\ j = 6,\ n = 20)$



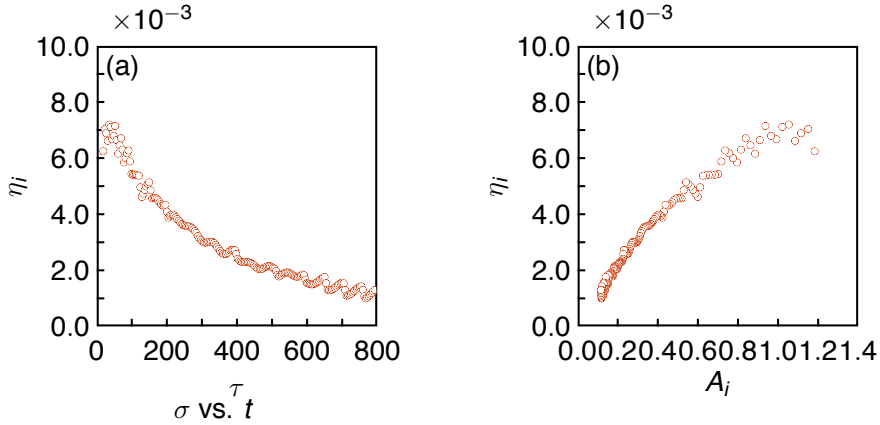
Conventional Iwan interface Reduced-order Iwan interface

$s = 4$

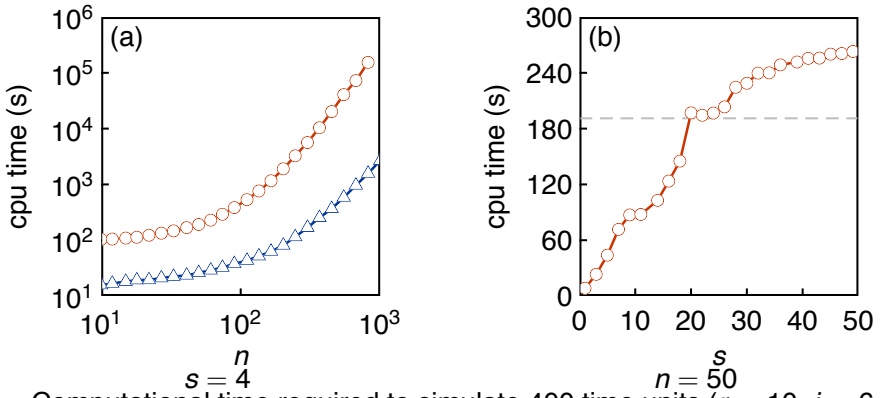
$\delta = 0.25,\ \mu = 1,\ T = 4\ (r = 10,\ j = 6,\ n = 20)$

Exponential Decay Rate

$$\sigma(t_i) = \frac{\ln(A_{i+1}) - \ln(A_{i-1})}{t_{i+1} - t_{i-1}}.$$



Exponential decay rate σ as identified from simulation ($r = 10$, $j = 6$;



Computational time required to simulate 400 time units ($r = 10$, $j = 6$, $\delta = 0.25$, $\mu = 1.00$)

- Incorporate this model into a finite element formulation;
- Determine the appropriate loadings on this interface model.

Berger, E. J., Begley, M. R., Mahajani, M., 2000. Structural dynamic effects on interface response: Formulation and simulation under partial slipping conditions. J. Appl. Mech.—T. ASME 67, 785–792.

Cocu, M., Pratt, E., Raous, M., 1996. Formulation and approximation of quasistatic frictional contact. Intl. J. Engr. Sci. 34 (7), 783–798.

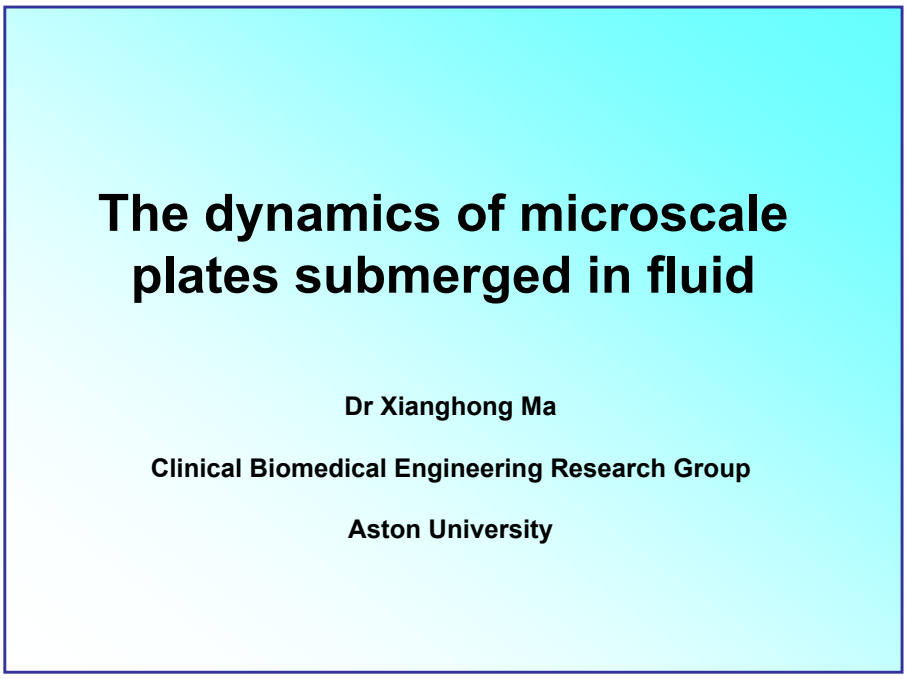
Kim, J., Kwak, B., 1996. Dynamic analysis of two-dimensional frictional contact by linear complementarity problem formulation. Intl. J. Solids Struct. 33, 4605–4624.

Menq, C.-H., Bielak, J., Griffin, J. H., 1986a. The influence of microslip on vibratory response, part I: A new microslip model. J. Sound Vib. 107 (2), 279–293.

Menq, C.-H., Griffin, J. H., Bielak, J., 1986b. The influence of microslip on vibratory response, part II: A comparison with experimental results. J. Sound Vib. 107 (2), 295–307.

Segalman, D. J., 2002. A four-parameter Iwan model for lap-type joints. Tech. Rep. SAND2002–3828, Sandia National Laboratories.

Song, Y., Hartwigsen, C. J., Bergman, L. A., Vakakis, A. F., 2002. A phenomenological model of a bolted joint. Presented at the 15th

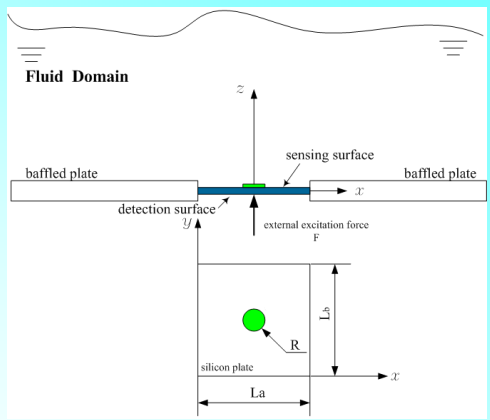


D.3.2 Slide Presentation of Xianghong Ma, Aston University: *The Dynamics of Microscale Plates Submerged in Fluid*

Dynamic Modelling

FIRST analytical solution with considerations of:

- **Distributed mass loading**
- **Fluid interaction**



$$D\nabla^4 w(x,y,t) + \rho_p h \frac{\partial^2 w(x,y,t)}{\partial t^2} + \overline{m}_c \mathcal{H}(x,y) \frac{\partial^2 w(x,y,t)}{\partial t^2} = p(x,y,t)$$

$$D\nabla^4 w(x,y,t) + \rho_p h \frac{\partial^2 w(x,y,t)}{\partial t^2} + \overline{m}_c \mathcal{H}(x,y) \frac{\partial^2 w(x,y,t)}{\partial t^2} = p(x,y,t)$$

Heaviside function: $\mathcal{H}(x,\ y) = \{H[x-(x_c-R)] - H[x-(x_c+R)]\} \cdot \left\{H\left[y-(y_c-\sqrt{R^2-x^2})\right] - H\left[y-(y_c+\sqrt{R^2-x^2})\right]\right\}$

The classical thin plate theory is valid when:

$$\frac{\omega h}{\pi c_s} < 0.1$$

C_s is material shear wave speed

The displacement: a summation of a series of eigenfunctions:

$$w(x,\ y,\ t) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} W_{mn} X_m(x)\, Y_n(y) \cdot \theta(t)$$

The effect of cross-modal coupling between fluid and plate:

1. Acoustic reactance force, generate inertial forces on the plate
2. Resistive force, appears as damping/energy dissipation due to acoustic radiation

Simplified Solution using Rayleigh-Ritz method

Assume fluid is inviscid (μ=0) and incompressible (c=∞)

$$\nabla^2 \Phi(x,\ y,\ z,\ t) = \frac{1}{c^2}\frac{\partial^2 \Phi(x,\ y,\ z,\ t)}{\partial t^2}$$

For steady state harmonic motion: $\Phi(x,\ y,\ z,\ t) = \phi(x,\ y,\ z)\mathrm{e}^{-\mathrm{i}\omega t}$.

$$\phi(x,\ y,\ z) = \frac{\mathrm{i}\omega}{2\pi}\iint_S \frac{w(\xi,\eta)\exp\left(\mathrm{i}k\sqrt{(x-\xi)^2+(y-\eta)^2+z^2}\right)}{\sqrt{(x-\xi)^2+(y-\eta)^2+z^2}}\,\mathrm{d}\xi\,\mathrm{d}\eta.$$

The fluid-induced inertial effect is proportional to the kinetic energy of fluid:

$$T_\mathrm{f} = -\frac{1}{2}\rho_\mathrm{f}\int_{-\infty}^{\infty}\int_{-\infty}^{\infty} \frac{\partial \phi(x,\ y,\ z)}{\partial z}\Big|_{z=0} \phi(x,\ y,\ 0)\ \mathrm{d}x\,\mathrm{d}y$$

$$T_\mathrm{f} = \frac{\rho_\mathrm{f}\omega^2}{4\pi}\iint_S\iint_S \frac{w(x,\ y)w(\xi,\eta)}{\sqrt{(x-\xi)^2+(y-\eta)^2}}\ \mathrm{d}x\,\mathrm{d}y\,\mathrm{d}\xi\,\mathrm{d}\eta$$

Solution using Rayleigh-Ritz method

Kinetic energy includes that of the plate, the fluid and the attached mass

$$\omega_c^2 = \frac{U_p}{T_p^* + T_f^* + T_m^*}$$

$$U_p = \frac{D}{2}\iint_S \left\{ \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2}\right)^2 - 2(1-\nu)\left[\frac{\partial^2 w}{\partial x^2}\frac{\partial^2 w}{\partial y^2} - \left(\frac{\partial^2 w}{\partial x \partial y}\right)^2\right]\right\} \, dx\, dy$$

$$T_p^* = \frac{1}{2}\rho_p h \iint_S w^2(x, y) \, dx\, dy \qquad T_f^* = T_f/\omega^2$$

$$T_m^* = \frac{1}{2}\iint_S \bar{m}_c \mathcal{H}(x, y) w^2(x, y) \, dx\, dy.$$

The total energy: $V = U_p - \omega^2 (T_p^* + T_f^* + T_m^*)$

Eigenfunctions can be obtained by minimizing V wrt unknown deflection coefficient:

$$\frac{\partial V}{\partial W_{qr}} = 0 \Rightarrow \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} \{ U_{p,mnqr} - \omega^2 (T_{p,mnqr}^* + T_{f,mnqr}^* + T_{m,mnqr}^*)\} W_{mn} = 0.$$

Derive the eigenfunctions:

$$\sum_{m=1}^{\infty}\sum_{n=1}^{\infty} \{ U_{p,mnqr} - \omega^2 (T_{p,mnqr}^* + T_{f,mnqr}^* + T_{m,mnqr}^*)\} W_{mn} = 0.$$

$$\boxed{\{[\boldsymbol{A}] - \omega^2 [\boldsymbol{I}]\}\{x\} = 0} \qquad \boxed{\boldsymbol{A} = \boldsymbol{M}^{-1}\boldsymbol{K}}$$

$$K_{ij} = U_{p,mnqr}$$
$$M_{ij} = T_{p,mnqr}^* + T_{f,mnqr}^* + T_{m,mnqr}^*$$
$$i = l(q-1) + r, \quad j = l(m-1) + n, \quad l \in \mathbb{N}^+$$

$$U_{p,mnqr} = \frac{D}{2}\iint_S \{ \ddot{X}_m(x)\ddot{X}_q(x) Y_n(y) Y_r(y) + X_m(x) X_q(x) \ddot{Y}_n(y) \ddot{Y}_r(y)$$
$$+ 2\nu \ddot{X}_m(x) X_q(x) Y_n(y) \ddot{Y}_r(y) + 2(1-\nu)\dot{X}_m(x)\dot{X}_q(x)\dot{Y}_n(y)\dot{Y}_r(y)\} dx dy$$

$$T_{p,mnqr}^* = \frac{1}{2}\rho_p h \iint_S X_m(x) X_q(x) Y_n(y) Y_r(y) \, dx\, dy$$

$$T_{f,mnqr}^* = \frac{\rho_f}{4\pi}\iint_S\iint_S \frac{X_m(x) Y_n(y) X_q(\xi) Y_r(\eta)}{\sqrt{(x-\xi)^2 + (y-\eta)^2}} \, dx\, dy\, d\xi\, d\eta$$

Challenge: integral singularity in acoustic impedance

$$T_{m,mnqr}^* = \frac{1}{2}\bar{m}_c \iint_S X_m(x) X_q(x) Y_n(y) Y_r(y) \mathcal{H}(x, y) \, dx\, dy.$$

Realistic model: fluid acoustic radiation and viscous effect

$$D\left(\frac{\partial^4 w}{\partial x^4} + 2\frac{\partial^4 w}{\partial x^2 y^2} + \frac{\partial^4 w}{\partial y^4}\right) + \rho_p h \frac{\partial^2 w}{\partial t^2} = F(x,y,t)$$

$$F(x,y) = F_{ex}\delta(x-x_0)\delta(y-y_0) + F_{hydro}(x,y,0)$$

$$F_{hydro}(x,y,0,t) = F_{hydro}(x,y,0-,t) - F_{hydro}(x,y,0+,t)$$

$$F_{hydro}(x,y,0-,t) = -F_{hydro}(x,y,0+,t)$$

Under Non-slip condition, hydrodynamic force depends on the fluid stress tensor:

$$F_{hydro}(x,y,0+) = -\sigma_z + \frac{h}{2}\left(\frac{\partial \tau_{zx}}{\partial x} + \frac{\partial \tau_{zy}}{\partial y}\right)$$

Substitute fluid velocity into fluid stress tensors:

$$\sigma_z = -2\mu\nabla^2\phi + \rho_{f0}\dot{\phi} + 2\mu\left(\frac{\partial^2\phi}{\partial z^2} + \frac{\partial^2\psi_y}{\partial x\partial z} + \frac{\partial^2\psi_x}{\partial y\partial z}\right)$$

$$\tau_{zx} = \mu\left(2\frac{\partial^2\phi}{\partial x\partial z} + \frac{\partial^2\psi_z}{\partial y\partial z} - \frac{\partial^2\psi_x}{\partial y\partial x} - \frac{\partial^2\psi_y}{\partial z^2} + \frac{\partial^2\psi_y}{\partial x^2}\right)$$

$$\tau_{zy} = \mu\left(2\frac{\partial^2\phi}{\partial y\partial z} + \frac{\partial^2\psi_y}{\partial x\partial y} - \frac{\partial^2\psi_z}{\partial x\partial z} - \frac{\partial^2\psi_x}{\partial y^2} + \frac{\partial^2\psi_x}{\partial z^2}\right)$$

The Navier-Stokes equation for the motion of a viscous compressible fluid:

$$\rho_{f0}\frac{\partial \mathbf{v}}{\partial t} - \mu\nabla^2\mathbf{v} - (\mu+\mu^v)\nabla(\nabla\cdot\mathbf{v}) + \nabla p = 0$$

The motion also satisfies the continuity equation: $\frac{\partial \rho_f}{\partial t} + \rho_{f0}\nabla\cdot\mathbf{v} = 0$

And momentum equation: $\frac{\partial p}{\partial \rho_f} = c^2$

The velocity field: $\mathbf{v} = \nabla\Phi + \nabla\times\Psi$ $\quad \nabla\cdot\Psi = 0$

Consider harmonic motion:

$$\Phi(x,y,z,t) = \phi(x,y,z)e^{-i\omega t}$$
$$\Psi(x,y,z,t) = \psi(x,y,z)e^{-i\omega t}$$

$$\nabla^2\phi + k_l^2\phi = 0 \qquad k_l^2 = \frac{\omega^2/c^2}{1-4i\mu\omega/3\rho_{f0}c^2}$$

$$\nabla^2\psi + k_s^2\psi = 0 \qquad k_s^2 = \frac{i\rho_{f0}\omega}{\mu}$$

Apply double Fourier transforms:

$$\phi(x,y,z) = \frac{1}{4\pi^2}\iint_{-\infty}^{\infty} A\cdot\exp\left(ik_x x + ik_y y + \sqrt{k_l^2 - k_x^2 - k_y^2}\cdot z\right)dk_x dk_y$$

$$\psi(x,y,z) = \frac{1}{4\pi^2}\iint_{-\infty}^{\infty} B\cdot\exp\left(ik_x x + ik_y y + i\sqrt{k_s^2 - k_x^2 - k_y^2}\cdot z\right)dk_x dk_y$$

Apply non-slip BCs at the fluid-plate interface:

$$\frac{\partial \mathbf{u}^p}{\partial t} = \mathbf{v}^f, \quad \vec{\sigma}^p = \vec{\sigma}^f$$

Expand it in the Cartesian coordinates,

$$v_x^f = v_x^p|_{z=0}, \quad v_y^f = v_y^p|_{z=0}, \quad v_z^f = v_z^p|_{z=0}$$

$$v_x^f = \frac{\partial \phi}{\partial x} + \frac{\partial \psi_z}{\partial y} - \frac{\partial \psi_y}{\partial z}$$
$$v_y^f = \frac{\partial \phi}{\partial y} + \frac{\partial \psi_x}{\partial z} - \frac{\partial \psi_z}{\partial x}$$
$$v_z^f = \frac{\partial \phi}{\partial z} + \frac{\partial \psi_y}{\partial x} - \frac{\partial \psi_x}{\partial y}$$

$$v_x^p = -\frac{h}{2}\frac{\partial^2 w}{\partial x \partial t}$$
$$v_y^p = -\frac{h}{2}\frac{\partial^2 w}{\partial y \partial t}$$
$$v_z^p = \frac{\partial w}{\partial t}$$

$$\nabla \cdot \Psi = 0$$

$$\begin{aligned}
i\sqrt{k_l^2 - k_x^2 - k_y^2} \cdot A + ik_y B_x - ik_x B_y &= \tilde{w} \\
k_x A + \sqrt{k_s^2 - k_x^2 - k_y^2} \cdot B_y + k_y B_z &= \frac{h}{2} k_x \tilde{w} \\
-k_y A + \sqrt{k_s^2 - k_x^2 - k_y^2} \cdot B_x + k_x B_z &= \frac{h}{2} k_x \tilde{w} \\
-k_x B_x - k_y B_y + \sqrt{k_s^2 - k_x^2 - k_y^2} \cdot B_z &= 0
\end{aligned}$$

$$\phi(x,y,z) = \frac{1}{4\pi^2} \iint_{-\infty}^{\infty} A \cdot \exp\left(ik_x x + ik_y y + \sqrt{k_l^2 - k_x^2 - k_y^2} \cdot z\right) dk_x dk_y$$

$$\psi(x,y,z) = \frac{1}{4\pi^2} \iint_{-\infty}^{\infty} B \cdot \exp\left(ik_x x + ik_y y + i\sqrt{k_s^2 - k_x^2 - k_y^2} \cdot z\right) dk_x dk_y$$

The analytical expression of the hydrodynamic force:

$$F_{hydro}(x,y,0) = 2 \cdot \frac{1}{4\pi^2} \iint_{-\infty}^{\infty} T(k_x,k_y)\tilde{w} \exp(ik_x x + ik_y y) dk_x dk_y$$

The essential matrix form of the vibration:

$$\sum_m^{\infty} \sum_n^{\infty} (\mathbf{\Gamma} + i\omega \mathbf{I_{mnqr}}) \{W_{mn}\} = F_{qr} \quad q,r = 1,2,\cdots,\infty$$

Where external force:

$$F_{qr} = \iint_S F_{ex} \delta(x - x_0)\delta(y - y_0) X_q(x) Y_r(y)\, dx\, dy$$

Modal coefficient of the plate stiffness:

$$\mathbf{\Gamma} = M\left[\omega_{mn}^2 - \omega^2\right]\mathbf{E}$$

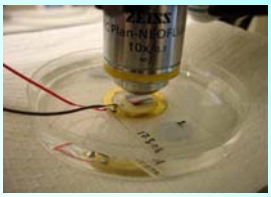
$$\omega_{mn}^2 = \frac{D}{\rho_p h}\left(k_m^4 + 2\frac{\iint_S X_m \ddot{X}_m Y_n \ddot{Y}_n dxdy}{\iint_S X_m^2 Y_n^2 dxdy} + k_n^4\right)$$

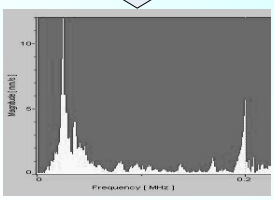
The direct and mutual mode fluid-loading impedance

$$I_{mnqr} = 2 \cdot \frac{1}{4\pi^2} \iint_{-\infty}^{\infty} T(k_x,k_y)\chi_{mn}(k_x,k_y)\chi_{qr}^*(k_x,k_y) dk_x dk_y$$

$$\chi_{mn}(k_x,k_y) = \iint_S X_m(x) Y_n(y) \exp(-i(k_x x + k_y y))$$

The experimental validation





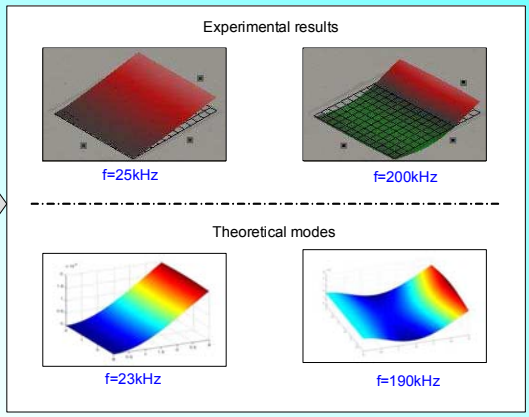
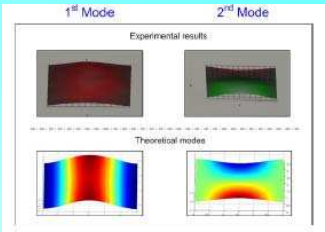


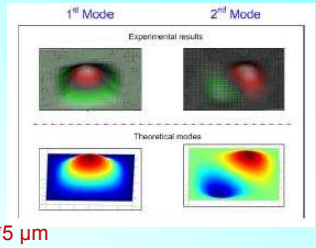
Plate dimension: 294x295X5 μm, cantilever BC

Modes	In Vacuo	In Air Theo.	In Air Expe.	In Water Theo.	In Water Expe.
1st	436.80	433.3	371.87	130.40	117.2
2nd	537.59	534.5	542.19	227.98	278.1
3rd	911.98	909.6	929.06	453.65	405.0

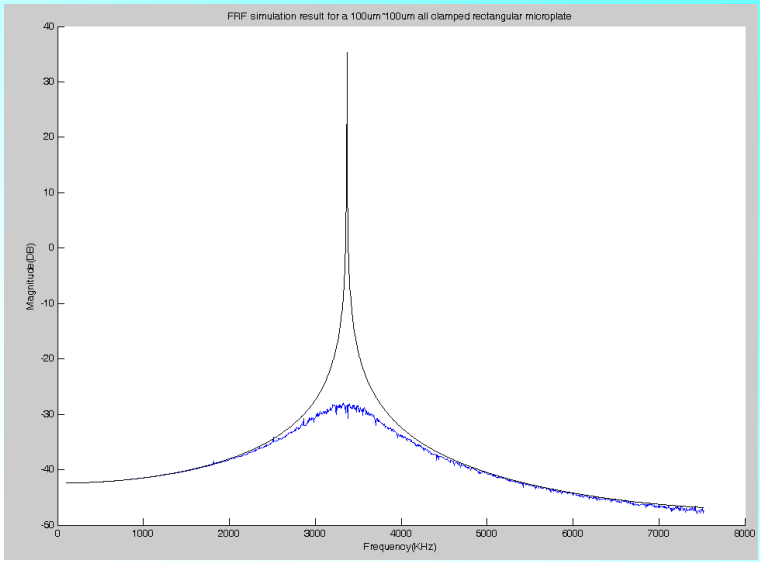


Two opposite edges clamped and the other two edge free, 296μm*309 μm*5 μm

Modes	In Vacuo	In Air Theo.	In Air Expe.	In Water Theo.	In Water Expe.
1st	523.08	519.67	460.0	152.67	153.44
2nd	1065.8	1062.6	1018.1	436.44	445.3
4th	1581.3	1577.8	1542.8	733.78	793.75



All edges clamped, 358μm*360 μm*5 μm

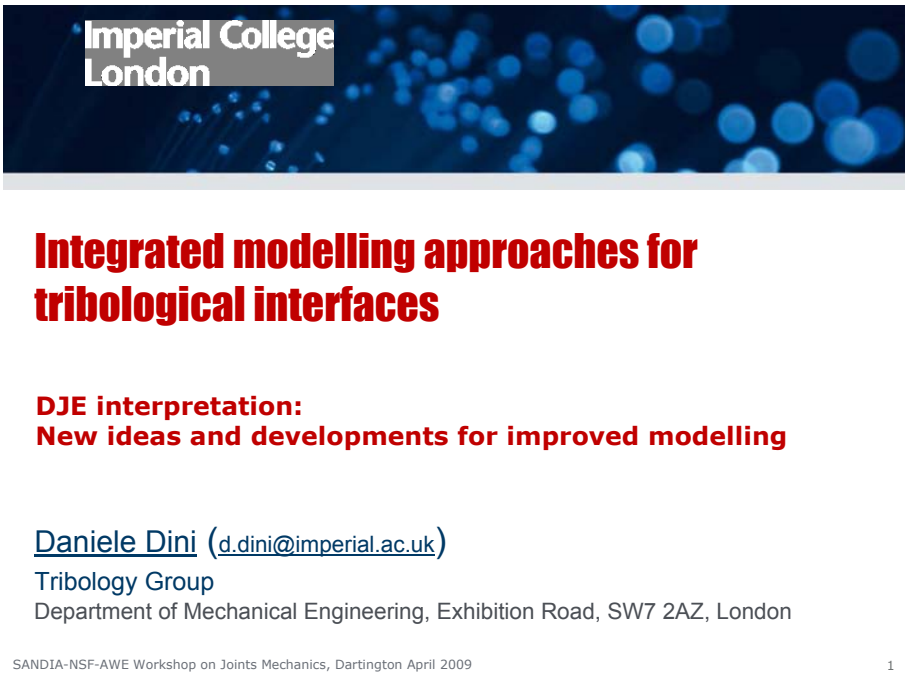


The comparison of the simplified solution and the realistic model solution

Conclusion

- A 3-D theoretical model and its analytical solution for the vibration of microplates in compressible viscous fluid is developed.
- The hydrodynamic force applied by fluid is developed.
- Fluid-loading impedance matrix is derived and used to study the effect of acoustic radiations and viscous loss.
- Experimental results correspond well with analytical simulations.

Thoughts on joints?

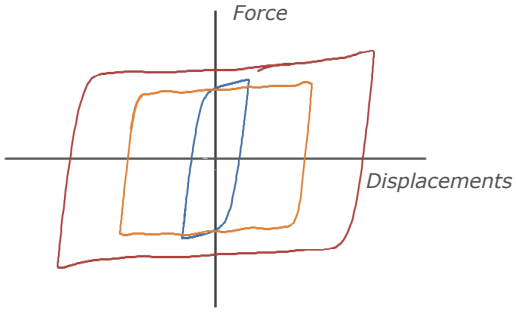


D.3.3 Slide Presentation of Daniele Dini, Imperial College: *Integrated Modelling Approaches for Tribological Interfaces*

Imperial College London

Background 1 – The continuum side

- Frictional hysteresis loops recorded for reciprocating sliding of representative samples of material
- Friction coefficient and tangential contact stiffness obtained from hysteresis trace
- Difficult to predict friction, but…
  - Are results scalable?
  - Can initial curvature (stiffness due to partial slip) be predicted?
  - Can energy dissipation be predicted?

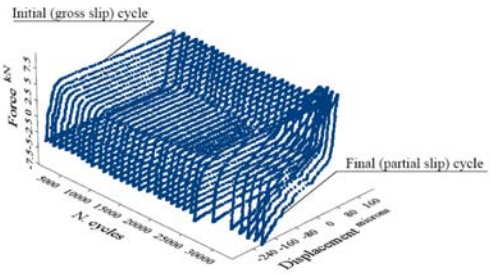


- **Focusing on surface roughness effects**
- **Interested primarily in energy dissipation and tangential contact stiffness**

Imperial College London

Background 1 – The continuum side

- Frictional hysteresis loops recorded for reciprocating sliding of representative samples of material
- Friction coefficient and tangential contact stiffness obtained from hysteresis trace
- Difficult to predict friction, but…
  - Are results scalable?
  - Can initial curvature (stiffness due to partial slip) be predicted?
  - Can energy dissipation be predicted?
- What about history and evolution ???



- **Focusing on surface roughness effects**
- **Interested primarily in energy dissipation and tangential contact stiffness**

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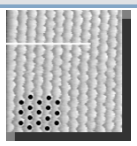
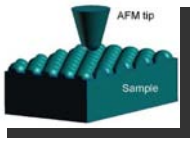
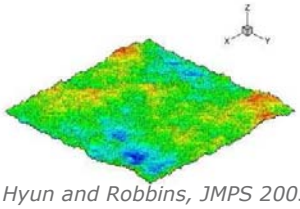
Background 2 – Bridging scales



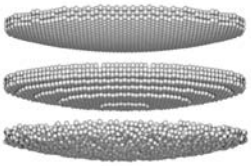
Carpick et al., JOM 2004



AFM supporters 1990's-



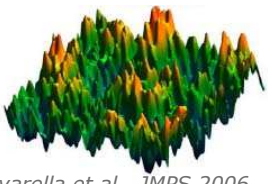
Hyun and Robbins, JMPS 2005



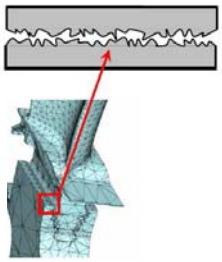
Luan and Robbins, Nature 2005

3D nano- or micro-scale contacts

3D macro-scale contacts



Ciavarella et al., JMPS 2006



Petrov et al., ASME 2004

Where does continuum brake down? Is it question of size or models?

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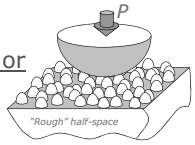
Effect of roughness (Method)

Rough surfaces contact analysis



- Uses either real roughness from optical profilometry or randomised surfaces

- Using Multilevel Multi-integration method (Bradt & Lubrecht; Venner & Lubrecht)
- Coarser grids allow long range influences to pass through Jacobi relaxation process faster, and faster solution
- Good for memory usage; critical for future work on experimental comparison with real surfaces requiring very large grid sizes
- Ciavarella / Jäger method for obtaining partial slip tractions
- Limitations: linear elastic, half-space and Coulomb's friction

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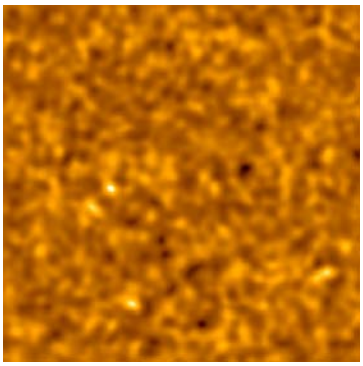
Rough surface generation

- Rough surfaces generated using moving average method to control correlation length
- Template surfaces generated
- Rescaled to give different RMS
- Translated using Johnson curves to give different skewness and kurtosis
- Ensures asperity location remains same and reduces scatter



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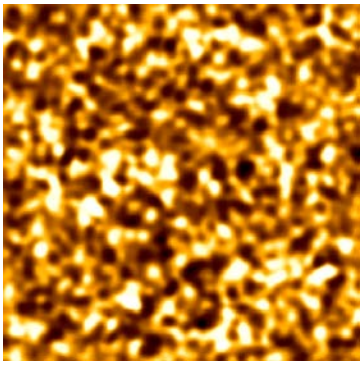

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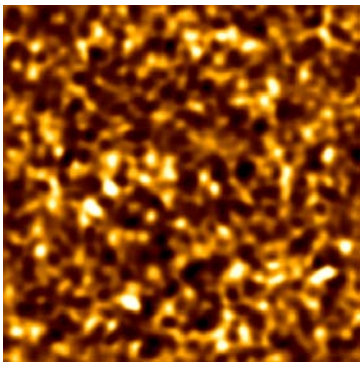

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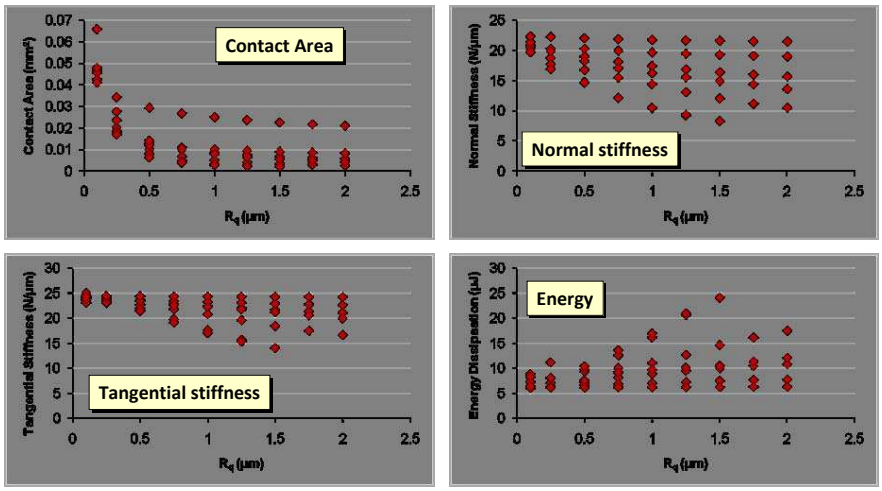
Influence of roughness



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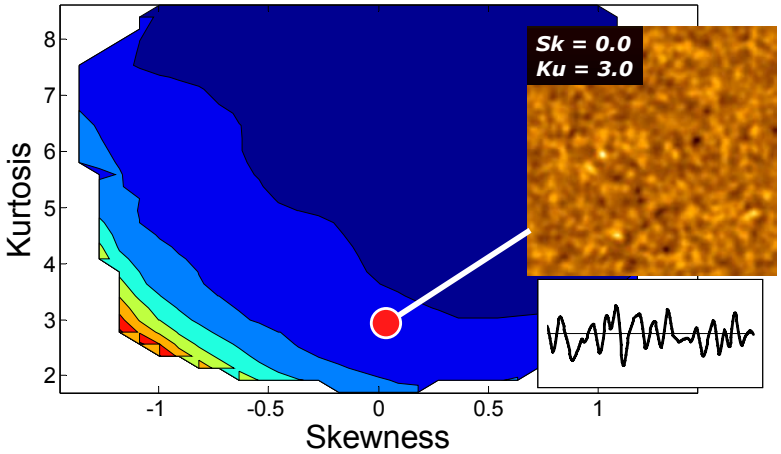
Surface forms – Sk / Ku



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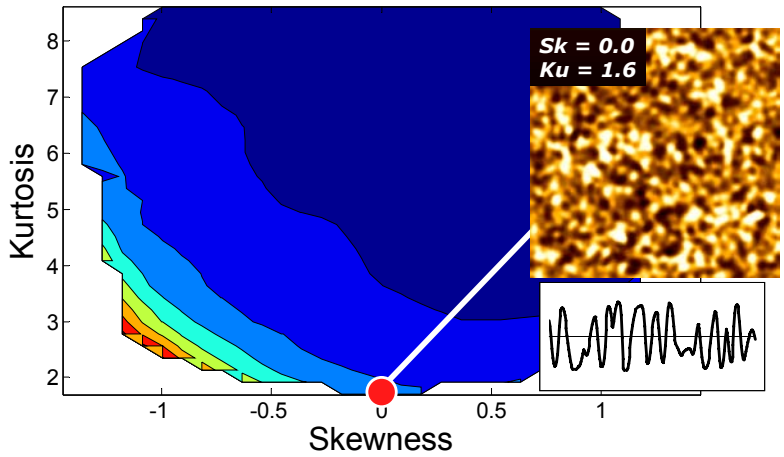
Surface forms – Sk / Ku



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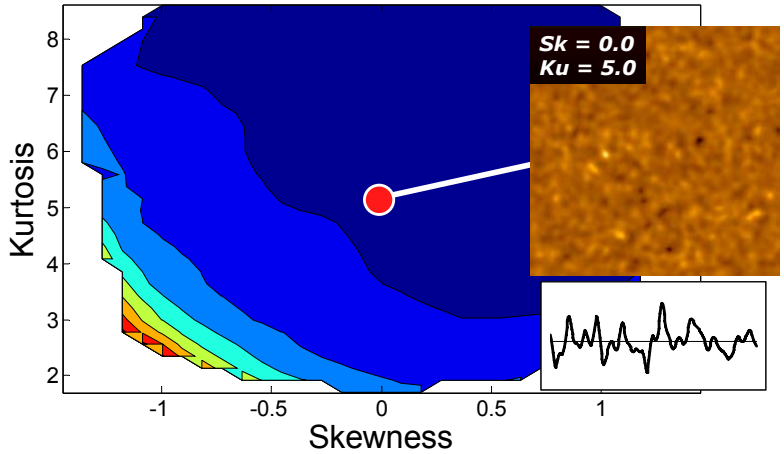
Surface forms – Sk / Ku



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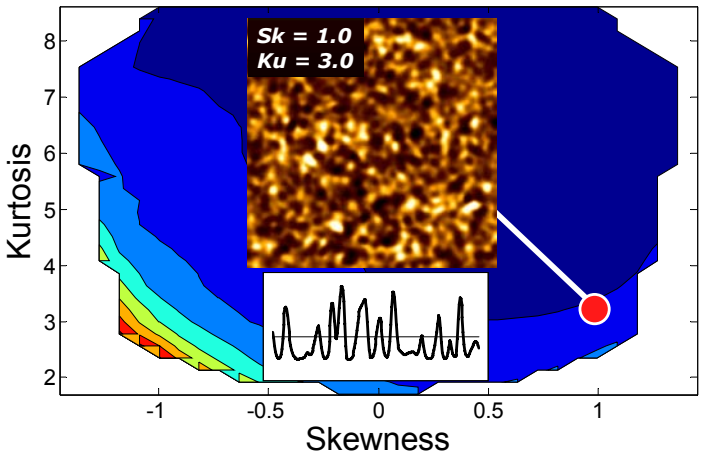
Surface forms – Sk / Ku



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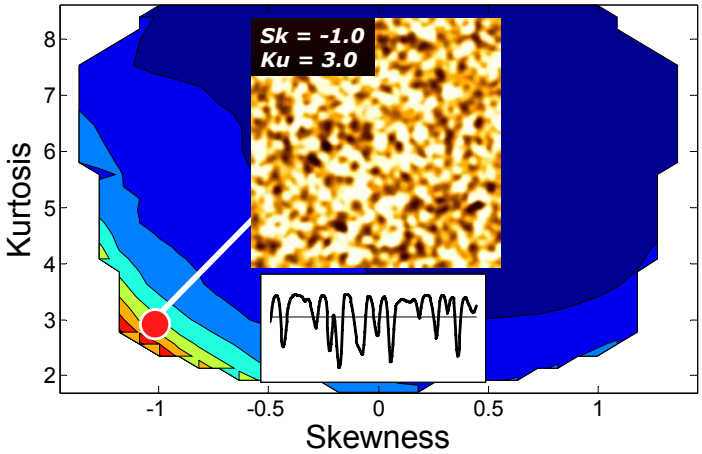
Surface forms – Sk / Ku



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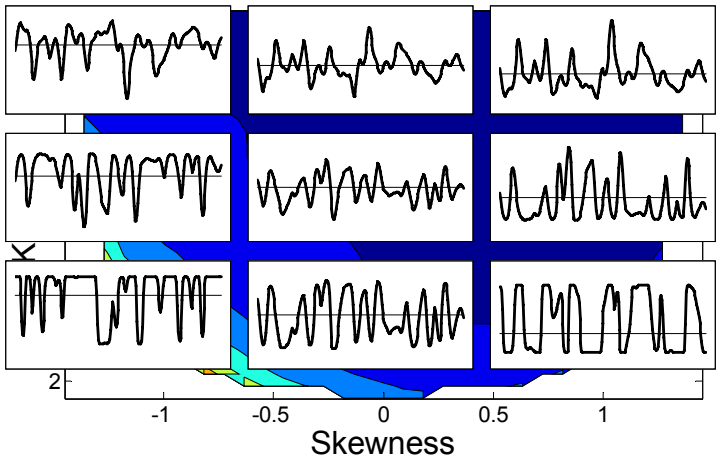
Surface forms – Sk / Ku



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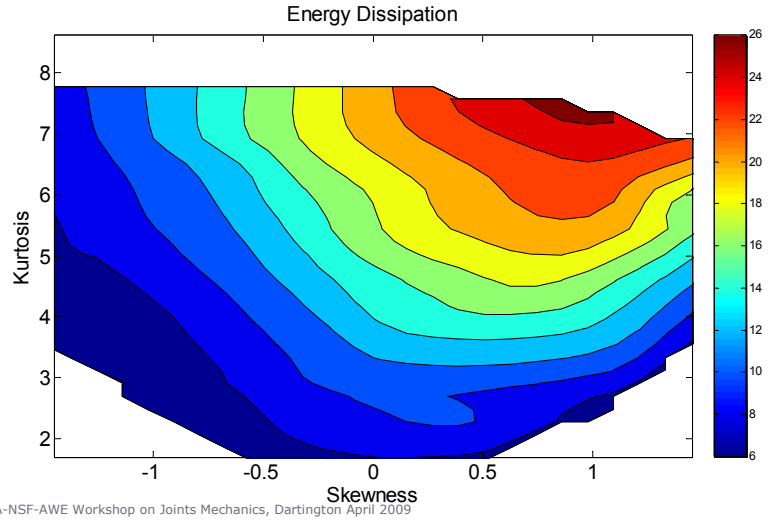
Influence of roughness

CL=10
RMS = 1
Q/fP = 0.8



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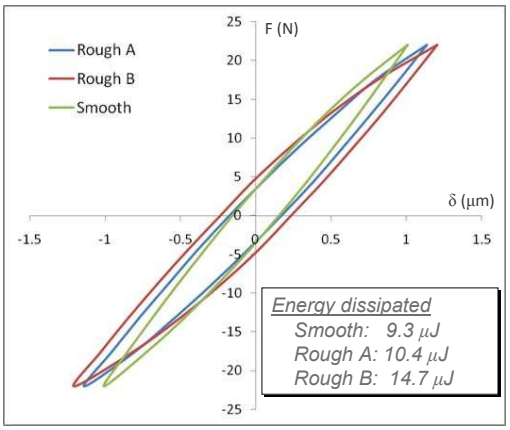
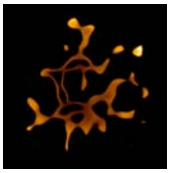
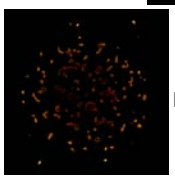
Energy Dissipation in Partial Slip



Smooth



Rough A



Rough B

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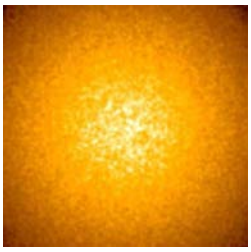
Asperity interaction (two rough surfaces)

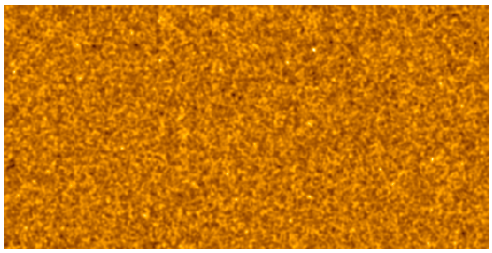
RMS 0.2μm

Top surface Rad 10mm

Load 50N



Top surface



Bottom surface

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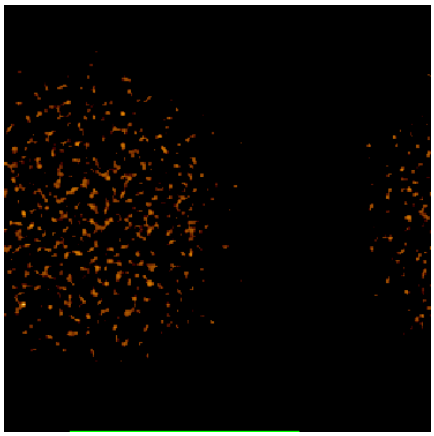
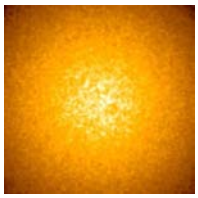
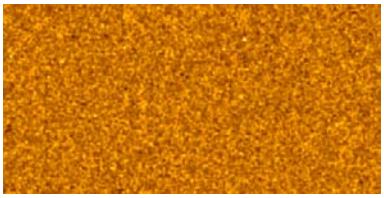
Asperity interaction (two rough surfaces)







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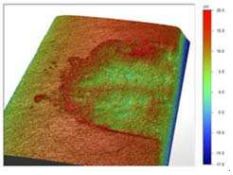
Experimental surface analysis

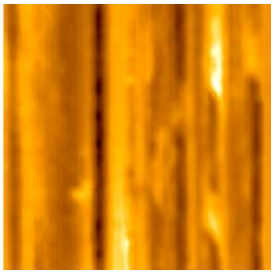
Small section of a Nickel sample, profiled edges

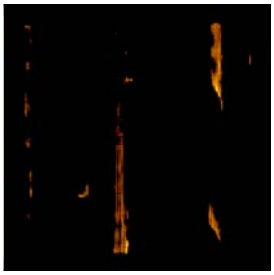
Raw data could not be solved – contact on 2 nodes

Low pass filter applied to remove spikes

Normal load solved satisfactorily







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Open questions 1

How can we make sure that our models are a close representation of the real components?

How do we extract the processes and the parameters which characterise the behaviour of our assemblies? (very strong link to well characterised experiments)

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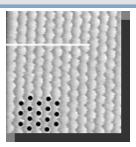
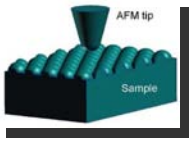
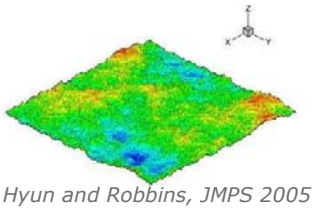
Background 2 – Bridging scales



Carpick et al., JOM 2004

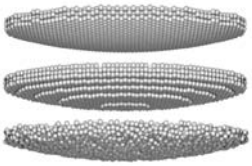


AFM supporters 1990's-



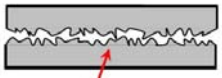
Hyun and Robbins, JMPS 2005

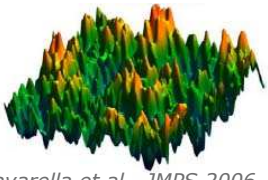
3D nano- or micro-scale contacts



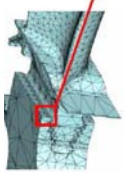
Luan and Robbins, Nature 2005

3D macro-scale contacts





Ciavarella et al., JMPS 2006



Petrov et al., ASME 2004

Where does continuum brake down? Is it question of size or models?

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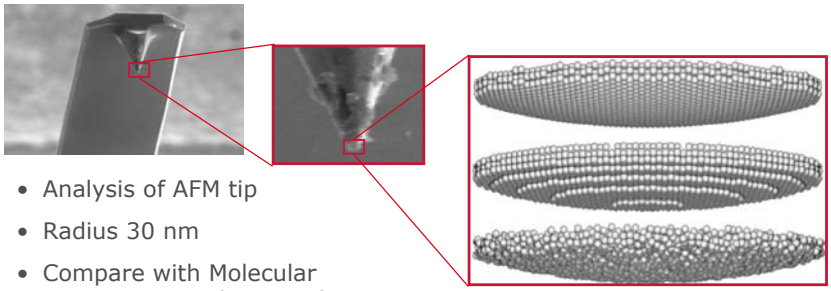
Which Model When? (AFM tip analyses)



- Analysis of AFM tip
- Radius 30 nm
- Compare with Molecular Dynamics Simulations of Luan & Robbins 2006
- Suitability of continuum approach at molecular level
- Surfaces generated using hard sphere model of atoms

B. Luan and M.O. Robbins, 2006, "Contact of single asperities with varying adhesion: Comparing continuum mechanics to atomistic simulations", Physical Review E 74, 026111

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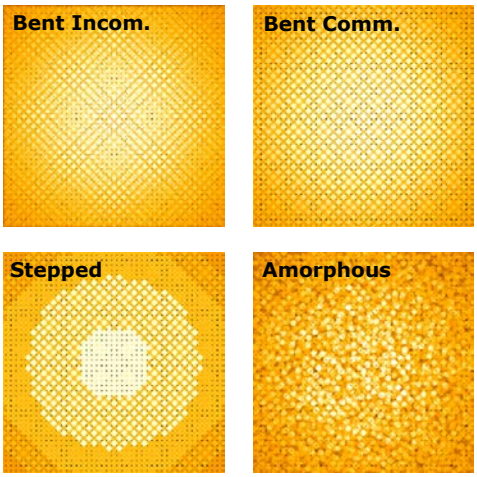
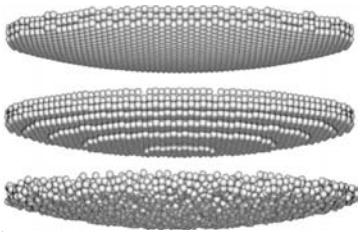
AFM tip profiles





Molecular model

Surface model

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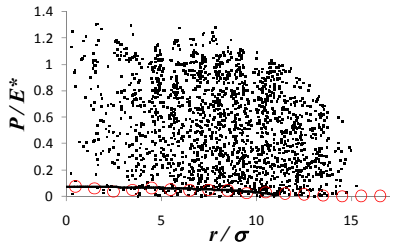
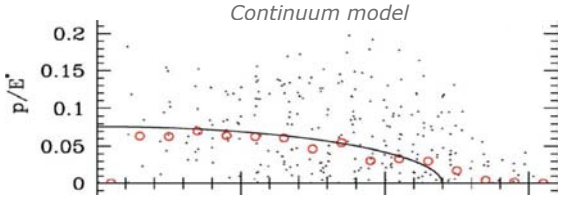
AFM tip results – Contact pressures

Amorphous Tip



Continuum model



Molecular Dynamics

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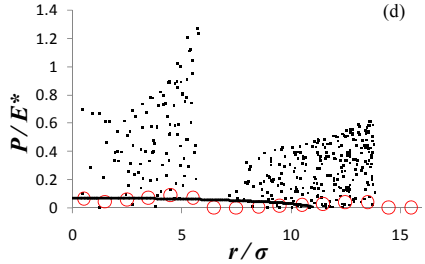
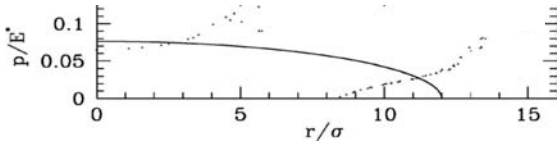
AFM tip results – Contact pressures

Stepped Tip

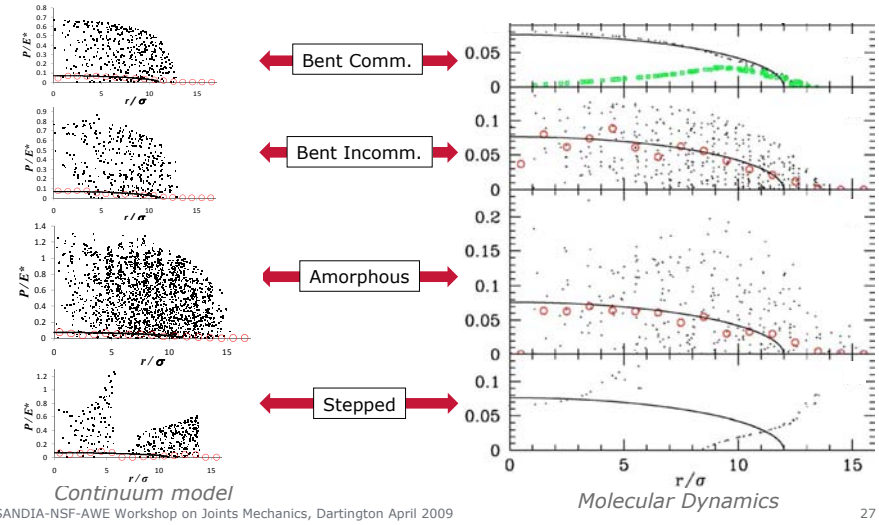


Continuum model



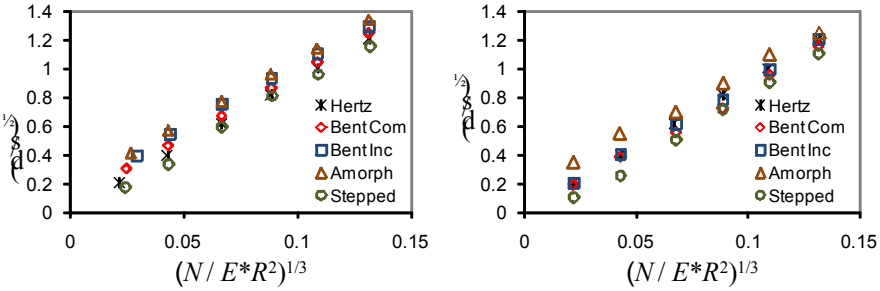
Molecular Dynamics

AFM tip results – Contact pressures



Continuum model *Molecular Dynamics*

AFM tip results – Approach



Continuum model *Molecular Dynamics*

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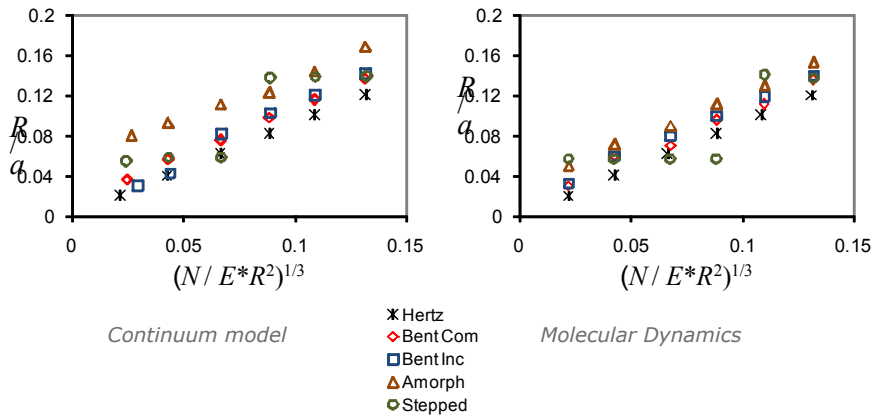
AFM tip results – Contact area



Continuum model *Molecular Dynamics*

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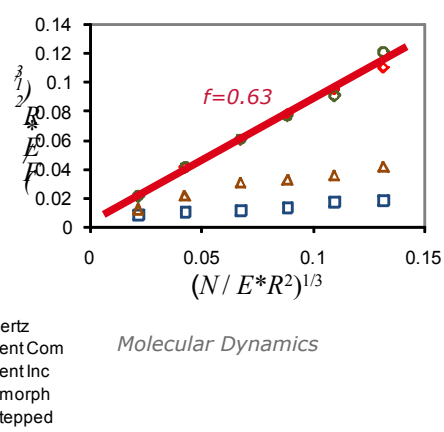
AFM tip results – Friction force

?

Not Available
Friction coefficient assigned



Continuum model *Molecular Dynamics*

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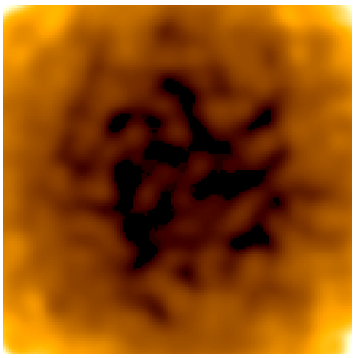
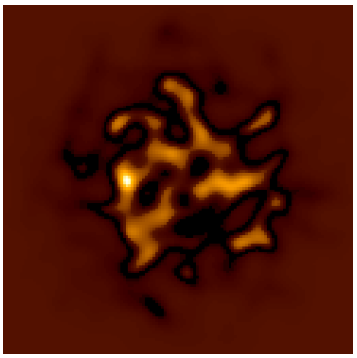
Adhesion – Rough surface (v. small scale)

Surface Separation



Pressures



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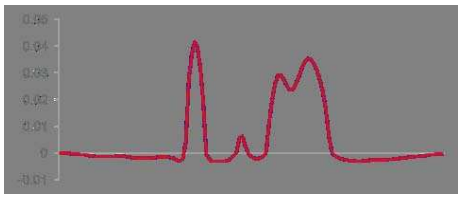
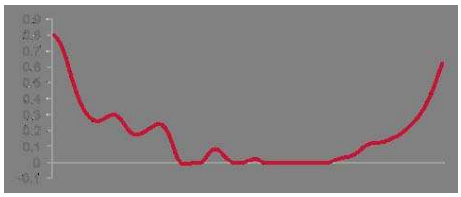
Adhesion – Rough surface (v. small scale)

Pressures



Surface Separation



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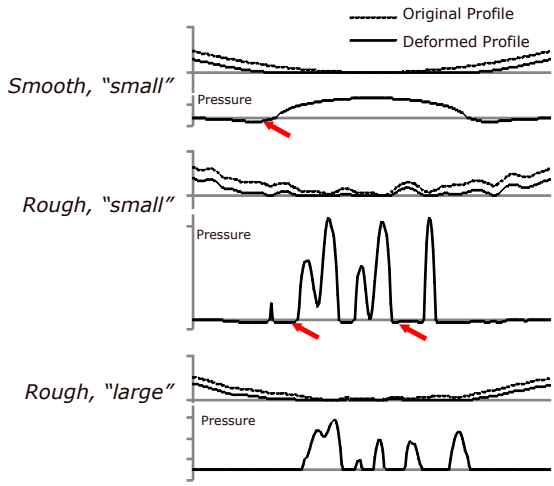
Adhesion – Effect of scale and roughness



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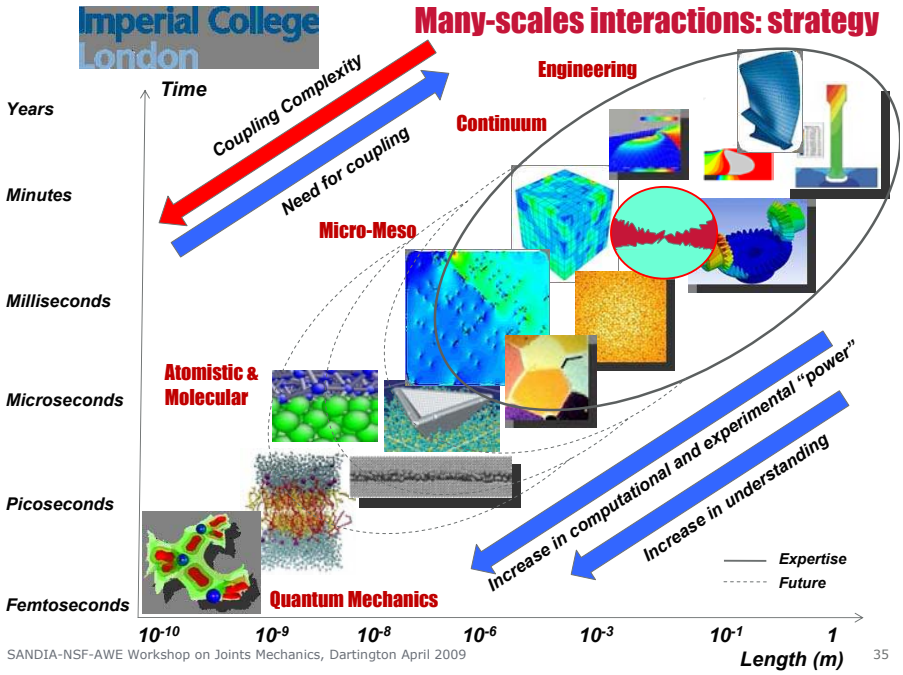
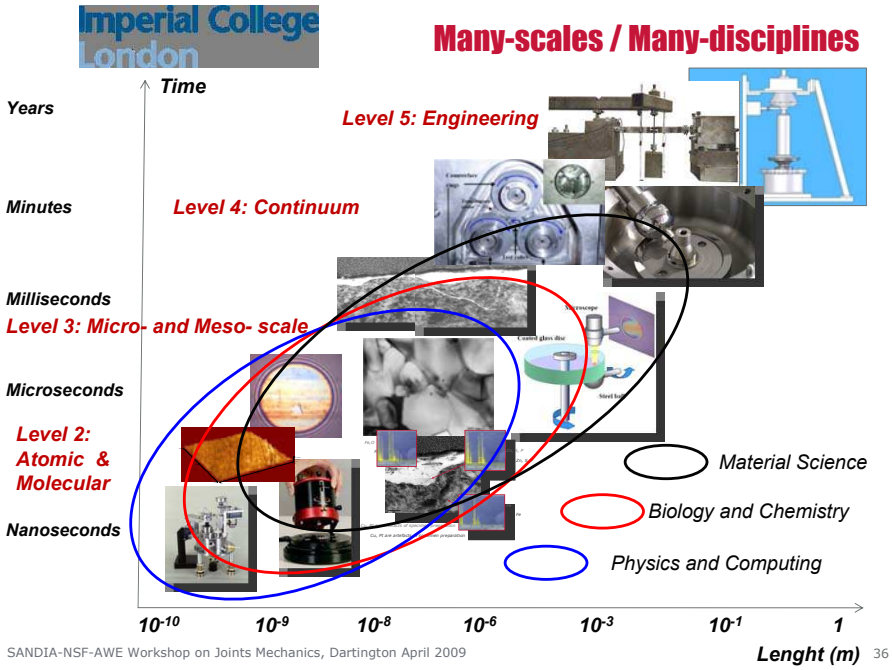
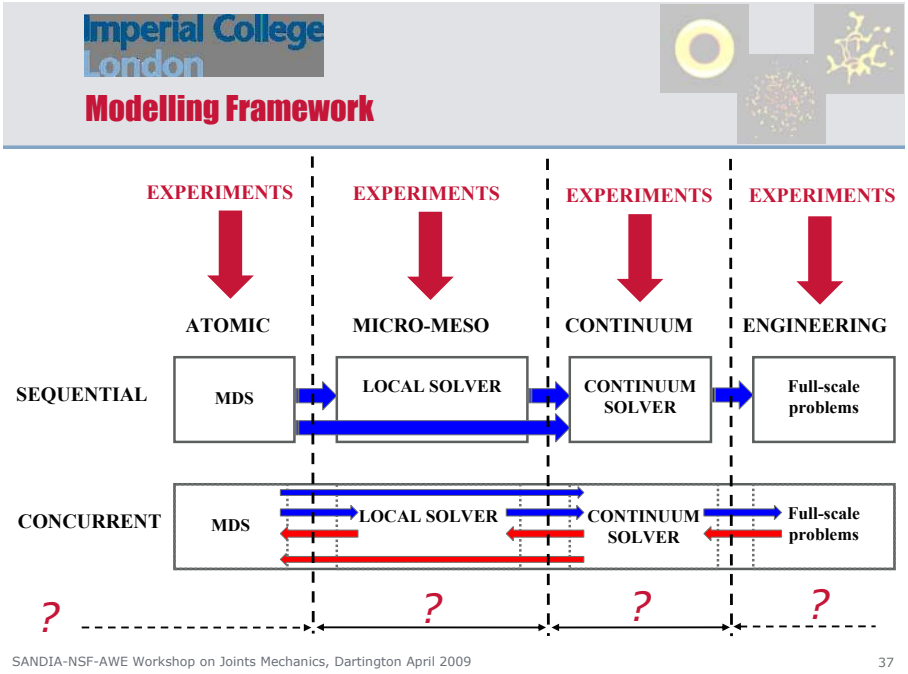
Open questions 2

How do we define the limits of applicability of a model in terms of length- and time-scales

If we could define a modelling framework for the future of joint mechanics modelling, shall we consider the two-way coupling between different scales or shall we just use the information at the lower scales to generate constitutive laws for our continuum descriptions?


Imperial College London
Many-scales interactions: strategy
Time
Years
Minutes
Milliseconds
Microseconds
Picoseconds
Femtoseconds
Coupling Complexity
Need for coupling
Engineering
Continuum
Micro-Meso
Atomistic & Molecular
Quantum Mechanics
Increase in computational and experimental "power"
Increase in understanding
Expertise
Future
10-10
10-9
10-8
10-6
10-3
10-1
1
Length (m)
SANDIA-NSF-AWE Workshop on Joints Mechanics, Dartington April 2009
35


Imperial College London
Many-scales / Many-disciplines
Time
Years
Minutes
Milliseconds
Microseconds
Nanoseconds
Level 5: Engineering
Level 4: Continuum
Level 3: Micro- and Meso- scale
Level 2: Atomic & Molecular
Material Science
Biology and Chemistry
Physics and Computing
10-10
10-9
10-8
10-6
10-3
10-1
1
Lenght (m)
SANDIA-NSF-AWE Workshop on Joints Mechanics, Dartington April 2009
36


Imperial College London
Modelling Framework
EXPERIMENTS
EXPERIMENTS
EXPERIMENTS
EXPERIMENTS
ATOMIC
MICRO-MESO
CONTINUUM
ENGINEERING
SEQUENTIAL
MDS
LOCAL SOLVER
CONTINUUM SOLVER
Full-scale problems
CONCURRENT
MDS
LOCAL SOLVER
CONTINUUM SOLVER
Full-scale problems
?
?
?
?
SANDIA-NSF-AWE Workshop on Joints Mechanics, Dartington April 2009
37

Appendix E

Flip Charts and Summary Slides of Breakout Sessions at the SNL/NSF Joints Workshop

E.1 Summaries and Flip Chart pages of Breakout Sessions of 27 April 2009

What exactly does the engineering community want from better models of joints? What will be the specific benefits (scientific, economic) of achieving a predictive model for joints? Discussions to include:

- What do we need to be able to do by way of 'joint predictions'?
- What levels of accuracy and fidelity are required to achieve a valuable engineering benefit?
- Can we quantify the potential benefits (and thereby make a self-fulfilling business case for the research to follow?)
- How can we include an economic analysis of the benefits of better joint models (and hence better joints)? Weight saving in aerospace? Reduced uncertainty in weapons?
- What advantages will follow from a reliable joint mechanics prediction capability? i.e. not just the ability to predict what will be the characteristics of jointed structures (stresses, wear, vibration levels etc. but the ability to re-design structures so that they have characteristics that we specify.
- How to manage the 'un-repeatability' characteristic that plagues most engineering joints?
- Can/should we include uncertainty and variability (or robustness) considerations into joint design?

E.1.1 Breakout Session A1

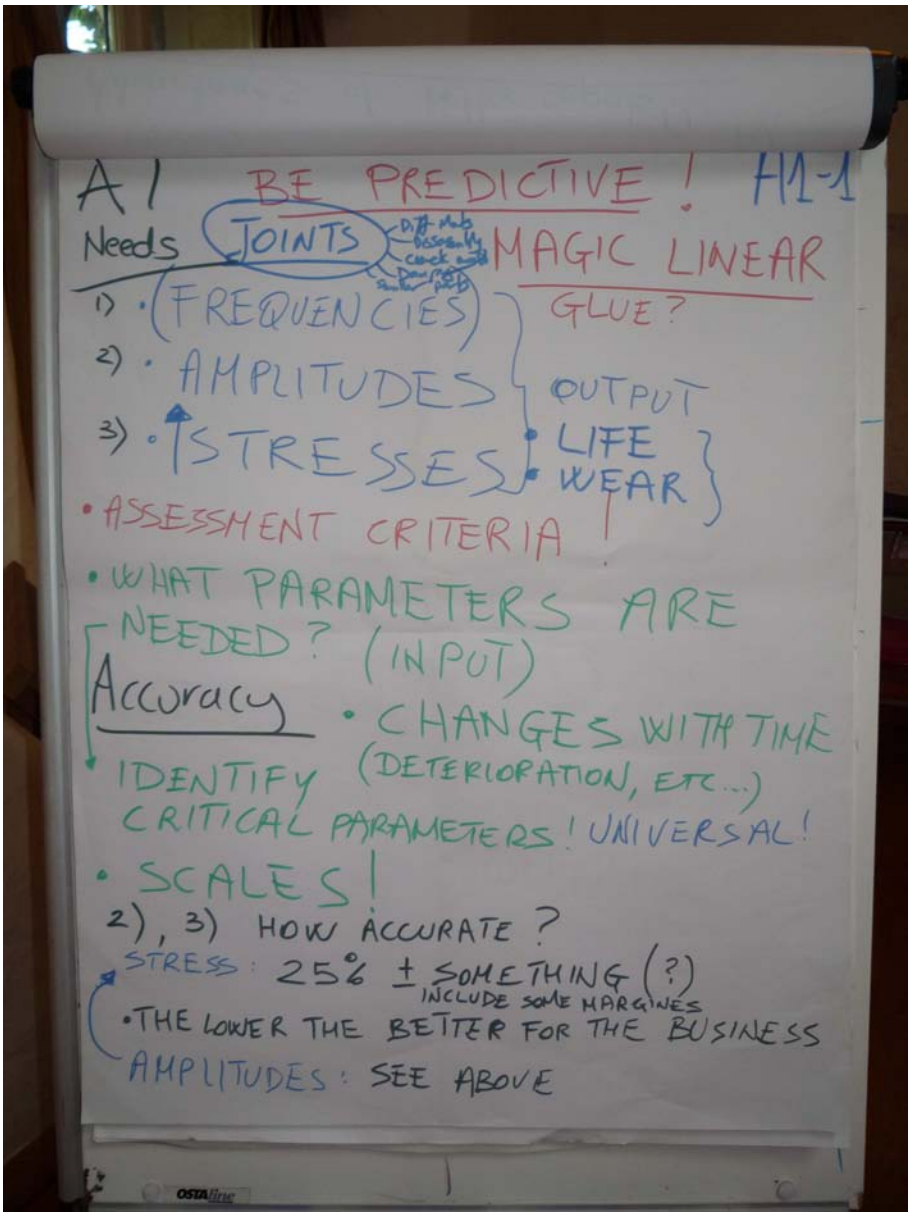


Figure E.1. Flipchart 1 of break outsession A1.

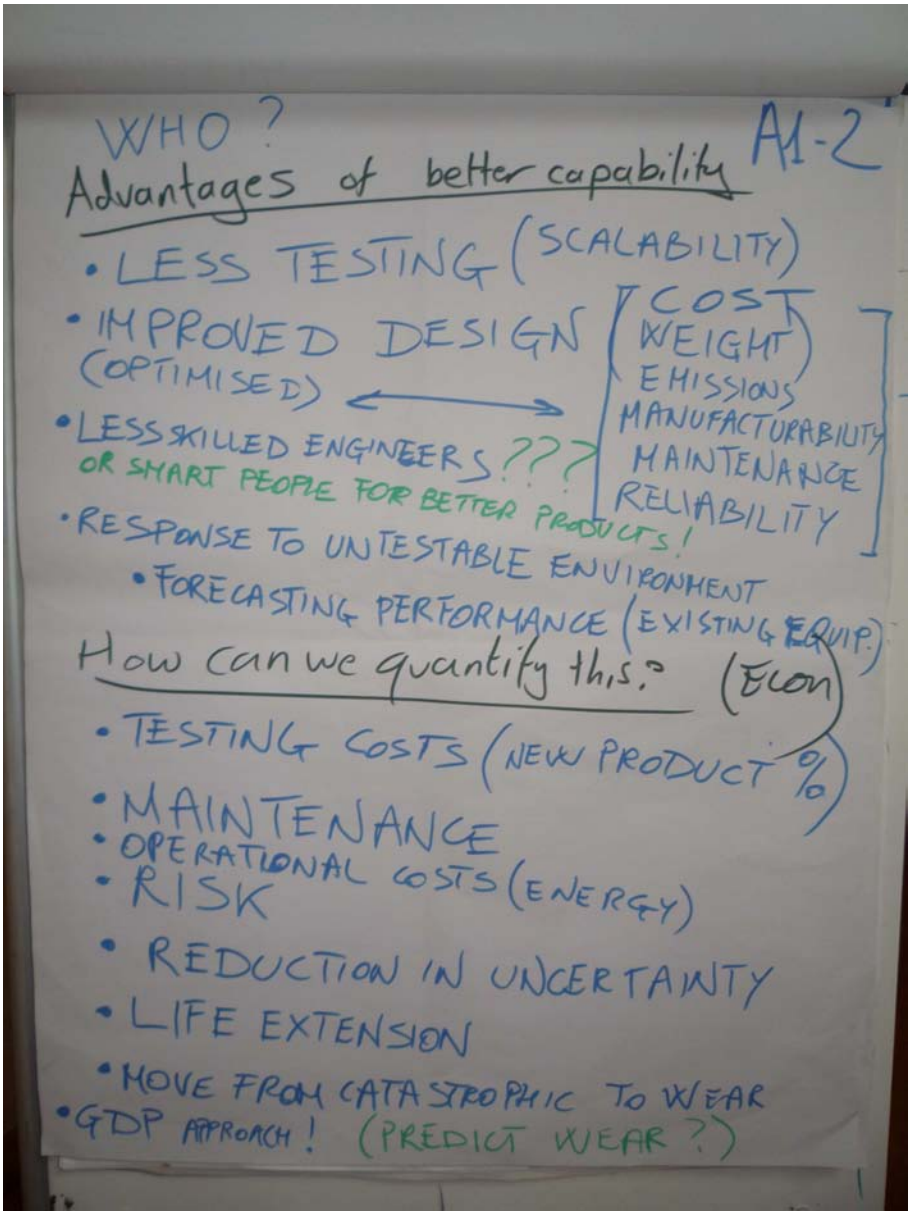


Figure E.2. Flipchart 2 of break outsession A1.

E.1.2 Breakout Session A2

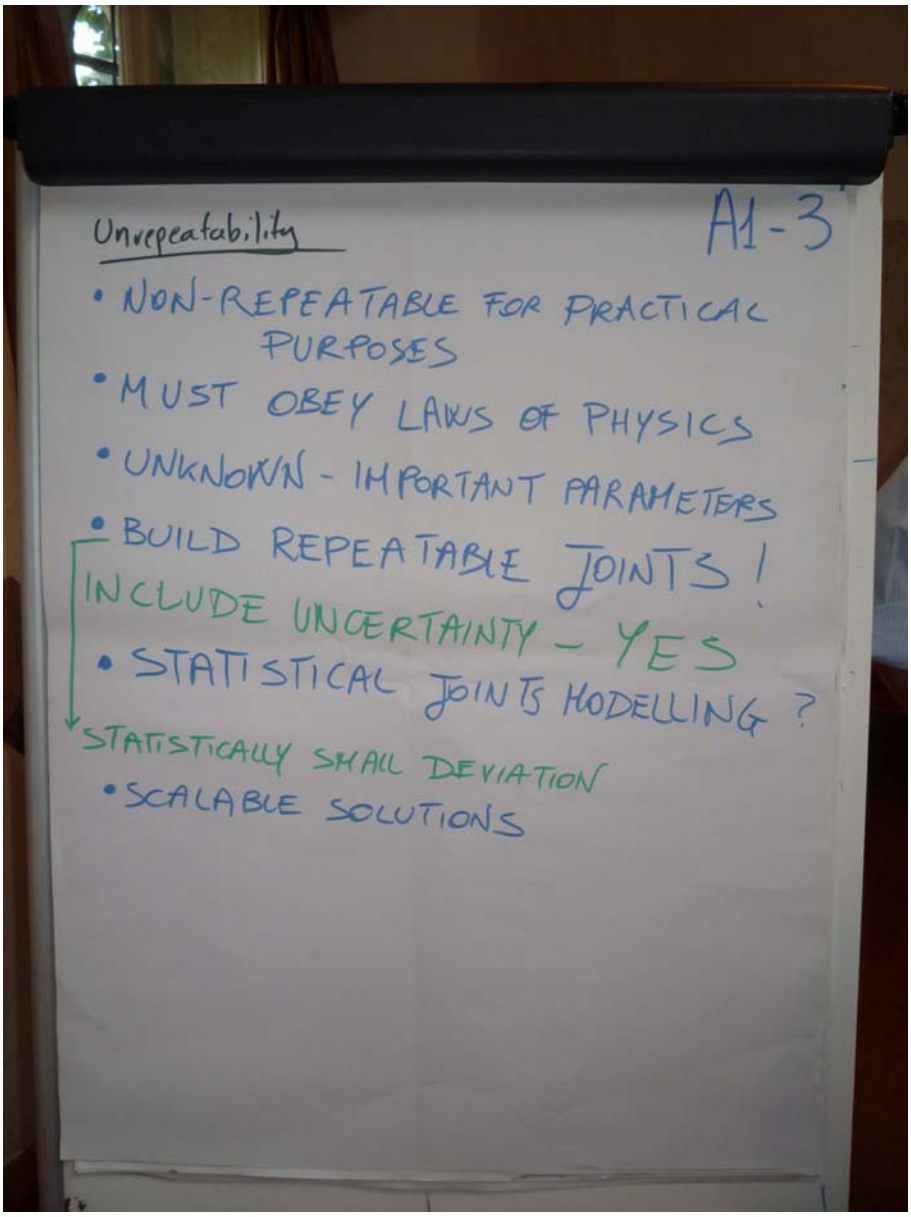


Figure E.3. Flipchart 3 of break outsession A1.

Before addressing end user needs and requirements we must answer some fundamental questions:

- How much knowledge do we have?

There is a need to advance our understanding of interface physics by developing new experimental and measurement techniques to directly interrogate contact interfaces in situ

- What physical parameters are important for the characterization of our models?

We need to develop phenomenological linkages to advance our understanding of the important physics. This acts in tandem with robust validation of our calculations.

Figure E.4. Summary slide 1 of breakout session A2.

It is not realistic and perhaps not important to seek a comprehensive catalog of phenomena at all length scales

There is value in establishing a "hierarchy of analysis" to capture different users requirements for solution fidelity and accuracy at a given length scale of interest.

Figure E.5. Summary slide 2 of breakout session A2.

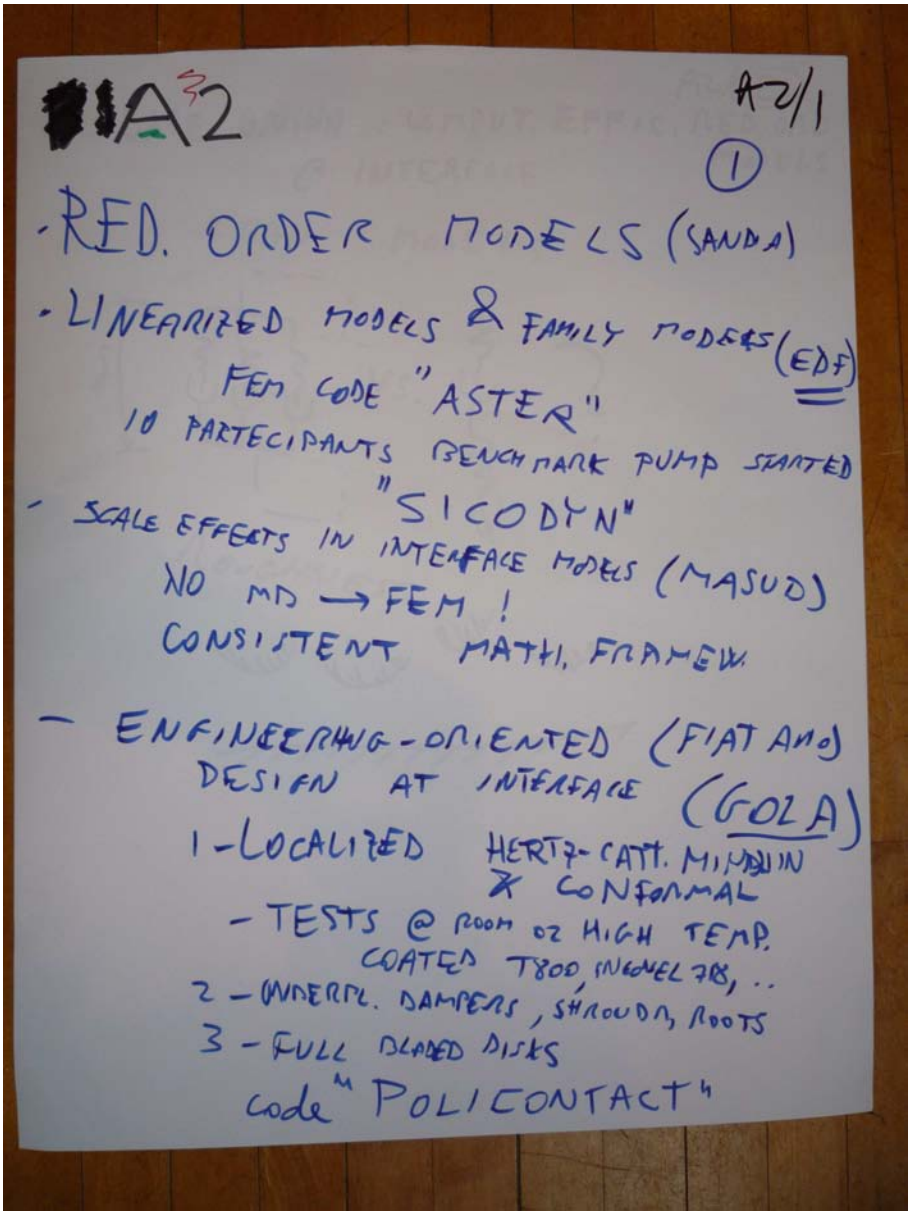


Figure E.6. Flipchart 1 of break outsession A2.

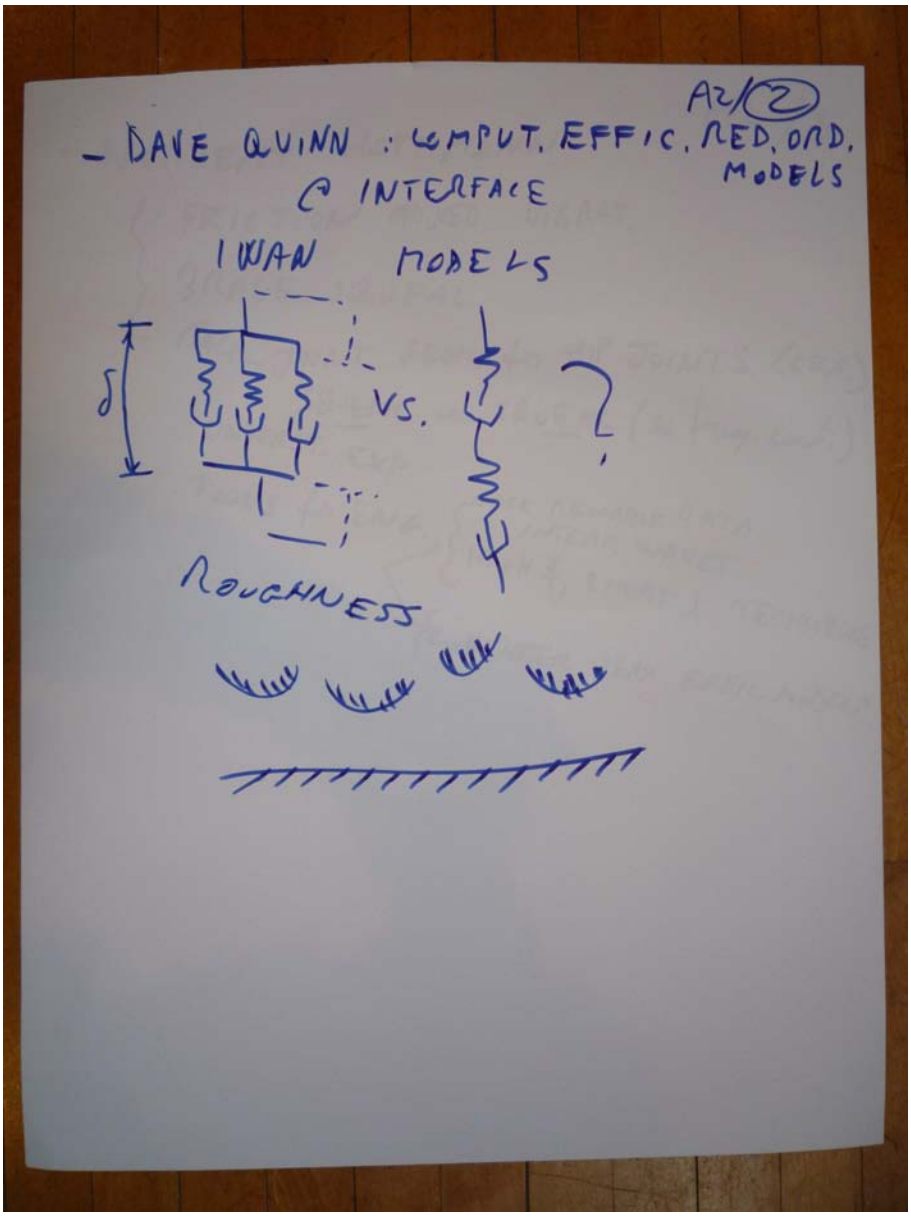


Figure E.7. Flipchart 2 of break outsession A2.

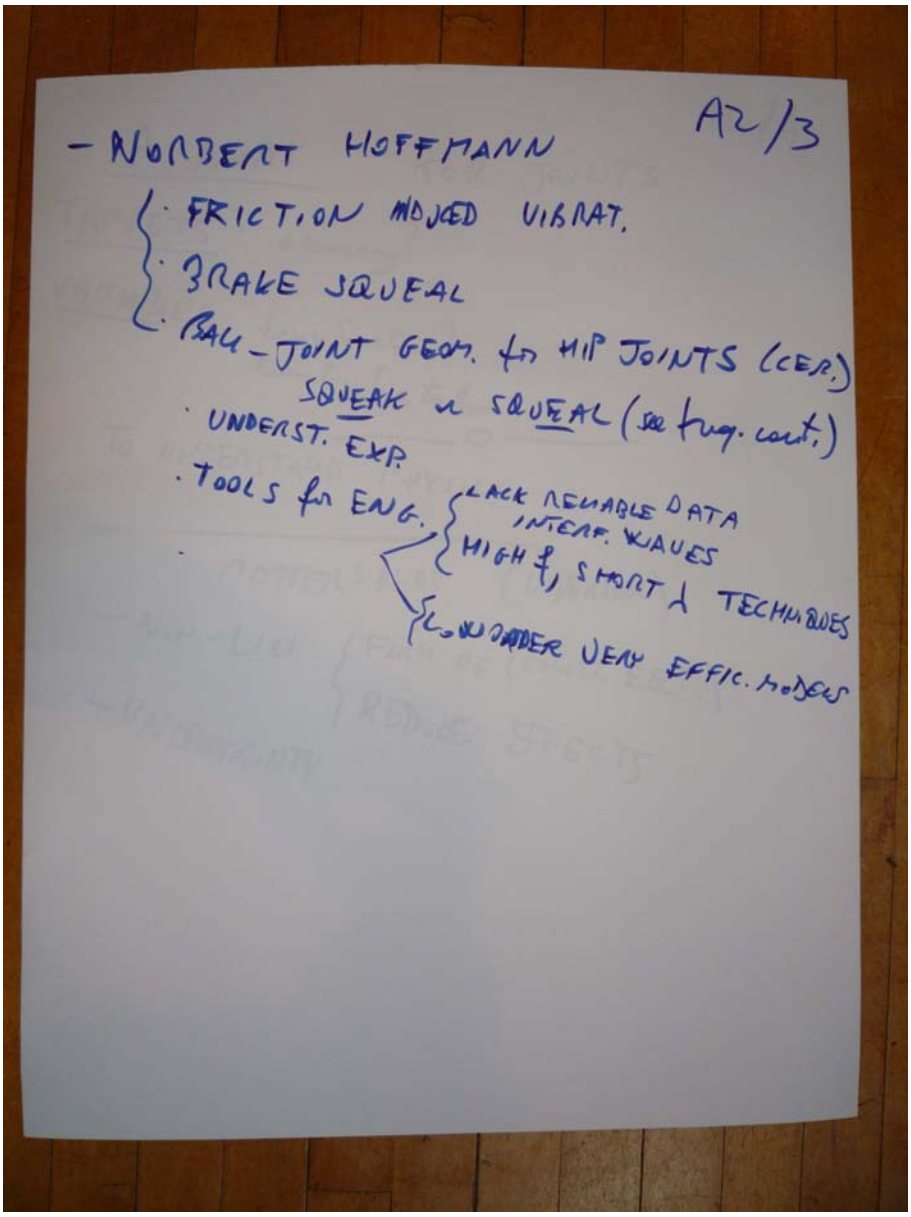


Figure E.8. Flipchart 3 of break outsession A2.

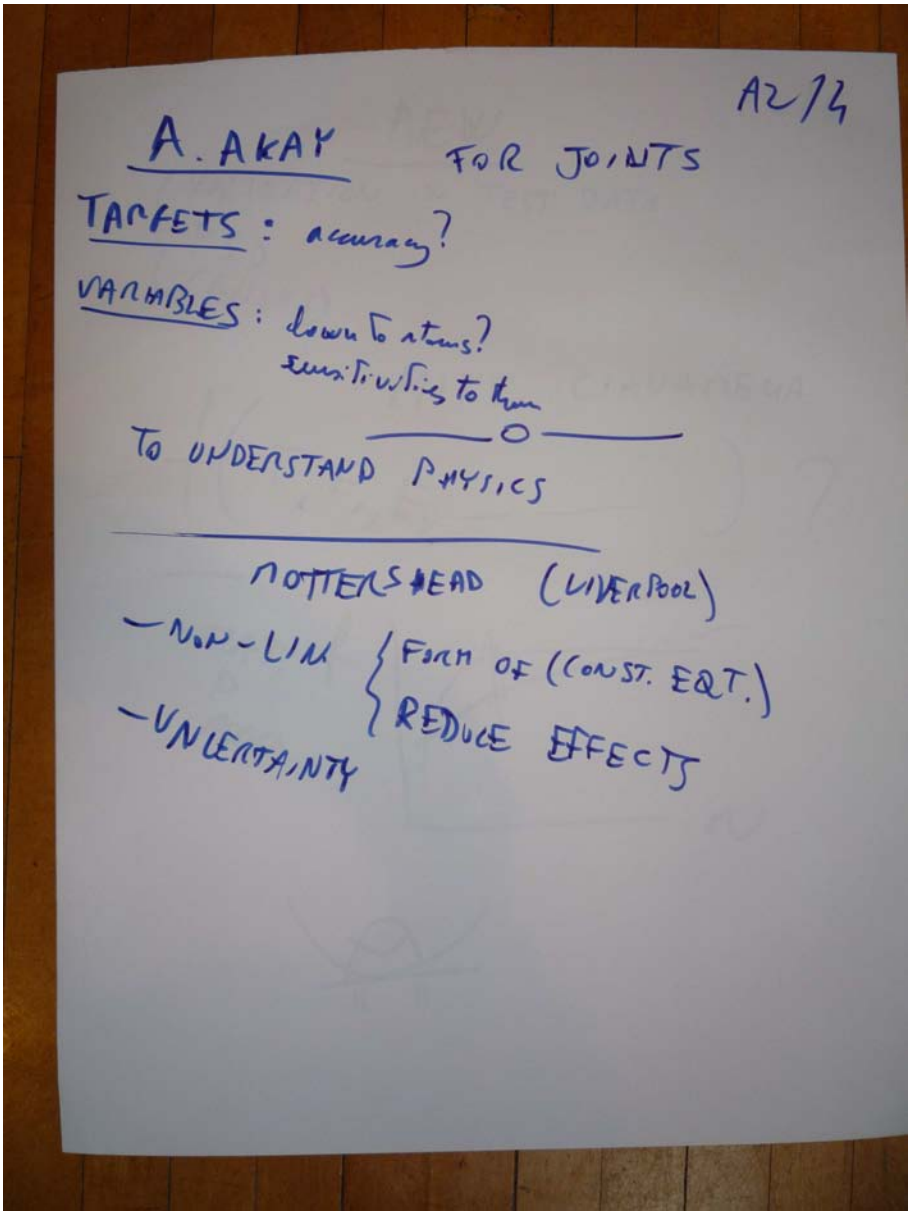


Figure E.9. Flipchart 4 of break outsession A2.

E.1.3 Breakout Session A3

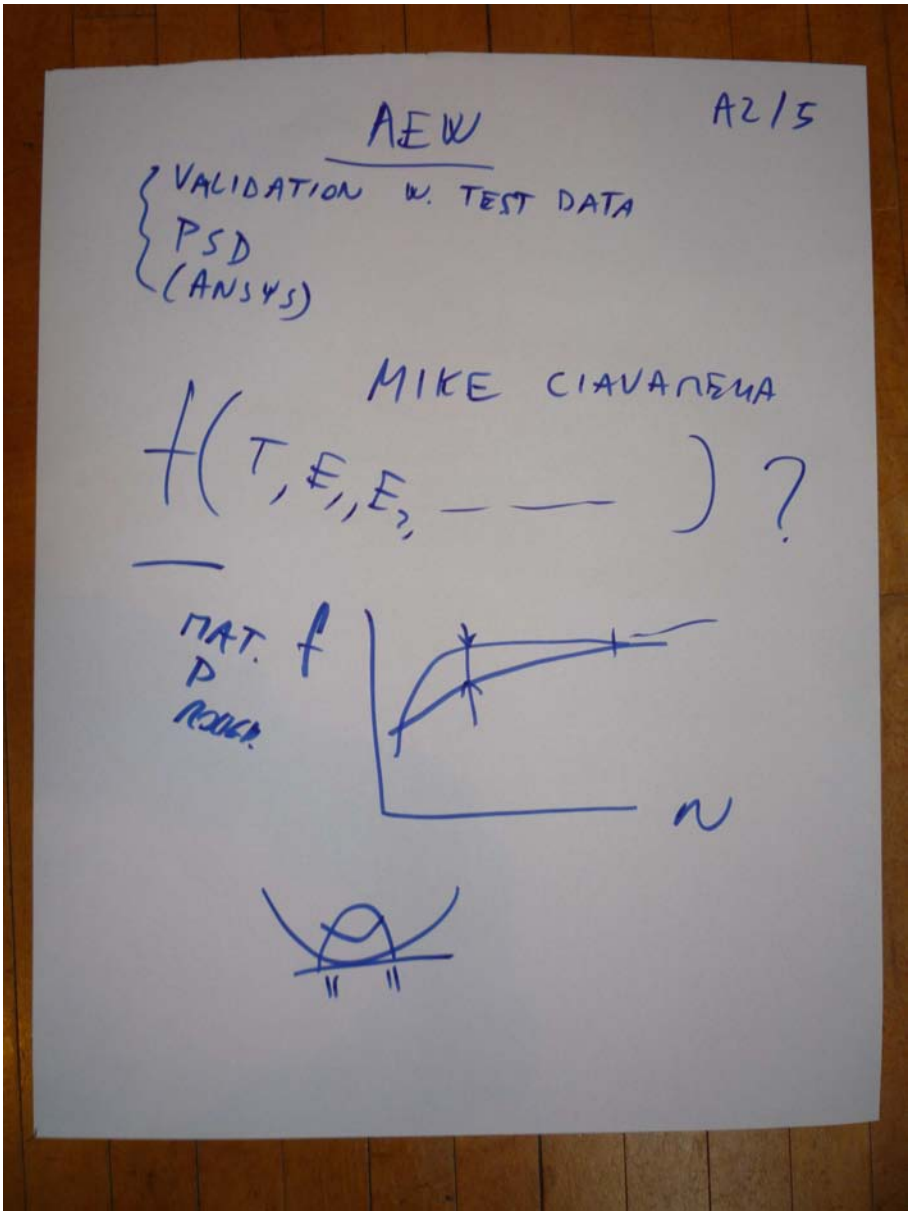


Figure E.10. Flipchart 5 of break outsession A2.

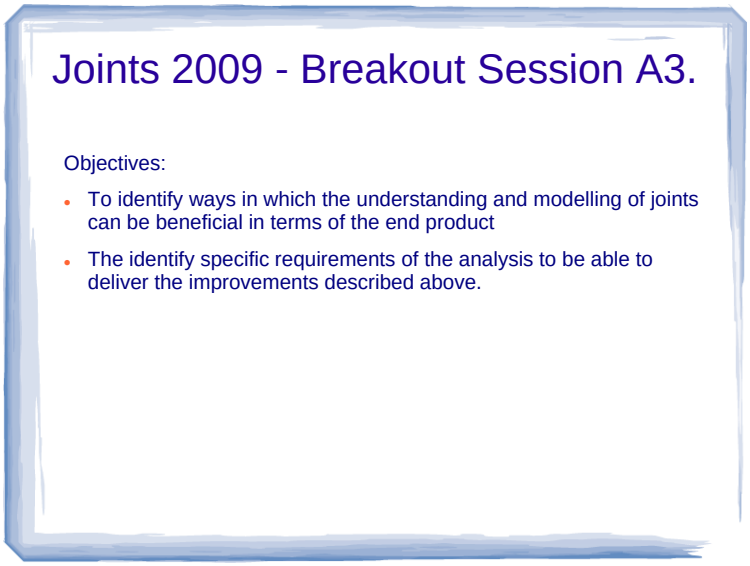


Figure E.11. Summary slide 1 of breakout session A3.

Figure E.12. Summary slide 2 of breakout session A3.

Joints 2009 - Breakout Session A3.

Potential Routes to Financial Benefit

- Reduction of number of tests. Fewer, more targeted tests.
- Extending life of existing structures
  - Extending life of nuclear power station through improved analysis only. E.g. extending life from 50 years to 60 years through improved analysis alone.
  - Reduced down time.(which can have financial penalties)
  - Use of prognostics to monitor condition of joints
- Reduced design time.
- Enables robust design => reduced component rejection & rebuild
- More optimum design (weight, unit cost, performance) which is currently compromised by fear of uncertain joint behaviour
- Long time horizon is important. Test-fix-test-fix cycle may be cheaper and quicker in isolation but provides no long term benefit.

Figure E.13. Summary slide 3 of breakout session A3.

Joints 2009 - Breakout Session A3.

Specification Must Be Met

- Requirement for design of systems which cannot be tested (eg nuclear weapons).

Prediction Capability Required

- Damping accuracy: 20% to factor of 2
  - Case dependent / amplitude dependent.
  - USAF target was 20% on prediction of forced response (still seems reasonable target).
- Stiffness accuracy: 20% to a factor of 2.
  - Maybe target should be based in frequency.
- Also need uncertainty on each parameter.

Figure E.14. Summary slide 4 of breakout session A3.

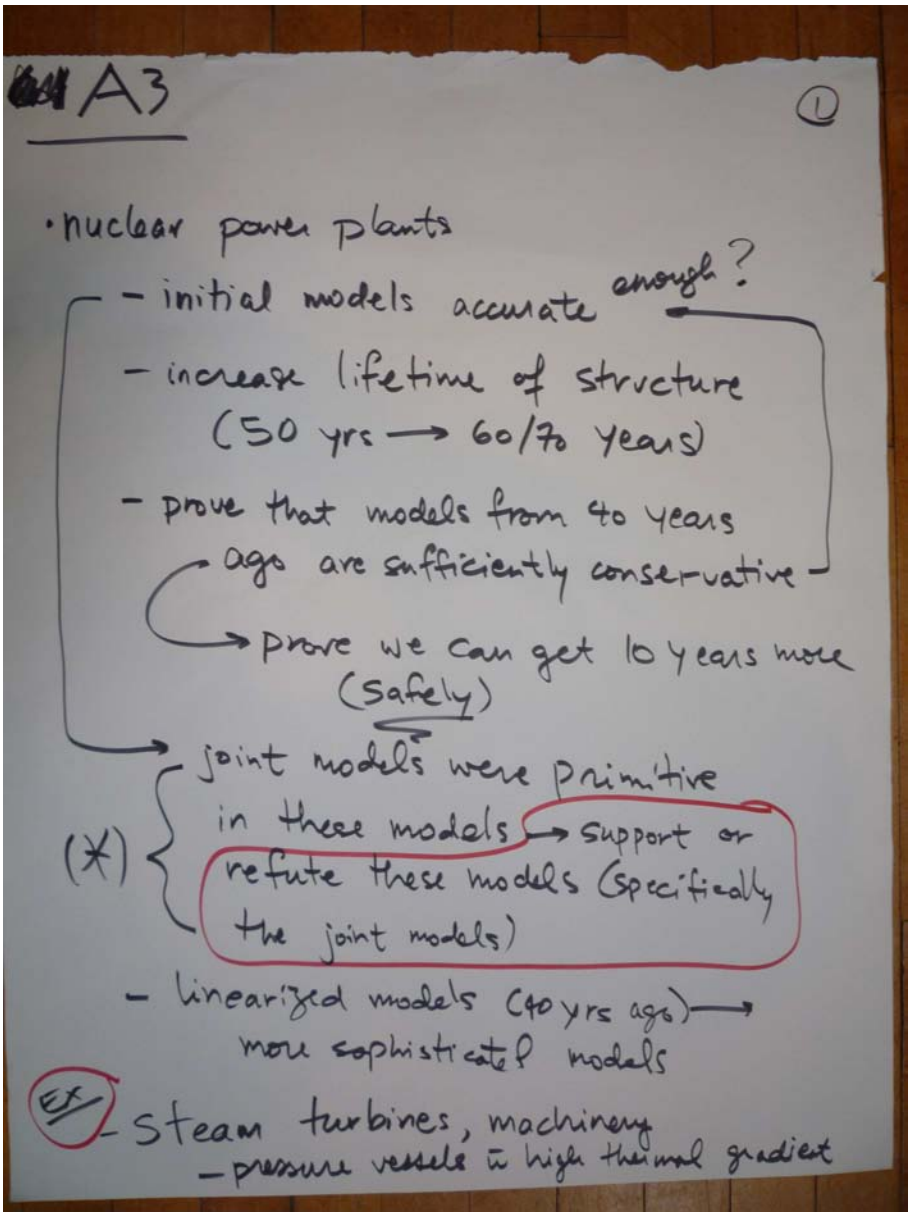


Figure E.15. Flipchart 1 of break outsession A3.

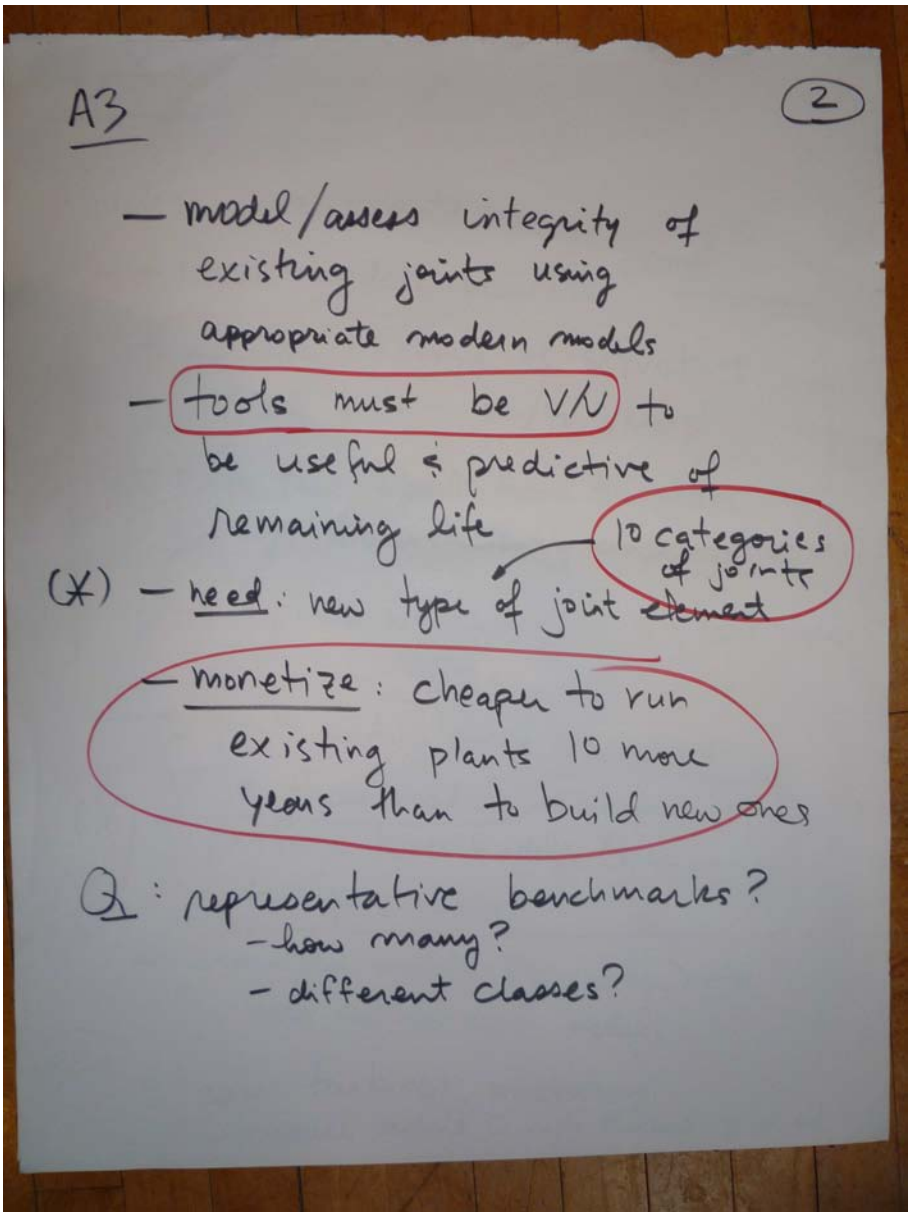


Figure E.16. Flipchart 2 of break outsession A3.

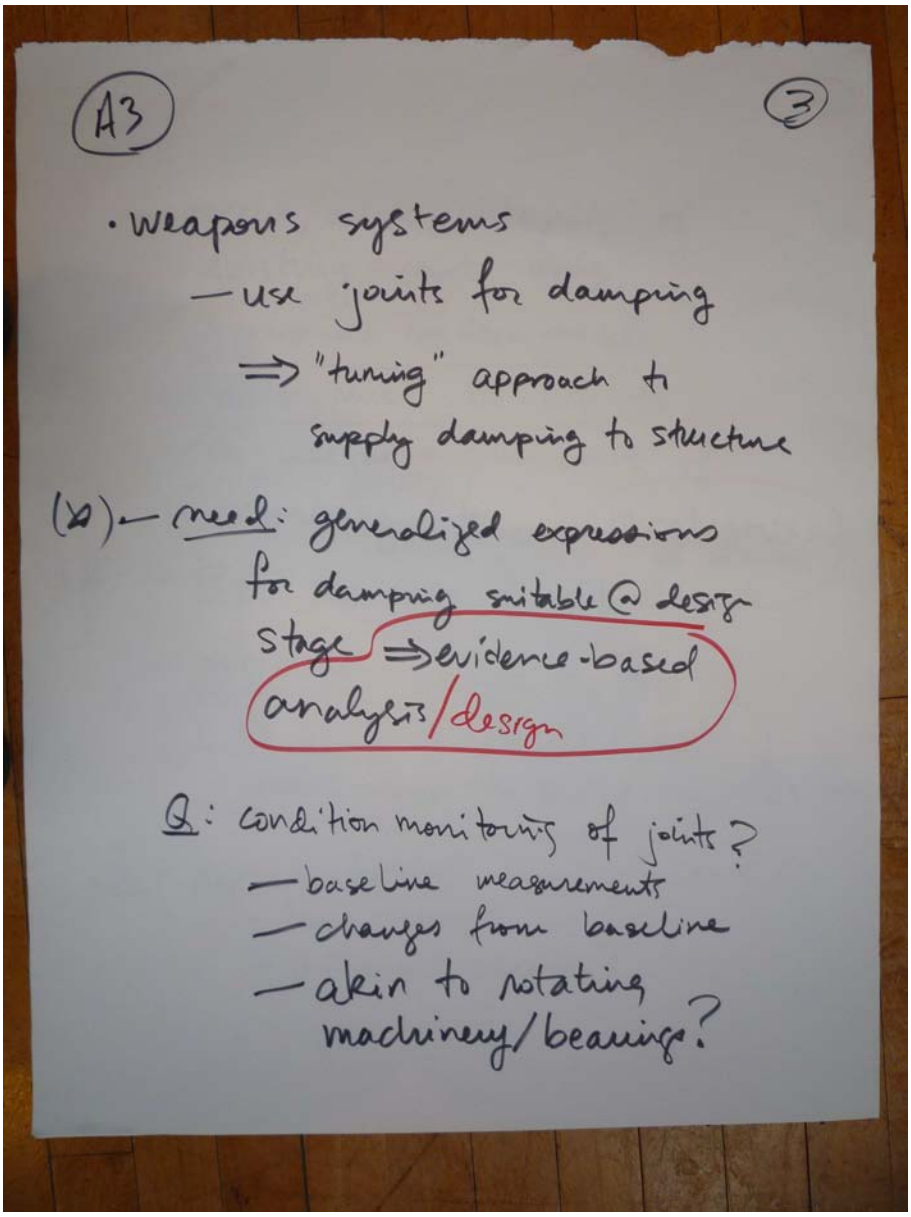


Figure E.17. Flipchart 3 of break outsession A3.

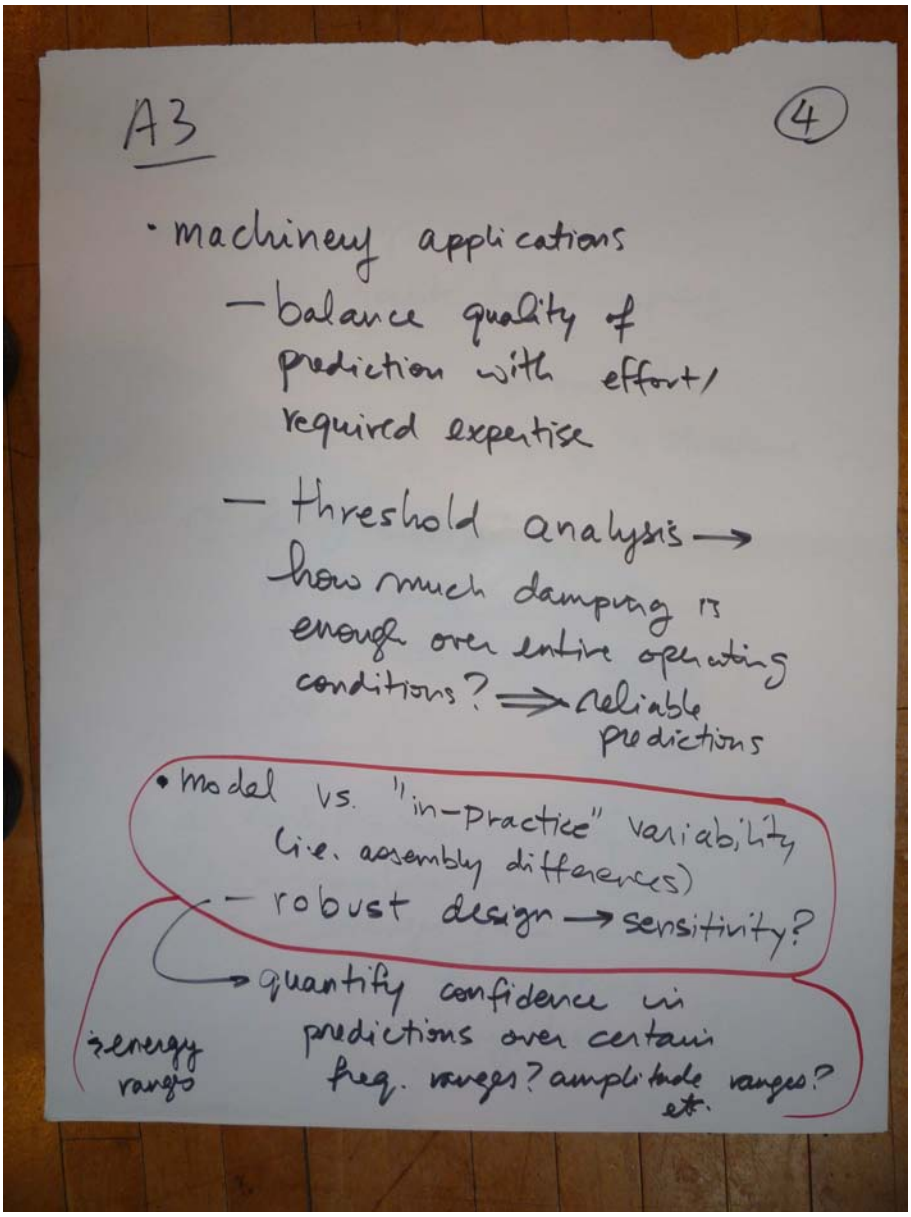


Figure E.18. Flipchart 4 of break outsession A3.

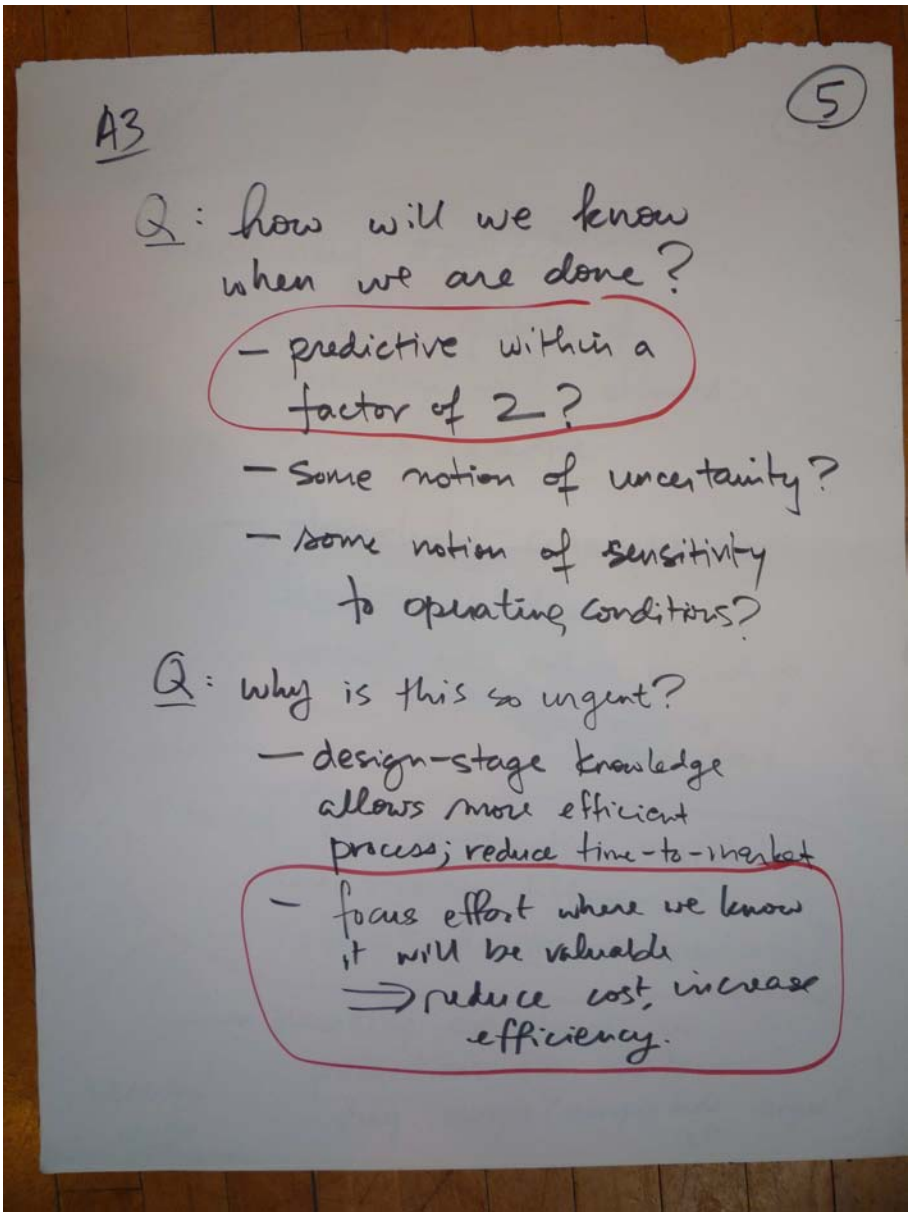


Figure E.19. Flipchart 5 of break outsession A3.

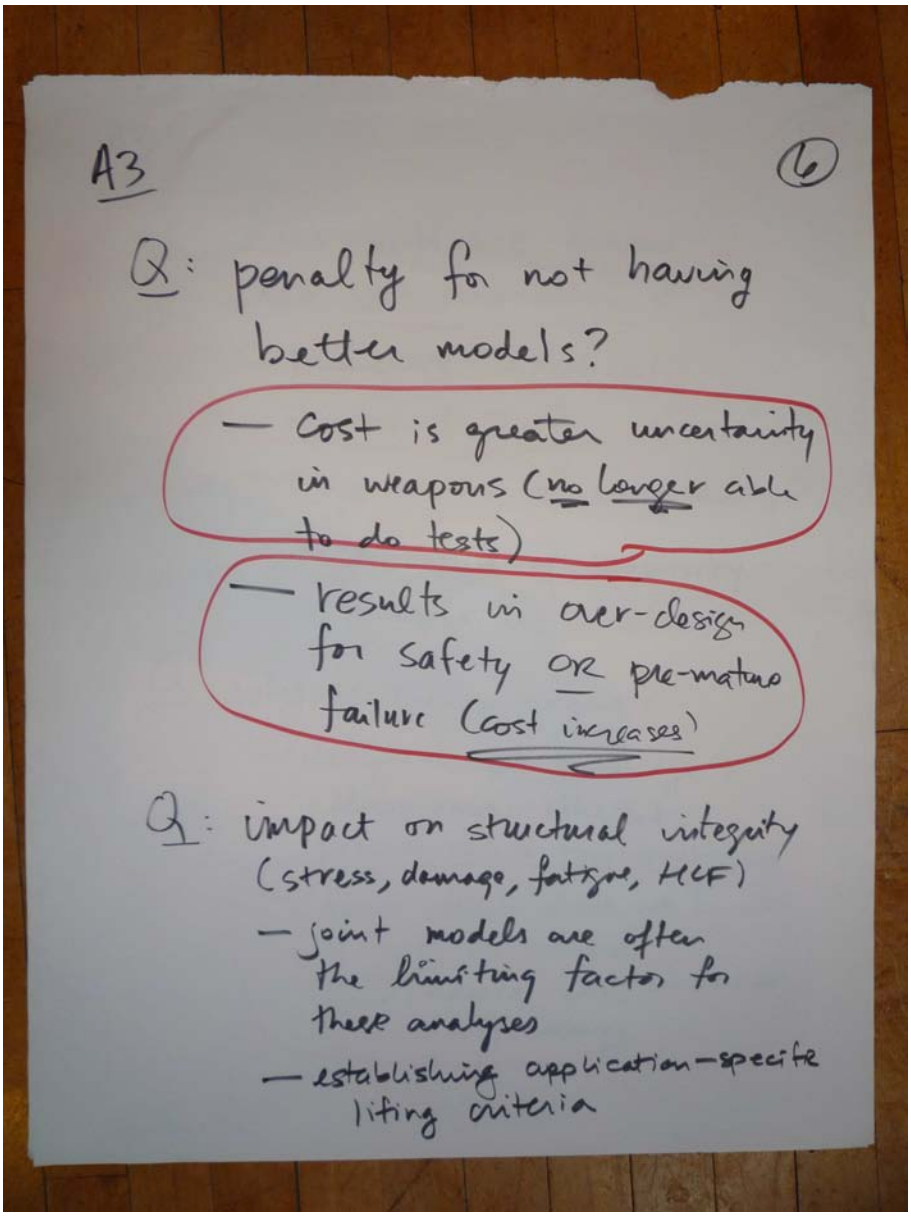


Figure E.20. Flipchart 6 of break outsession A3.

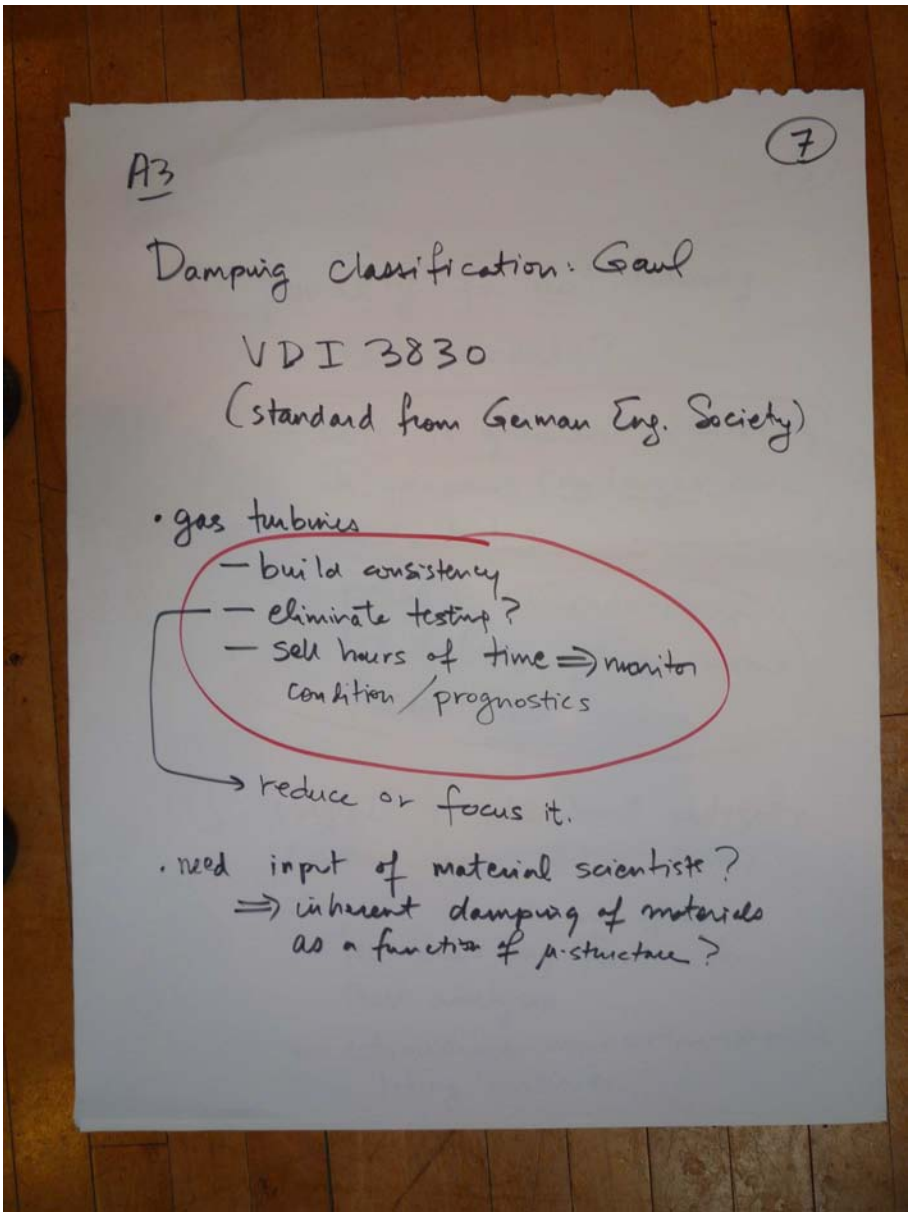


Figure E.21. Flipchart 7 of break outsession A3.

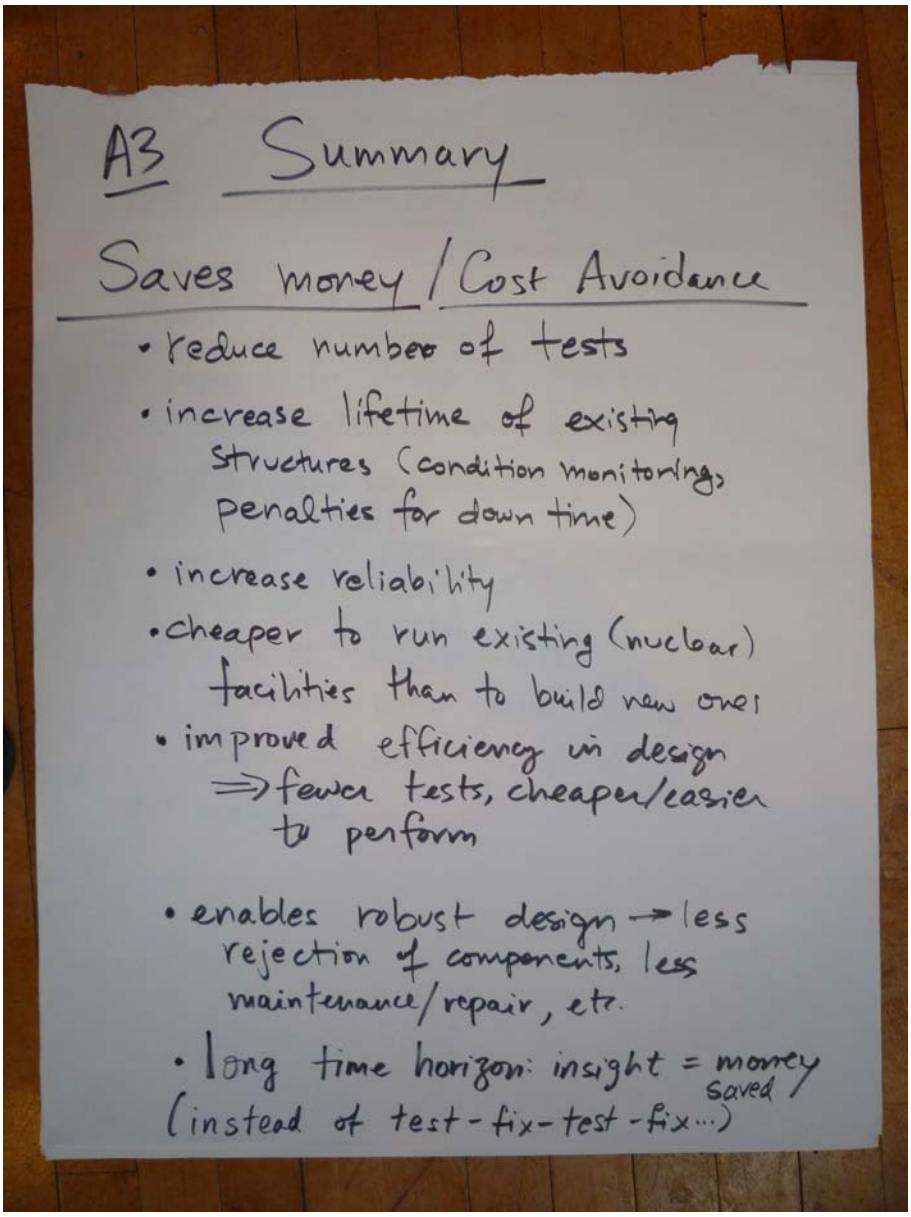


Figure E.22. Flipchart 8 of break outsession A3.

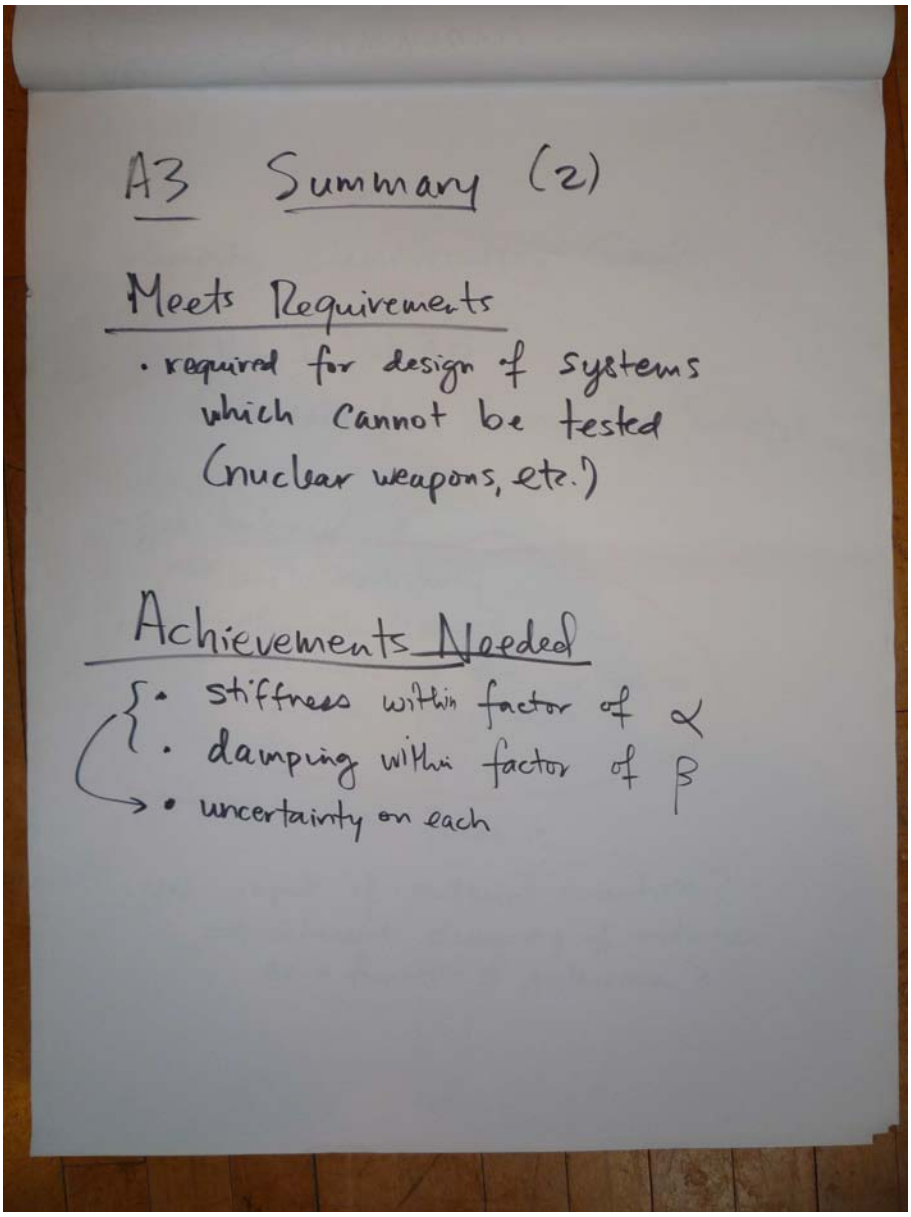


Figure E.23. Flipchart 9 of break outsession A3.

E.2 Summaries and Flip Chart pages of Breakout Sessions of 28 April 2009

A critical assessment of current capabilities (and known limitations) in respect of our ability to model, and predict the dynamic behavior of structures with joints. Topics to include:

- Which physical phenomena do we understand well enough to model?
- Which do we not understand adequately?
- What phenomena do we suspect are missing from these lists and how to we identify these and get them "on the map"?

This session to include Modelling Capabilities; (Numerical) Analysis Capabilities and Experimental Capabilities

What can we say about our capabilities in modelling/computation/measurements in respect of:

Phenomena (i.e. observing, understanding, modelling, predicting, validating etc.

And

Applications to engineering designs, optimization, problem resolution etc.

Individual joints; dynamics of jointed structures; dynamics of structures with many interfaces;...

E.2.1 Breakout Session B1

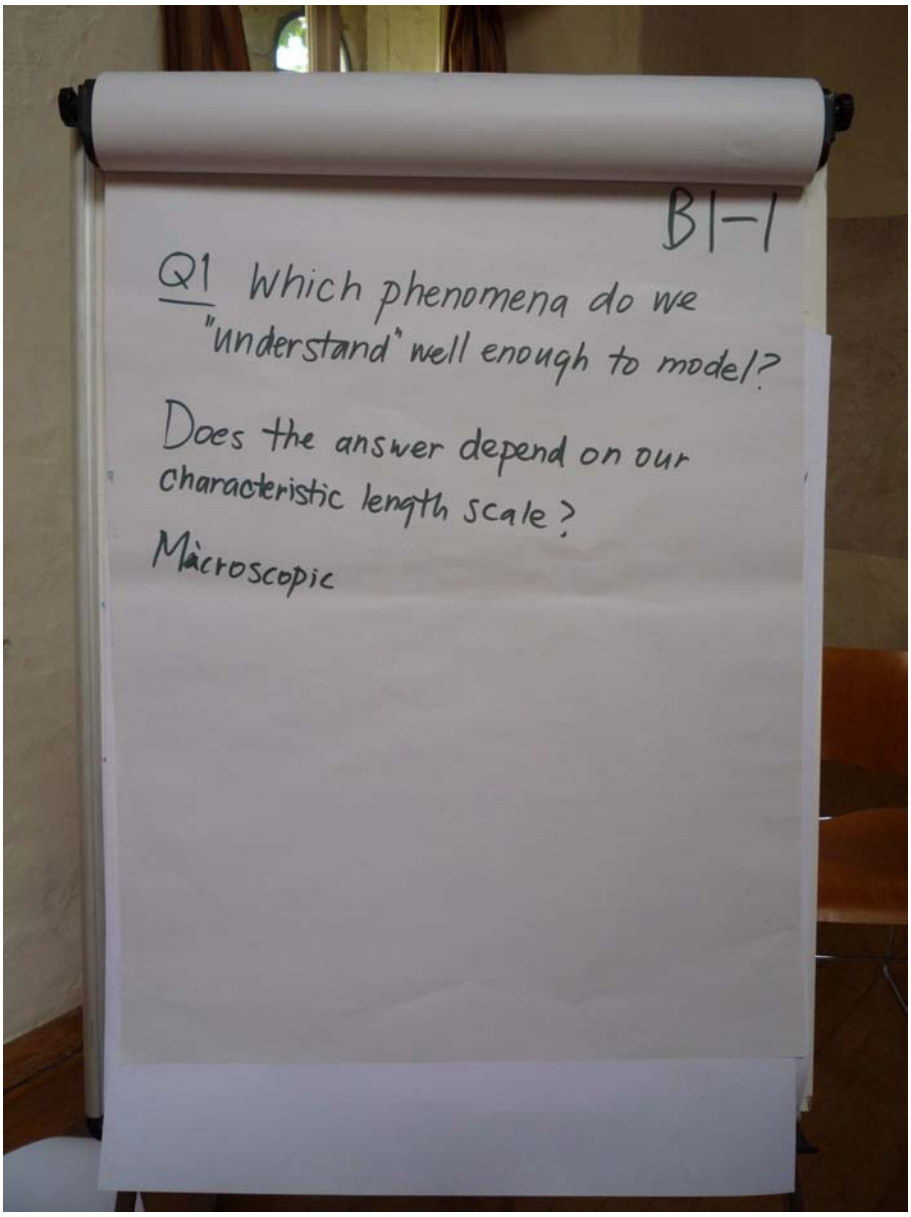


Figure E.24. Flipchart 1 of break outsession B1.

Figure E.25. Flipchart 2 of break outsession B1.

E.2.2 Breakout Session B2

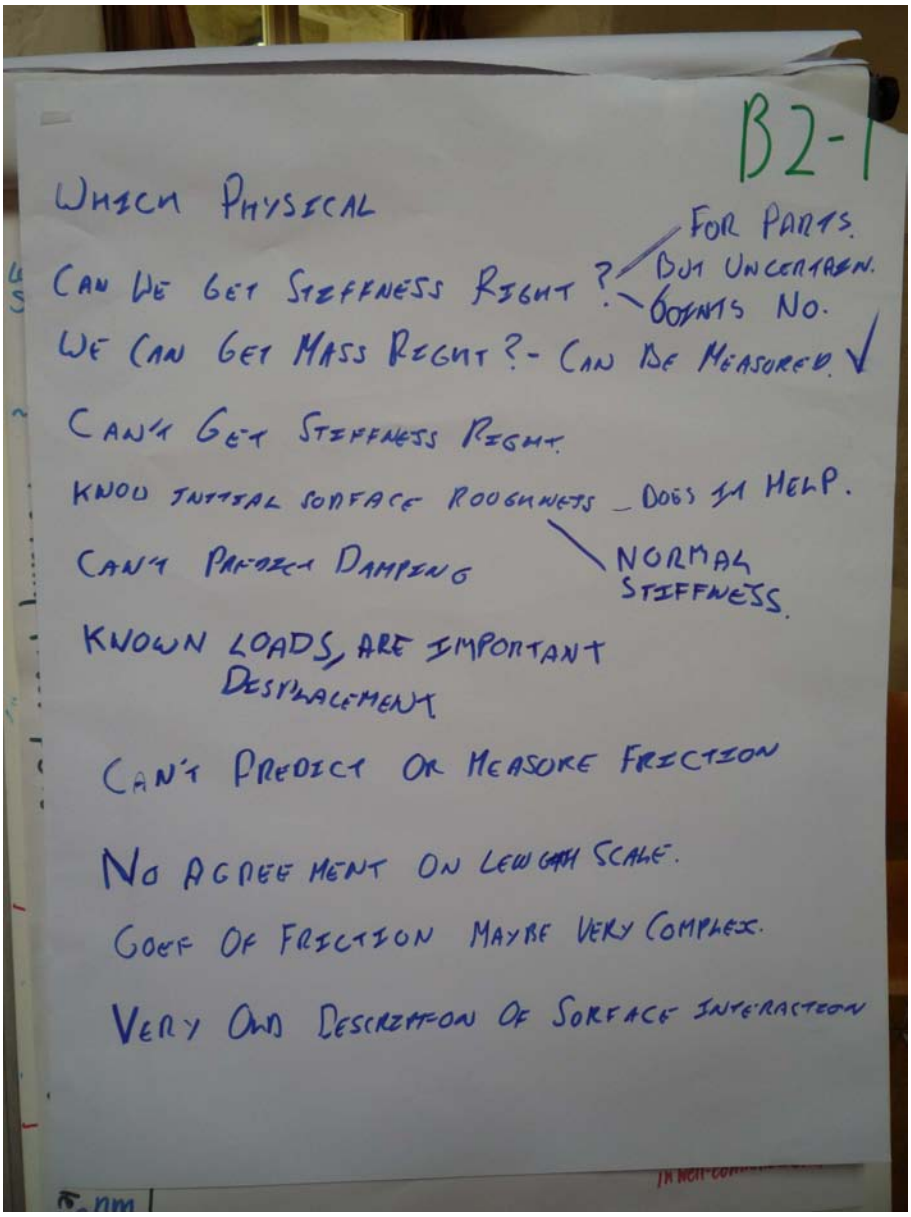


Figure E.26. Flipchart 1 of break outsession B2.

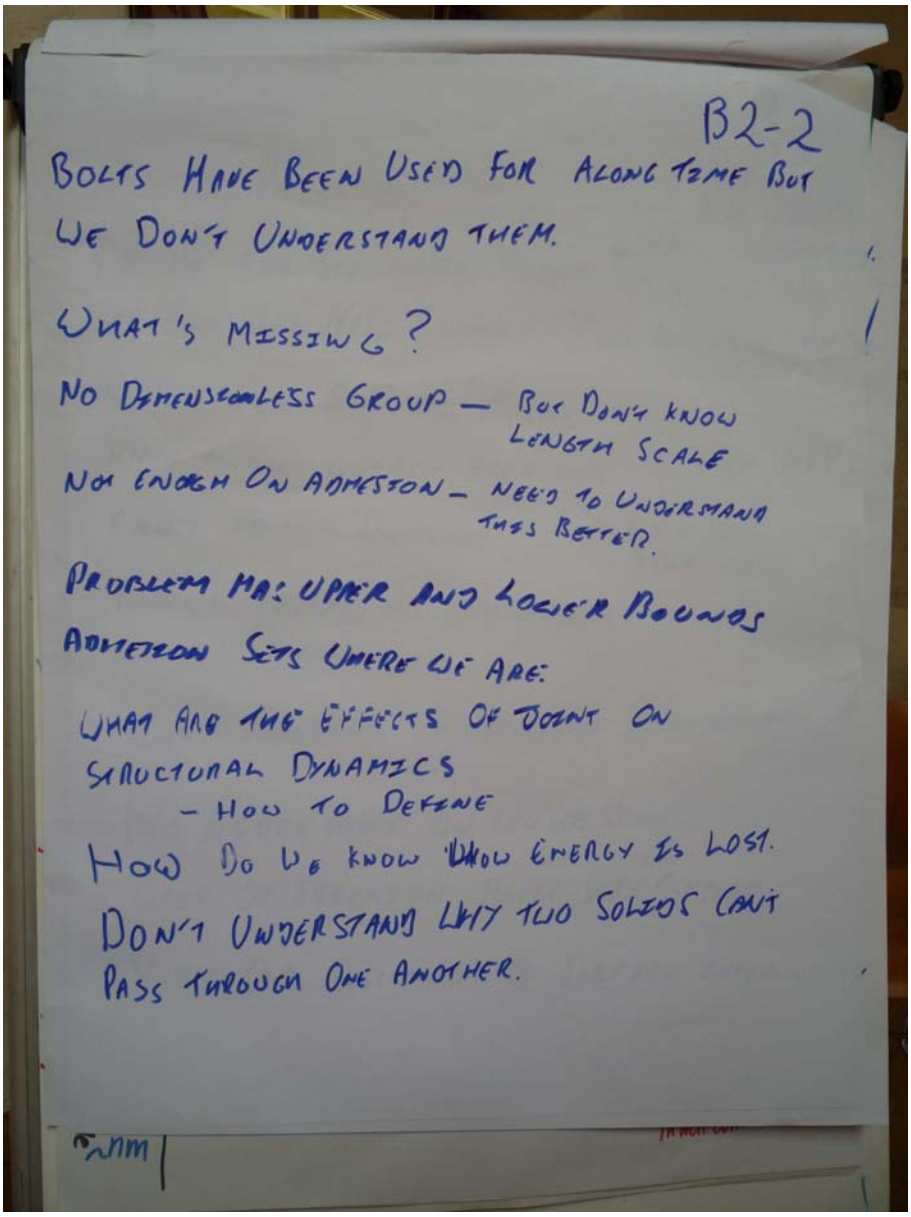


Figure E.27. Flipchart 2 of break outsession B2.

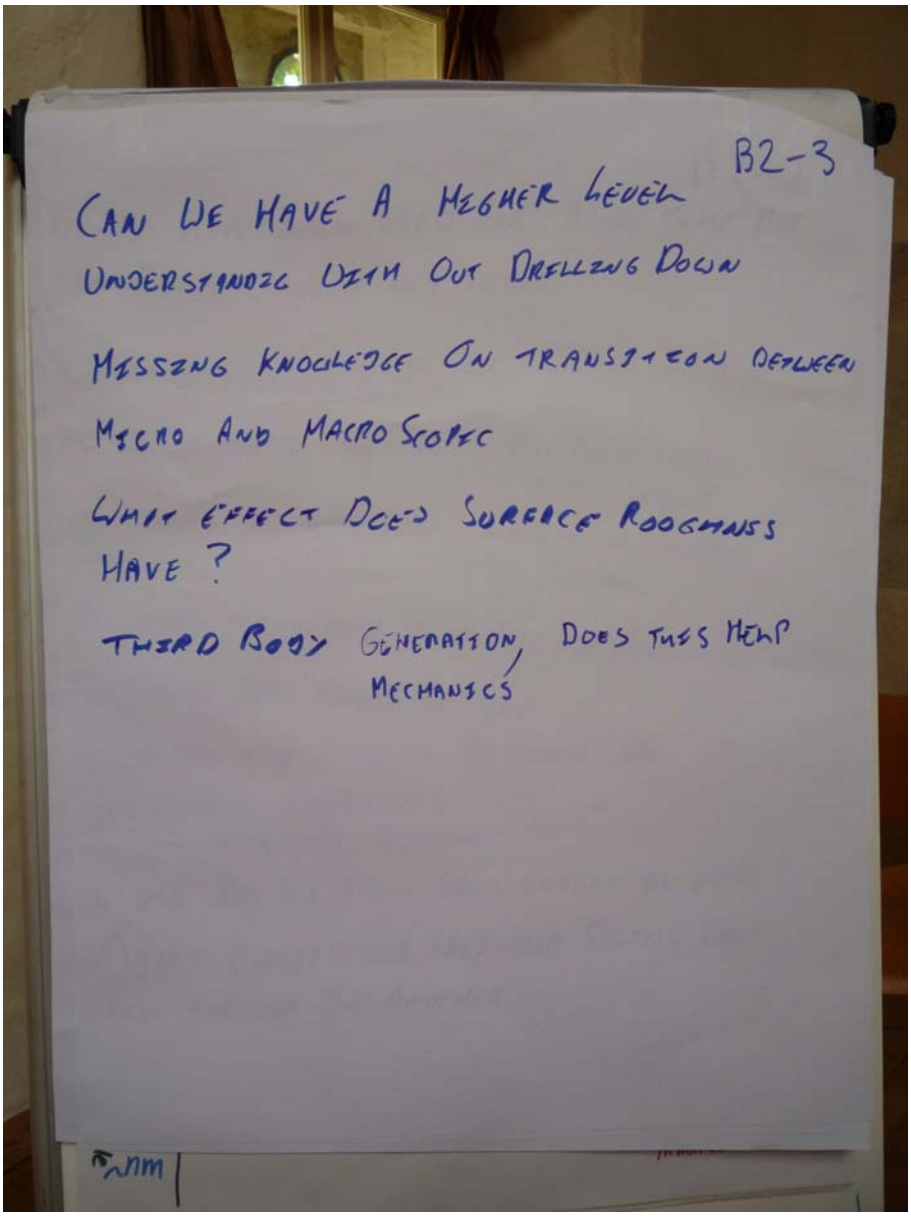


Figure E.28. Flipchart 3 of break outsession B2.

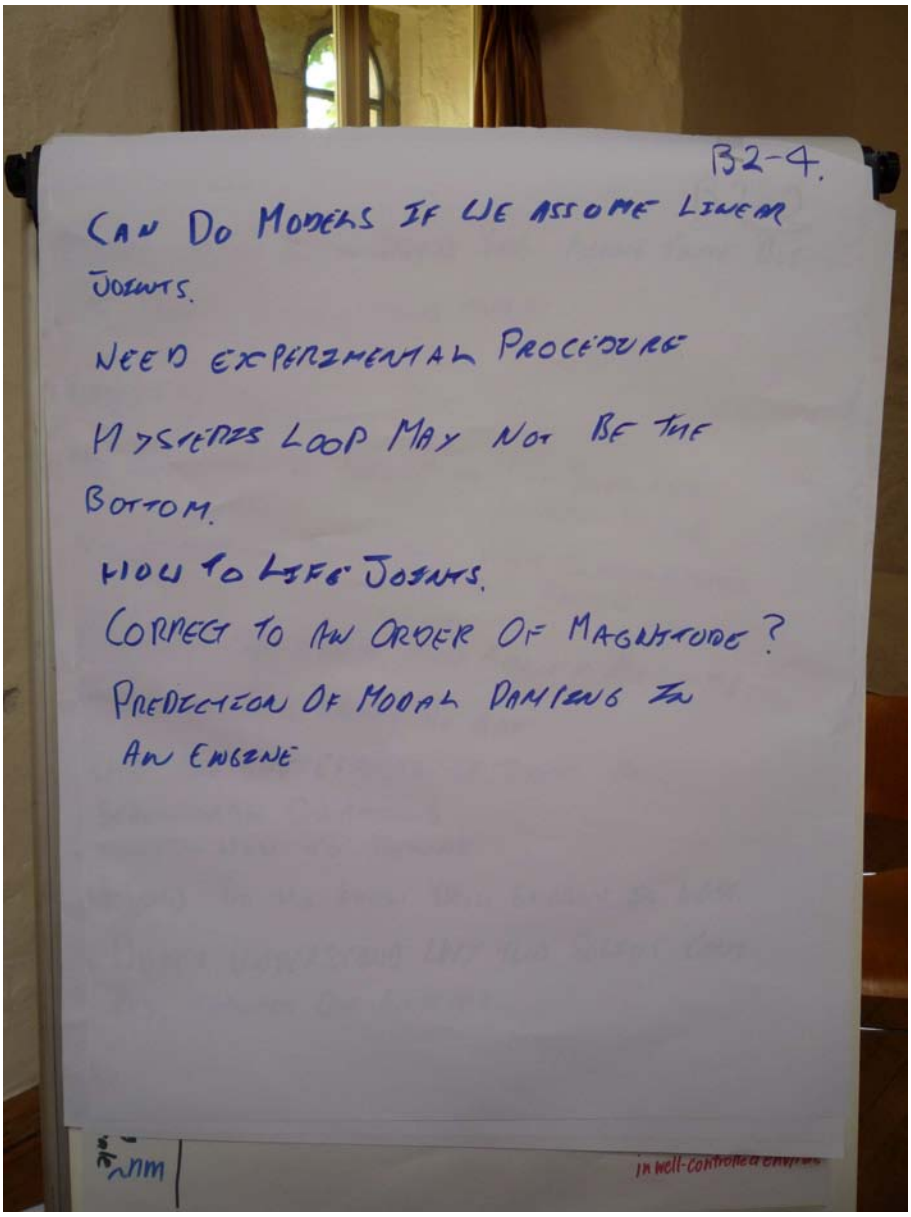


Figure E.29. Flipchart 4 of break outsession B2.

E.2.3 Breakout Session B3

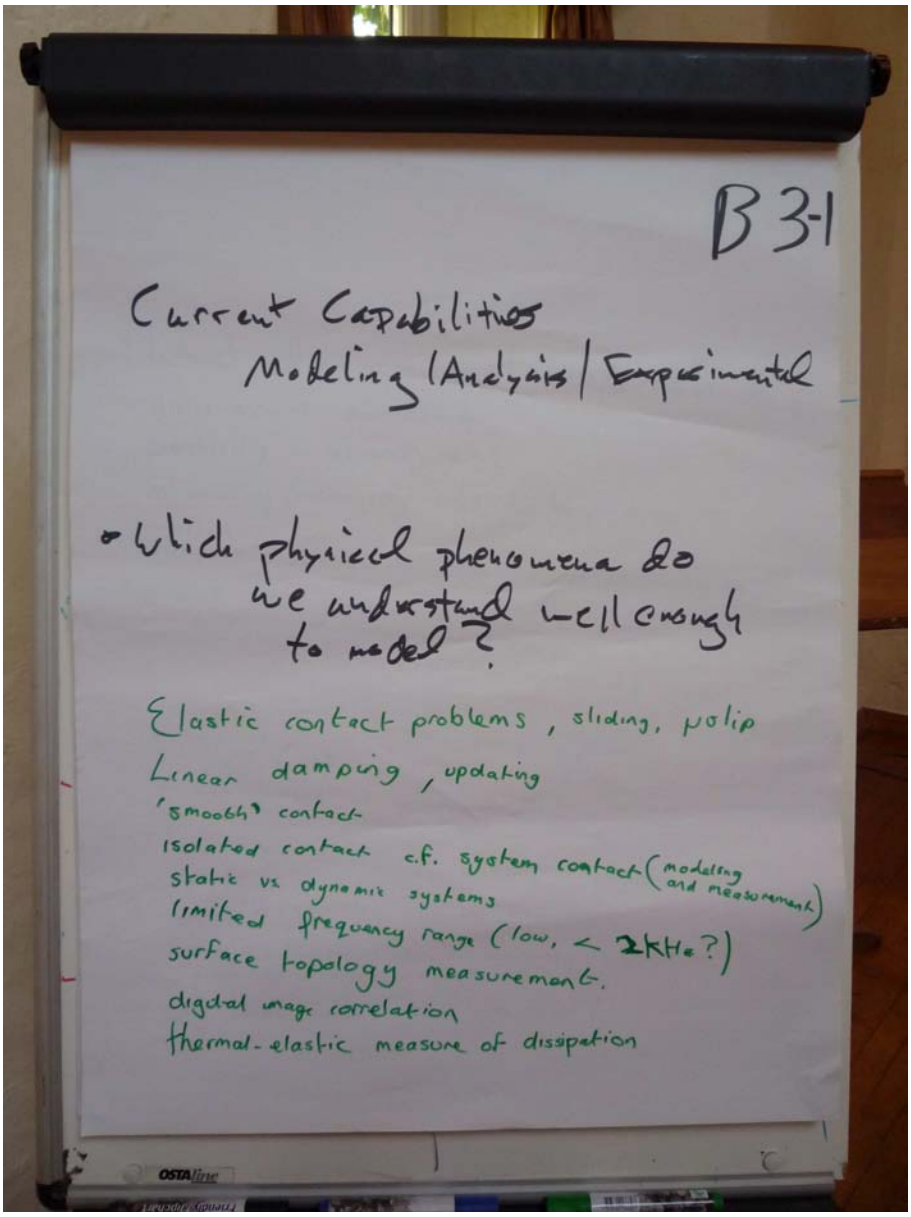


Figure E.30. Flipchart 1 of break outsession B3.

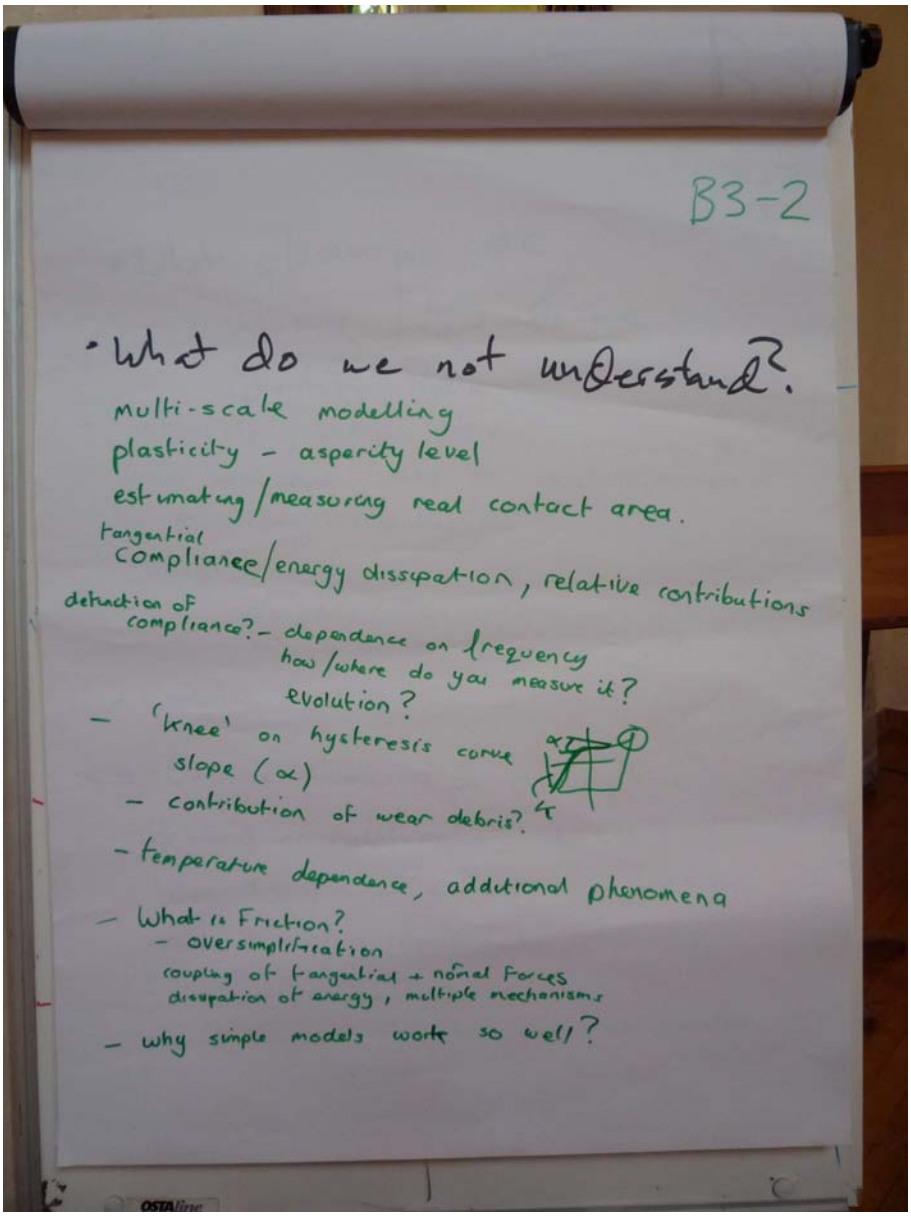


Figure E.31. Flipchart 2 of break outsession B3.

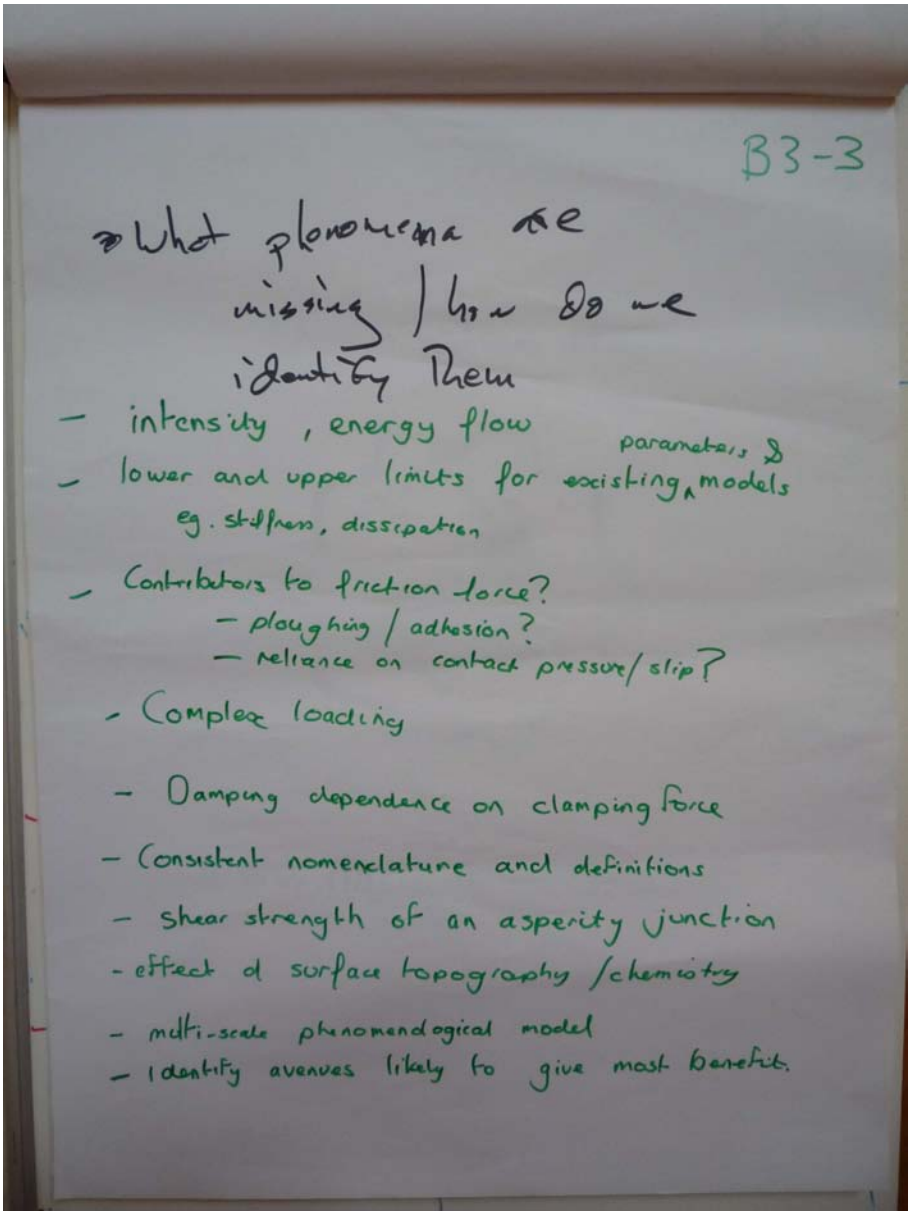


Figure E.32. Flipchart 3 of break outsession B3.

2009 Joints Workshop
C1 Tomorrow's Capabilities

Coordinated by Alex Vakakis and Tom Farris

Figure E.33. Summary slide 1 of breakout session C1.

E.3 Summaries and Flip Chart pages of Breakout Sessions of 29 April 2009

By reviewing current research in progress, and ideas for new initiatives, how can we best chart a way forward that takes advantage of the opportunity for the Joints Modelling community to develop a coordinated research strategy? Topics to include:

- What are the options for better modelling? E.g. Supermodels; Model reduction?
- Are there are any radical new ideas for the modelling approach?
- Is the goal (of developing working prediction methods for engineering joints) too ambitious? and, if so, how do we moderate our expectations to make some progress in the immediate future?
- What experimental evidence is required to help define an attainable capability? i.e. how can we determine empirically a realistic level of repeatability of (a) current joints and (b) future joints that we would aspire to provide? What repeatability would be a significant benefit to engineering designs? And how far away from today's norms is that?
- Can/should we define some benchmark structures for round robin testing and/or prediction test cases?
- Is there a need for a Roark's Joints Handbook?
- Should we pursue multi scale issues
- How to differentiate between uncertainties and nonlinearities?

E.3.1 Breakout Session C1

Modeling the Joint

- Predictive model + structure
- Understanding the physics
- Model based on constitutive parameters (without friction experiments)
- Model base on statistical characterization of surface roughness and microstructure
- Use fuzzy algebra to fit simple models to experimental data→Inverse Fuzzy Arithmetic(IFA)
- Length scale effect on contact mechanics (statics+dynamics)
- Transfer micro-properties to macro (averaged) properties
- Database for lots of materials/capabilities
- Physical theory of friction accepted by all

Figure E.34. Summary slide 2 of breakout session C1.

Structural Dynamics of Structures with Joints

- Benchmark problems (computational and experimental)
- Predictive model across different scales/different forcing (harmonic, random, etc.)
- Prediction of joint behavior (linear structure+local nonlinear—multi-scale
- Use global measurements to characterize average joint behavior
- Uncertainty propagation from the joint to structural performance
- Predictive capacity for mid, high-frequency structural dynamics→Uncertainty vs Sensitivity
- Condition monitoring of joints (model updating)—Prognosis

Open Remarks

- Energy-based FEM (Prediction of high-frequency dynamics)
- Combined vibration/wave approach for detecting damaged parts
- Failure prediction for joints—strengthening, coating, repair
- Predicting the evolution of joint properties

Figure E.35. Summary slide 3 of breakout session C1.

Wish List

Five Years

- Database from benchmarks
- Converged FEM in integrated systems with joints in vibration regime
- Better uncertainty quantification

- **Ten Years**
- Everything 10 times better, twice as good

Figure E.36. Summary slide 4 of breakout session C1.

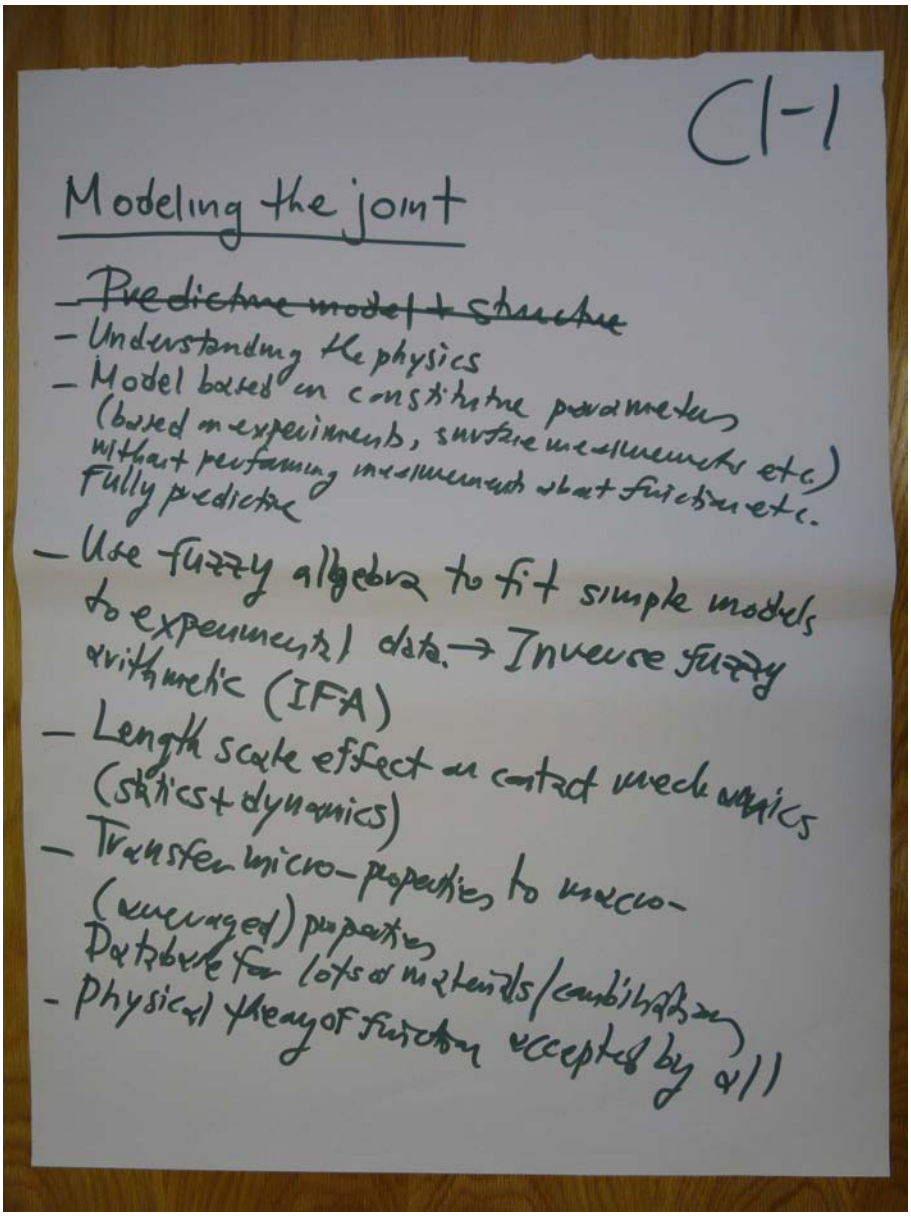


Figure E.37. Flipchart 1 of break outsession C1.

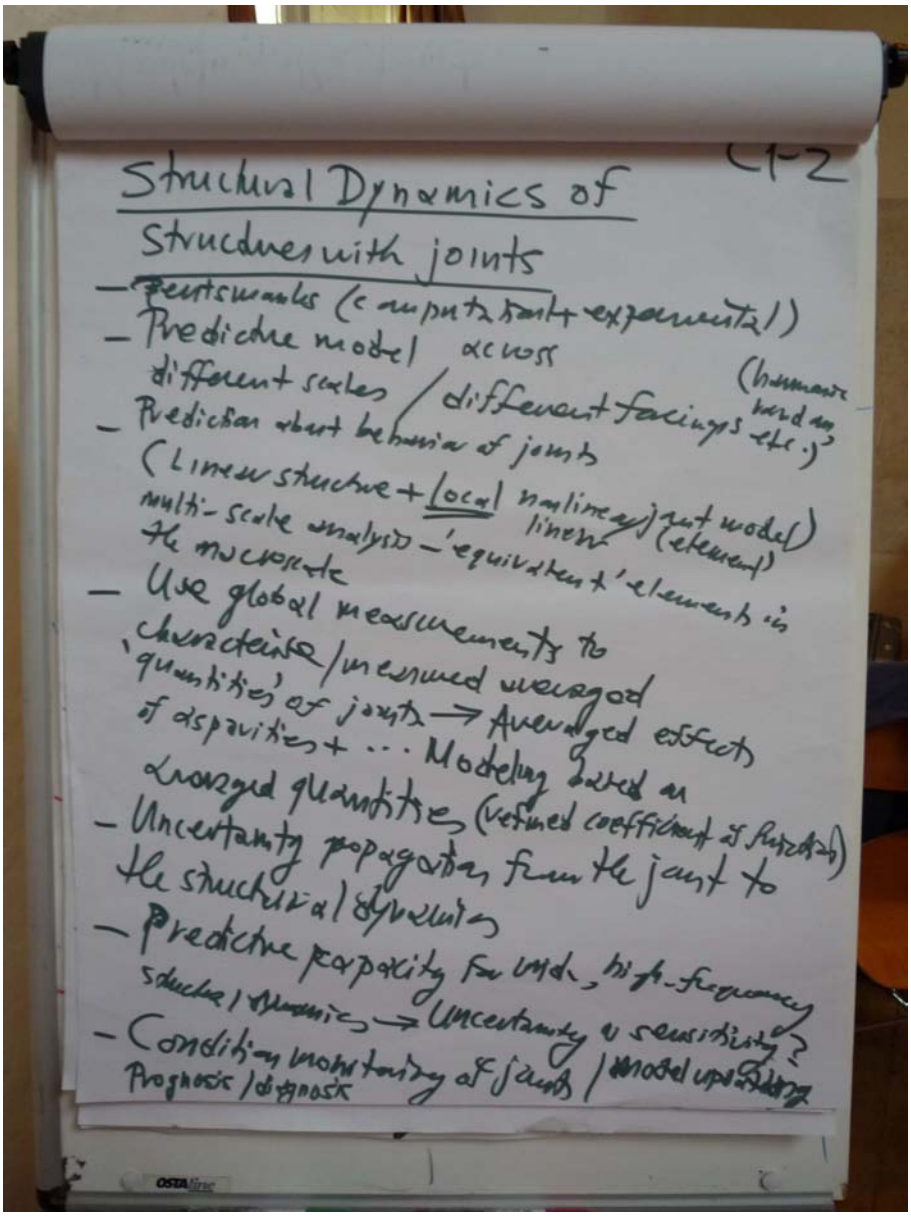


Figure E.38. Flipchart 2 of break outsession C1.

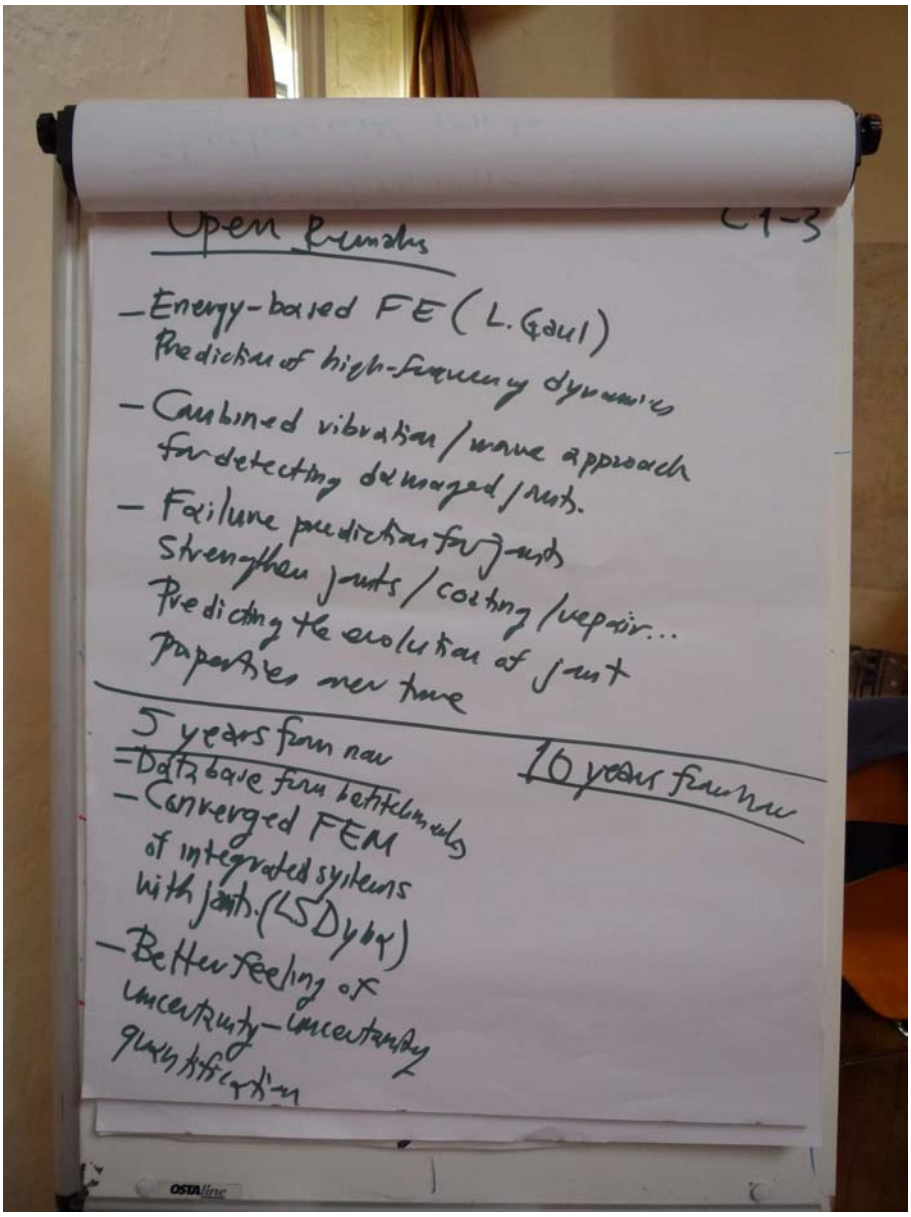


Figure E.39. Flipchart 3 of break outsession C1.

E.3.2 Breakout Session C2

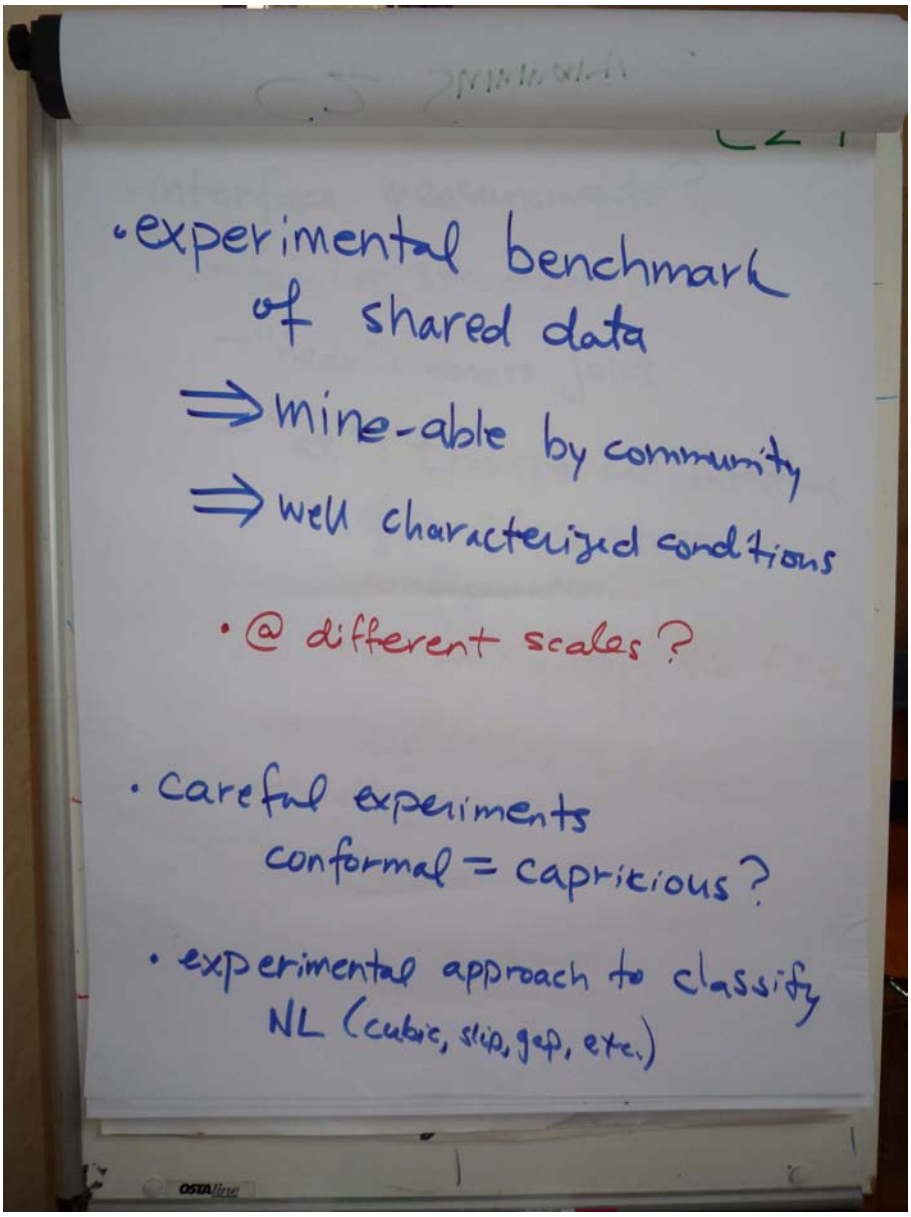


Figure E.40. Flipchart 1 of break outsession C2.

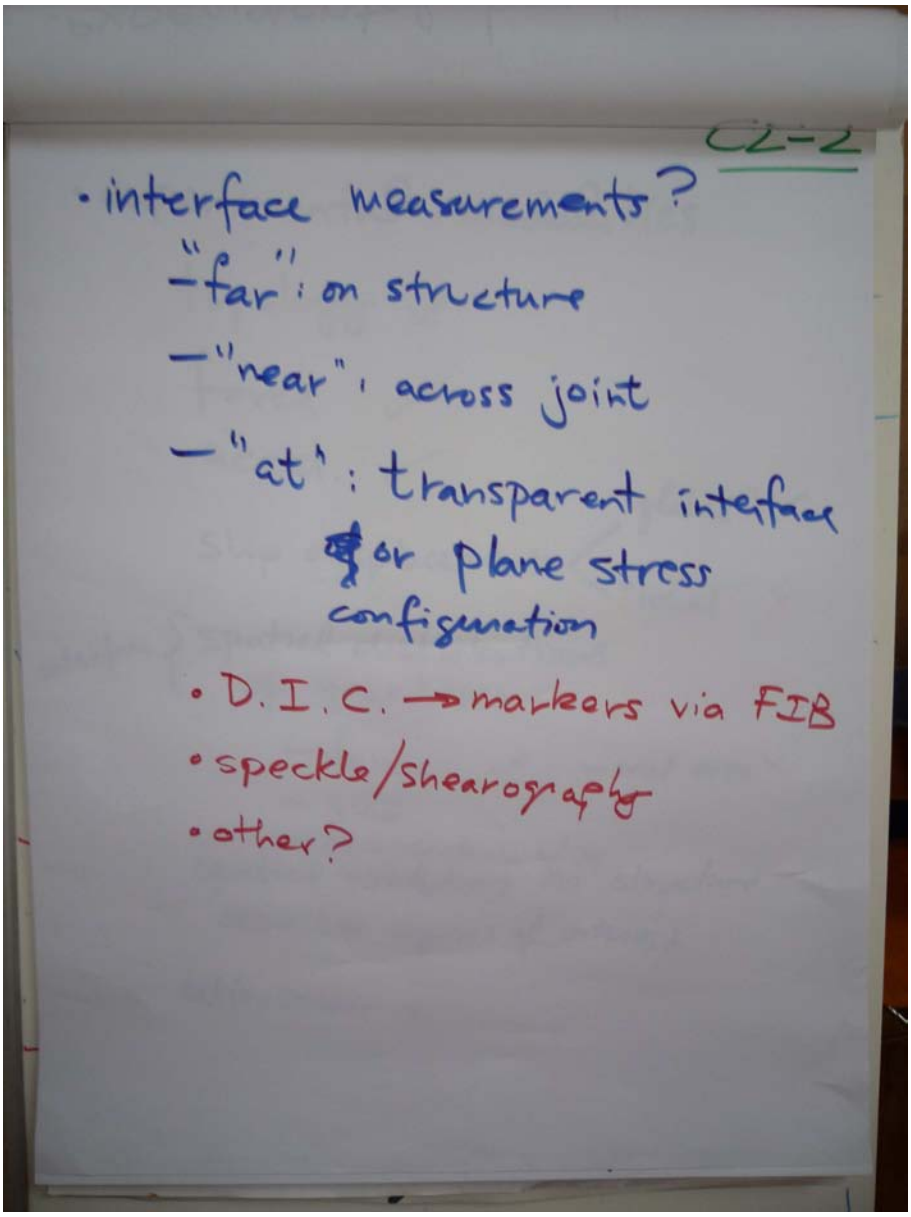


Figure E.41. Flipchart 2 of break outsession C2.

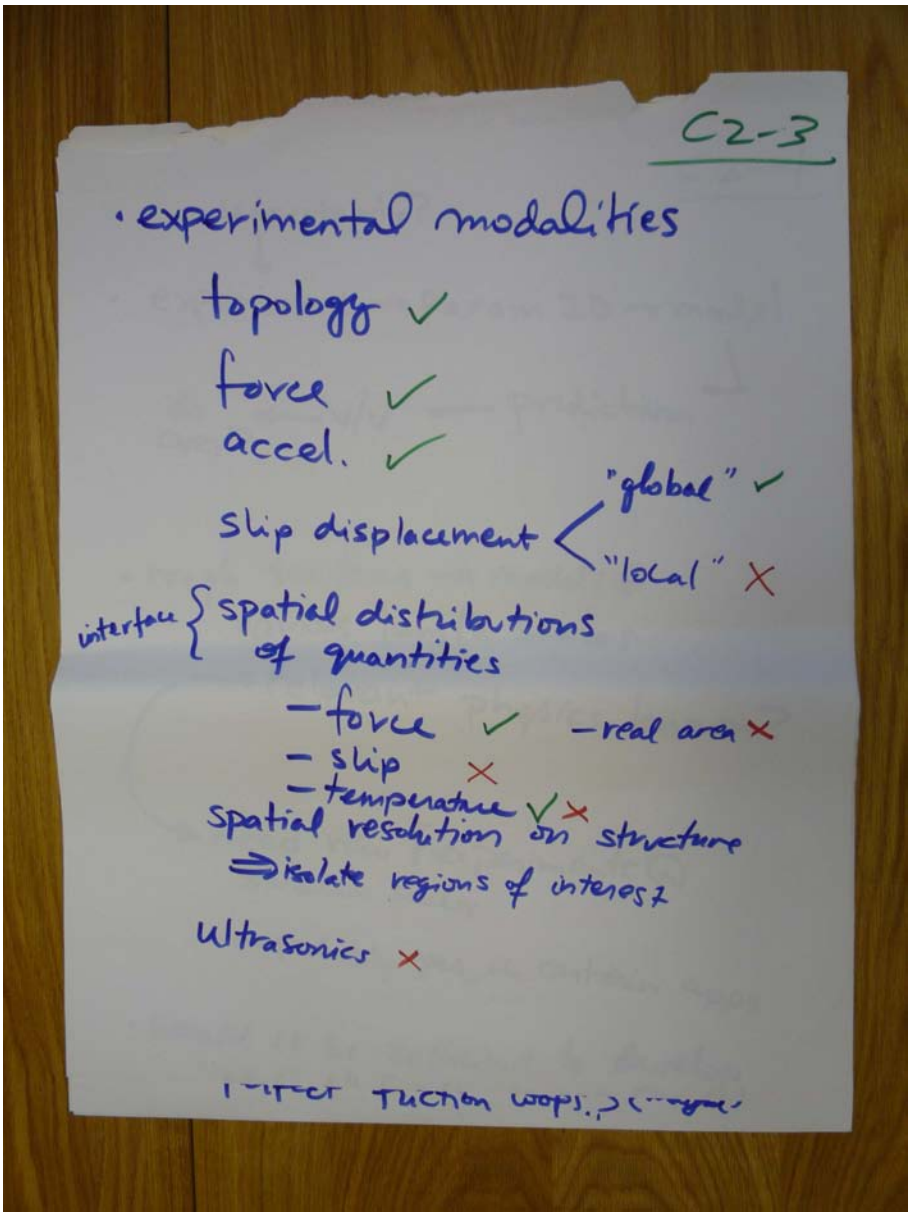


Figure E.42. Flipchart 3 of break outsession C2.

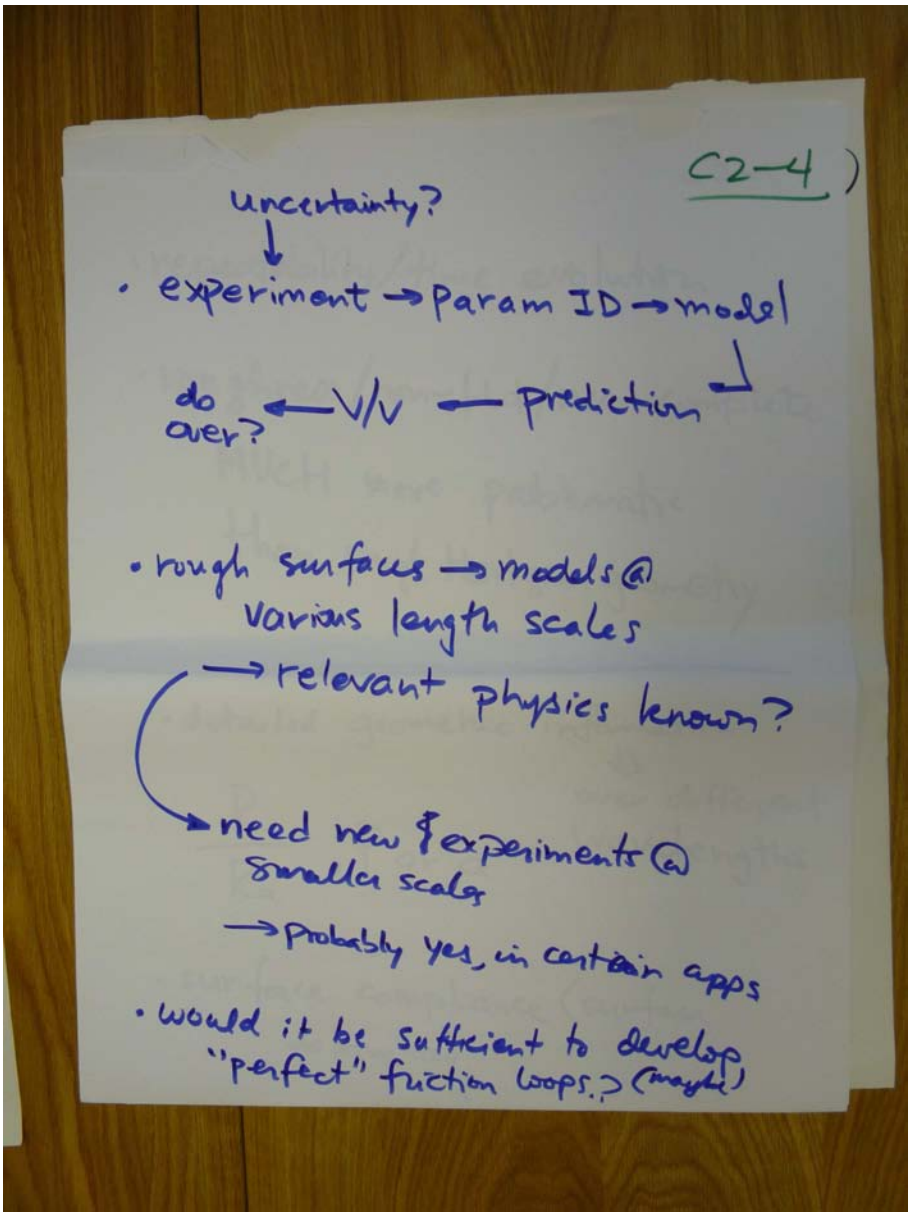


Figure E.43. Flipchart 4 of break outsession C2.

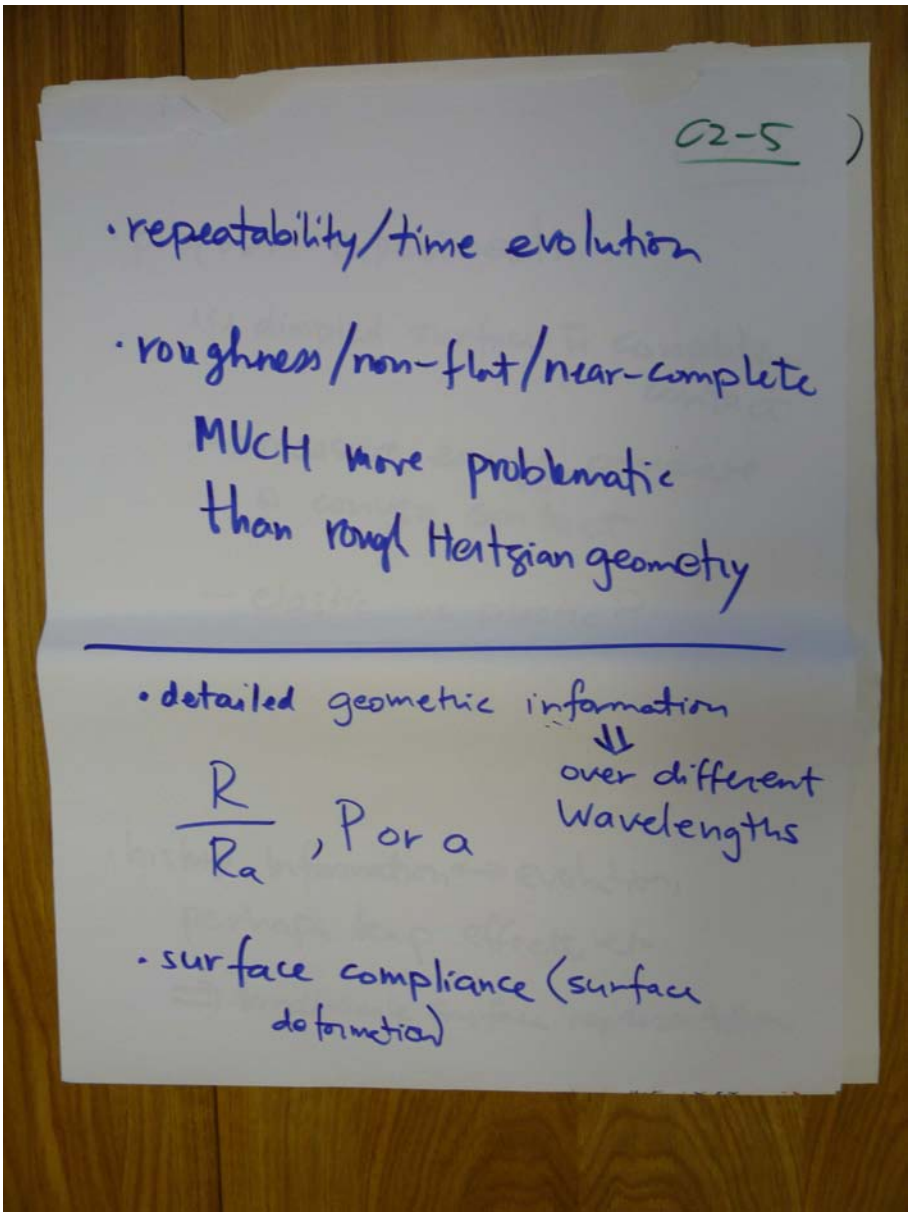


Figure E.44. Flipchart 5 of break outsession C2.

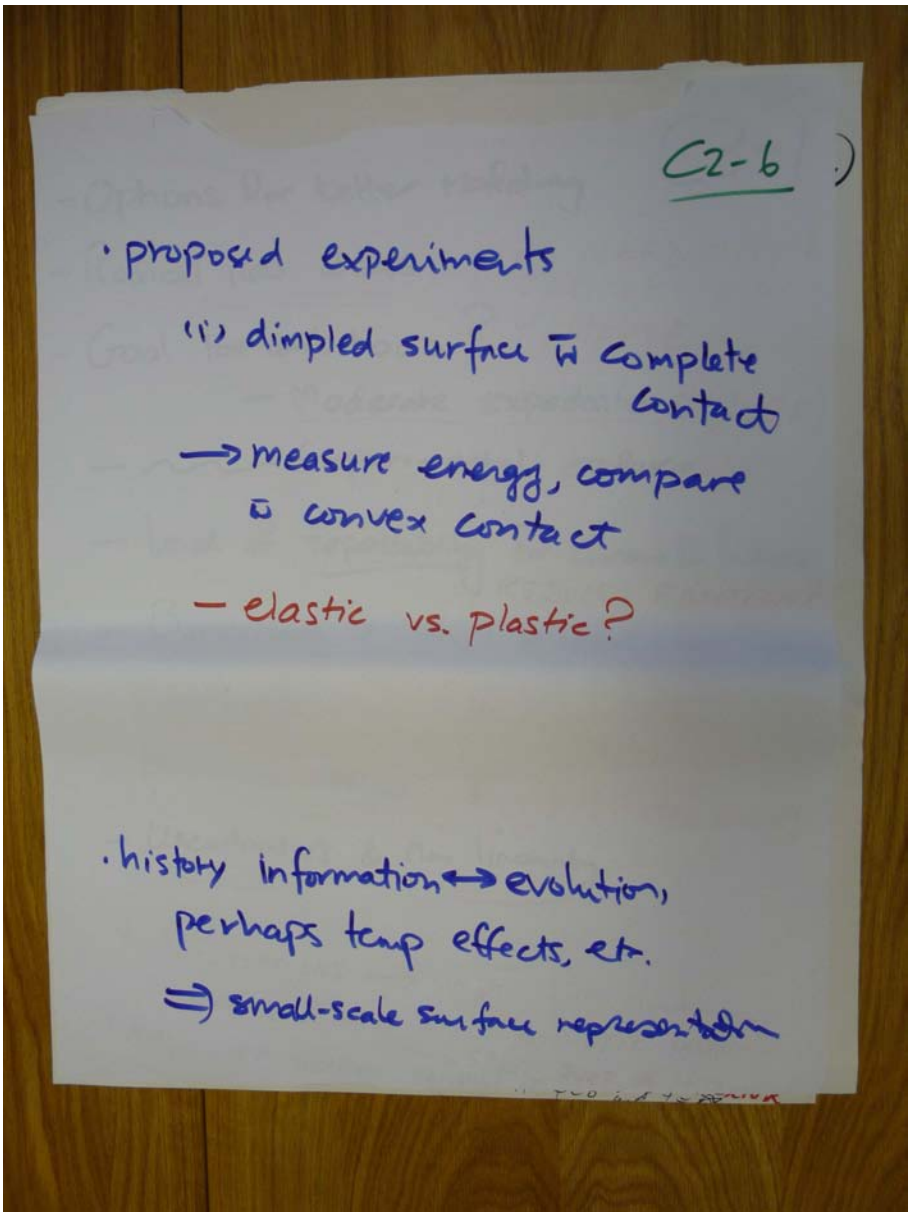


Figure E.45. Flipchart 6 of break outsession C2.

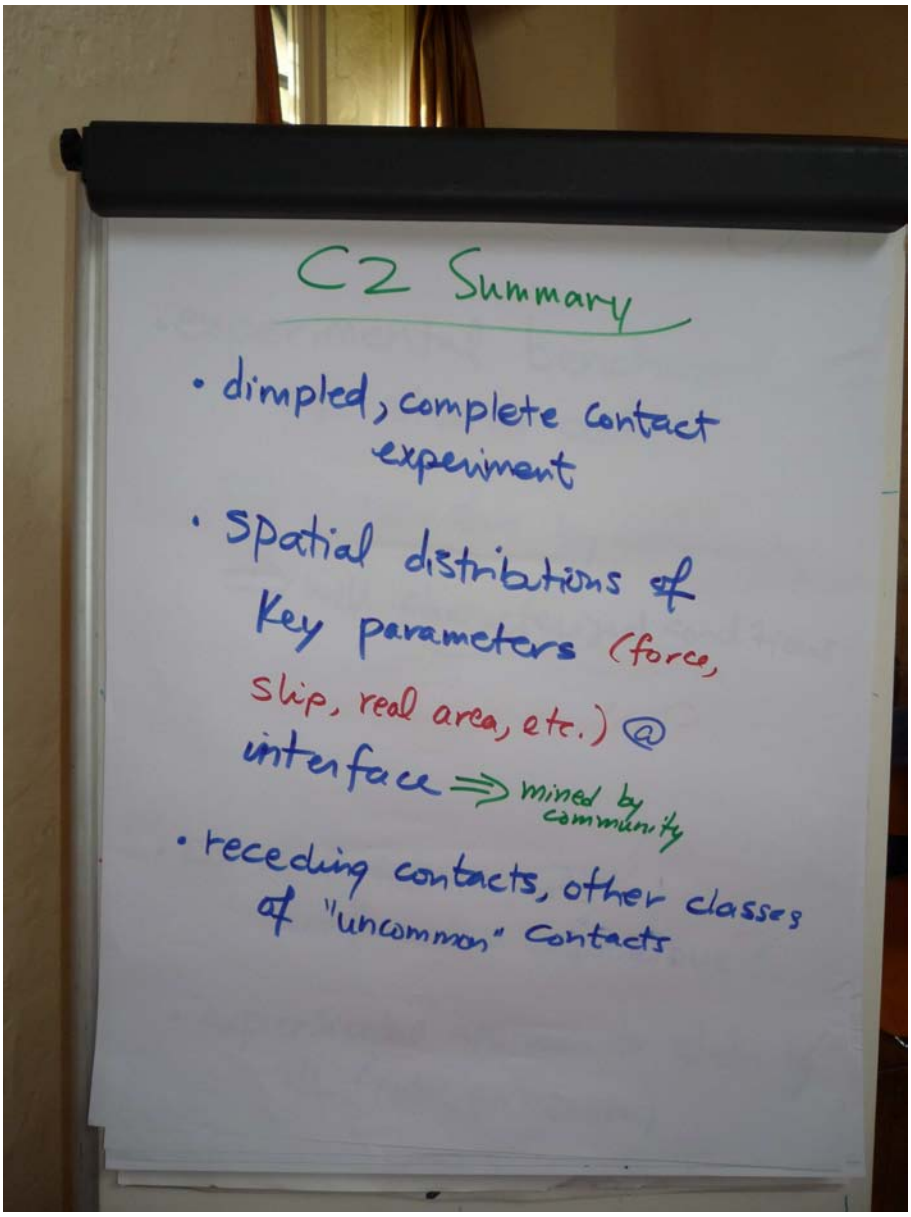


Figure E.46. Flipchart 7 of break outsession C2.

E.3.3 Breakout Session C3

C3 – ideas for new developments. Mike Ciavarella

1) Experimental evidence – a round robin would probably give results for f and hysteresis "all over the places". Discussion follows about "all over the places": agreed that it is not f=0.1 to 10, but somewhere tigther (how much of this is an intrinsic feature of friction even under nominally identical conditions?). This suggests little use of round robin, although it is unclear why deep study of a single lab case would lead us anywhere. 1 better than 10 in being repeatable?
2) Level of repeatability -- see above. Not clear eg if reducing friction would
3) Predictive model: agreed as too ambitious. To moderate expectations make DOE and "some" experiments

Figure E.47. Summary slide 1 of breakout session C3.

5. Handbook. Discussed if only catalogue (problem of impractical number of possible configuration of geometries, material combinations, env conditions). Add algorithms, perhaps under software form. Need to agree on the models
6. Multiscale issue: need of a "framework" of multiple analysis techniques (structural design , structural integrity , roughness, etc) addressing different scales in a consistent manner
7. Uncertainties and "non/linearities". Need to develop models different for small or large number of joints
8. Radical new ideas: return to the "science" of friction. Bowden and Tabor models for energy dissipation surface vs bulk, heat vs phonons, dislocations (contribution from KLJ who however suggests this is unlikely to be of use at engineering level (?)

Figure E.48. Summary slide 2 of breakout session C3.

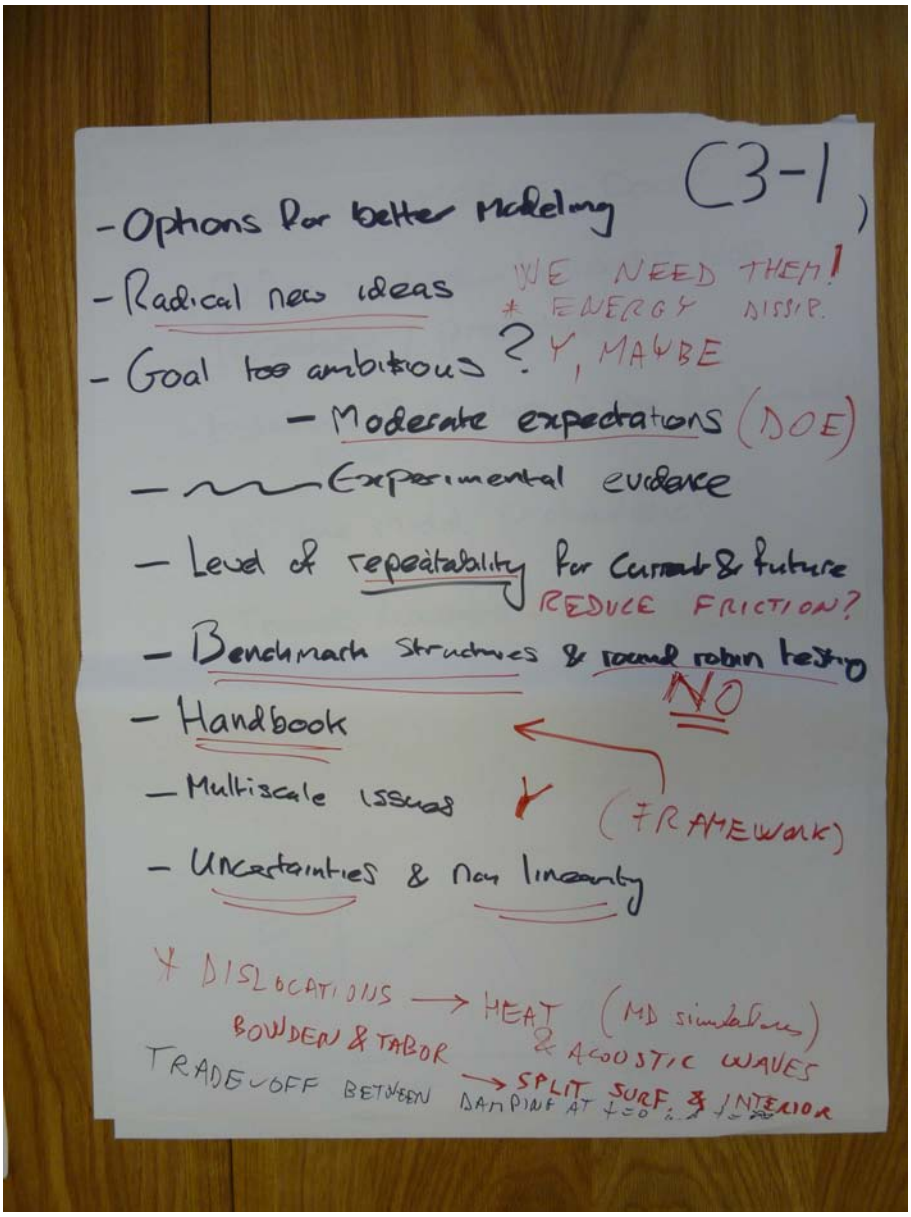


Figure E.49. Flipchart 1 of break outsession C3.

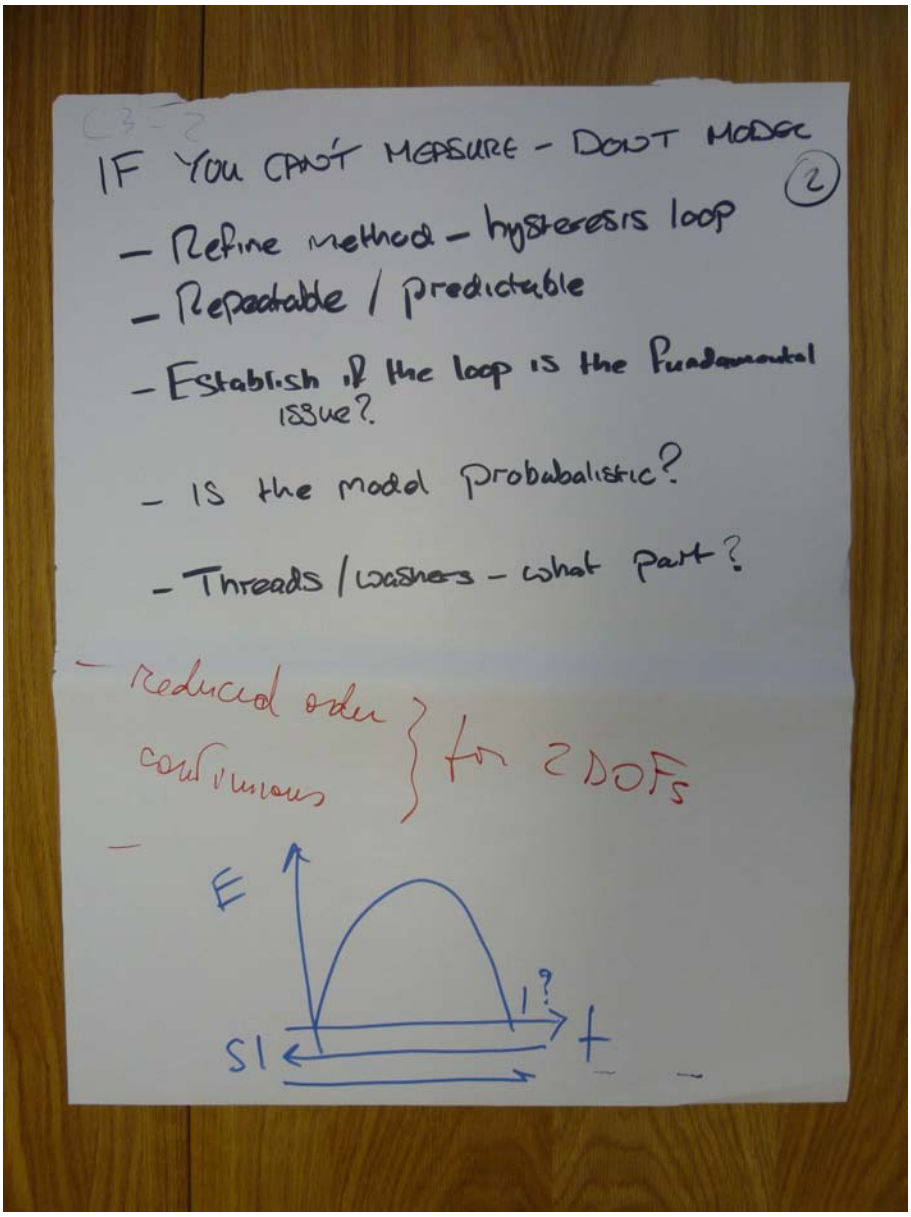


Figure E.50. Flipchart 2 of break outsession C3.

DISTRIBUTION:

10 Professor David Ewins
University of Bristol
Queen's Building
University Walk
Clifton, Bristol BS8 1TR
London
England

10 Professor Lawrence Bergman
306 Talbot Lab
104 S. Wright St.
University of Illinois
Urbana, IL 61801

1 Dr. Adnan Akay
College of Engineering
Bilkent University
06800 Bilkent, Ankara
Turkey

1 Professor James R. Barber,
Department of Mechanical Engineering
The University of Michigan
2350 Hayward Street
Ann Arbor, Michigan 48109-2125

1 Professor Edward Berger,
Department of Civil Engineering
The University of Virginia
Thornton Hall, Office: D203
351 McCormick Road
PO Box 400742
Charlottesville, VA 22904-4742

1 Charles Bodel
EDF - R&D,
1 Avenue du Général de Gaulle
92141
Clamart,
France

1 Dr. Michele Ciavarella,
CEMEC-PoliBA - V.le Japigia 182,
Politecnico di Bari, 70125 Bari
Italy

1 Richard Coghlan,
Rolls-Royce plc,
PO Box 31,
Derby DE24 8BJ,
UK

1 Clark Cooper,
National Science Foundation
4201 Wilson Boulevard
Arlington, VA 22230

1 Dr. Louise Couchman,
ONR
CODE ONR-02
800 N Quincy St
Rm 704
Arlington, VA 22217

1 Jian Ding
Mechanical & Design Engineering
School of Mechanical, Materials and Manufacturing Engineering
University of Nottingham
University Park
Nottingham, NG7 2RD
UK

1 Daniele Dini
Imperial College London,
South Kensington Campus,
London
SW7 2AZ
UK

1 Melih Eriten
Department of Mechanical & Industrial Engineering
University of Illinois at Urbana-Champaign
140 Mechanical Engineering Building MC-244
1206 West Green Street
Urbana, IL 61801

1 Thomas N. Farris
School of Aeronautics and Astronautics,
Purdue University
Armstrong Hall of Engineering,
701 W Stadium Ave,
West Lafayette,
Indiana 47907-2045

1 Greg Garay,
Leader, Damping Technology
GE Aircraft Engines
One Neumann Way MD-A413
Cincinnati, OH 45215

1 Kevin Garraway
Engineering Analysis Group
AWE Aldermaston
Reading
RG7 4PR
England, UK

1 Lothar Gaul
Institut A für Mechanik
Universität Stuttgart
Pfaffenwaldring 9
Zimmer 3.127
Stuttgart 70550
Germany

1 Muzio M. Gola
Dipartimento di Meccanica
Politecnico di Torino,
Corso Duca degli Abruzzi 24
Torino,
Italy

1 Hugh Goyder
Cranfield University,
College Road
Cranfield, Bedfordshire
MK43 0AL
UK

1 Jeffrey Green
Compression Systems (SinA-42)
Rolls-Royce plc, PO Box 31
Derby
DE24 8BJ
UK

1 Dr. Jennifer Gresham,
Air Force Office of Scientific Research
875 North Randolph Street, Suite 325
Arlington, VA 22203

1 Professor David Hills,
Department of Engineering Science
University of Oxford
Oxford, OX1 3PJ
England, UK

1 Norbert Hoffmann
Hamburg University of Technology
Room 0511
Eissendorfer Strasse 42 (M)
D-21073 Hamburg,
Germany

1 Professor Reuven Katz,
100-b Dow Building
Department of Mechanical Engineering
2350 Hayward Street
College of Engineering, University of Michigan
Ann Arbor, Michigan 48109-2125

1 Professor Kyung-Suk Kim,
Division of Engineering,
Brown University,
182 Hope Street,
Providence, RI 02912

1 Professor Tod A. Laursen,
127 Hudson Hall
Box 90287
Duke University
Durham, NC 27708

1 Professor Sean Leen, School of Mechanical Materials & Manuf Eng,
Faculty of Engineering
The University of Nottingham
Room B104 Coates
University Park
NG7 2RD
UK

1 Dr. Donald Leo,
Virginia Tech
3004 Torgersen Hall (0217)
Virginia Tech
Blacksburg, VA, 24061

1 Xianghong Ma
School of Engineering and Applied Science
Aston University
Birmingham
England, B4 7ET
UK

1 Professor Arif Masud
Department of Civil & Environmental Engineering,
University of Illinois at Urbana-Champaign
Urbana, IL 61801

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Mechanical & Aerospace Engineering
P.O. Box 876106
Tempe, AZ 85287-6106

1 Dr. Eduardo Misawa,
The National Science Foundation,
4201 Wilson Boulevard,
Arlington, Virginia 22230

1 Daniel Mulvihill
Department of Engineering Science
University of Oxford
Parks Road
Oxford
OX1 3PJ
UK

1 Ionel Nistor
EDF R&D
1 Avenue du Général de Gaulle
92141 Clamart
France

1 Professor David Nowell,
Department of Engineering Science
University of Oxford
Parks Road
Oxford OX1 3PJ
UK

1 Dr. Evgeny Petrov,
Mechanical Engineering Department
Imperial College London
South Kensington Campus
London, SW7 2AZ,
UK

1 Professor Andreas Polycarpou,
Department of Mechanical and Industrial Engineering
342 MEB,
1206 West Green Street
University of Illinois at Urbana-Champaign
Urbana, Illinois 61801

1 Dr. Marina Popova
Department of Mechanical Engineering
University of New Mexico
Albuquerque, NM 87131

1 Professor Dane Quinn
Dept. of Mechanical Engineering
The University of Akron
Akron, OH 44325-3903

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Dept. of Physics & Astronomy
The Johns Hopkins University
3400 North Charles Street
Baltimore, MD 21218-2686

1 Dr. Lawrence Robertson
U.S. Air Force Research Laboratory
AFRL/VSSV
3550 Aberdeen Avenue SE
Kirtland AFB, NM 87117

1 Surya Surampudi
EOARD
86 Blenheim Crescent
Hillingdon, Ruislip, Greater London
England
HA4 7HB
UK

1 Alexander Vakakis
3003 Mechanical Engineering Laboratory
1206 West Green Street, MC-244
Urbana, IL 61801

1 Dr. David Wang-Kerr
Texas A&M University
Department of Mechanical Engineering
3123 TAMU
College Station TX 77843-3123

1 Professor Kai Willner,
Institute of Applied Mechanics
University of Erlangen-Nuremberg
Egerlandstrasse 5, D-91058 Erlangen
Germany

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