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Proceedings of the Fourth International Workshop on Jointed Structures

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Abstract

The Fourth International Workshop on Jointed Structures was held from October 19-21, 2015, in Dartington, UK. Forty five researchers from both the United States and international locations convened to discuss the recent progress of mechanical joints related research and associated efforts in addition to developing a new roadmap for the evolution of joints research from academic to industrial applications over the next five to ten years. The workshop itself was organized around four themes: applications that can benefit from joints research (applicability), repeatability and variability issues in experiments (repeatability), challenges in developing predictive models (predictability), and potential paths forward (way forward). The outcomes of the workshop are still in progress as the joints community develops a new roadmap for joints research; however, there are many aspects that are related here within. The ultimate goal of this research community is to develop a validated method for the design and analysis of dynamically loaded structures with frictional joints.

Acknowledgement

The authors thank all of the participants of the workshop for their contributions to the discussion and goals laid out herein. Additionally, we would like to thank Sandia National Laboratories, the National Science Foundation, and the Atomic Weapons Establishment for their support of this workshop.

Contents

1. INTRODUCTION TO THE FOURTH INTERNATIONAL WORKSHOP ON THE MECHANICS OF JOINTED STRUCTURES

The mechanics of jointed structures is of fundamental importance for high consequence applications. While joints are ubiquitous in most engineered structures, the cost of over designing them is inconsequential in most applications. In high consequence applications (i.e. applications where optimal design is paramount or where failure is catastrophic), however, mechanical joints are the single largest area for driving improvement in design and structural dynamics [1]. The challenge related to modeling and predicting the dynamics of jointed structures is four-fold. First, predictive models do not exist as the fundamental nature of friction within a jointed interface is qualitatively different than the Coulomb friction model. Second, the nonlinearity created by the jointed interface results in a nonlinear dynamics that linear methods are inadequate for characterizing (such as damping and natural frequency that changes with response amplitude). Third, a further limitation for predicting the dynamics of jointed structures is numerical in nature – systems with strong nonlinearities require very large computational resources in order to calculate the dynamic responses. Fourth, jointed structures exhibit a high degree of variability and non-repeatability, and the source of this uncertainty is not understood.

Over the past decade, an international community has been established to address these four over-arching challenges. In order to focus, guide, and organize this international community, a series of workshops have been conducted. The first workshop was sponsored by Sandia National Laboratories and the National Science Foundation in Arlington, Virginia, 16-18 October, 2006 [2]. A follow up workshop in Dartington Hall, Totnes, Devon, UK 26-29 April 2009 [3] was sponsored by the British Atomic Weapons Establishment (AWE) and Sandia National Laboratories (SNL), and a third workshop sponsored by Sandia National Laboratories and the British Atomic Weapons Establishment was held 16-18 August 2012 at the conclusion of the ASME International Design Engineering Technical Conference in Chicago [4].

Over the first three workshops, a series of challenges were established that identified specific subject areas of pressing importance within the greater research area of joint mechanics. These challenges served to push the limits of what academia can do to develop a new generation of joint modeling techniques. However, the outlook at the start of the fourth international workshop on the mechanics of jointed structures is that these challenges have reached the end of their usefulness, and it is time for a new approach for organizing the international joints community. Thus, the focus of the fourth workshop was on the transition from academic to industrial problems in order for the recent advances to be transitioned into solutions that industry can benefit from.

2. MINUTES OF THE FOURTH INTERNATIONAL WORKSHOP ON JOINTED STRUCTURES

2.1. Organization

The fourth workshop was organized around the four central themes of Applicability, Repeatability, Predictability, and the Way Forward. Priming these four themes, a session was held to introduce the workshop and to discuss the current state of joints research, and a sixth session held at the end of the workshop focused on outside perspectives and discussion for going forward. Additionally, two evening sessions contained short, submitted presentations. All presentations from these eight sessions are included in Section 4. Discussion from each of the four themed sessions is summarized in what follows.

2.2. Session 1 – Applicability

What Are Industrial Needs and Potential Benefits – Technical and Economic – From a New Joints Modelling Capability?

Predictive models of joints can benefit a number of industrial sectors

Critical - willing to support and fund development

- Aviation
- Space
- Defense
- Automotive

Useful - would use advanced modelling capabilities

- Power Generation
- Marine
- Civil
- Electronics

Potential - might find benefit in joints modelling

- Biomedical
- Consumer Goods
- Building Services

Technical capabilities of advanced joint models

Predictable
Efficient
Can be used for design - describes the response (e.g., damping, stiffness, integrity) in terms of the system design parameters

Economic benefits of advanced joint models

	Ability to quantify (1-highly quantifiable)	Economic savings (1-significant savings)
Minimize Testing	1	1
Reduce Inspection	1	2
Reduce Service	1	1
Failure	3	1-3
Recall & Retrofits	2	2
Complaints	2	1

To better understand the industrial needs and desires for joints research, Sections 3.1 and 3.2 directly report the perspectives from researchers at multiple industrial institutions.

2.3. Session 2 – Repeatability

It was globally agreed that there is very high variability in measurements of local joint response in experiments. Some noted that at the system level, the variability in damping or energy dissipation did not have large variability, perhaps because of the averaging effect of many joints in a full system structural response. Several said that standardized tests were needed to reduce the variability that simply comes from poor test techniques or inadequate measurement capability. However, the definition of the proper tests to standardize is still an open question. One group noted that there are several levels of testing, i.e. surface tribology, low level tests to calibrate constitutive models, single joint tests for standard joints, system level experiments and field operation data gathering. We lack a general physical understanding of joints. Some general needs from experiment were voiced multiple times including:

1. Need to measure joint surface geometry to high resolution, and dynamic normal and traction force distributions and displacements in the joint for basic insight and ultimately to validate constitutive models;
2. As an alternative to number 1., a very well controlled experiment using analytically tractable geometry (such as sphere on sphere as opposed to the worst case flat on flat lap joint) and boundary conditions from which measurements can be made with fewer unknowns to gain quantitative information on frictional tractions and displacements.

Some additional ideas were:

1. With all the variability, we don't know how much is epistemic due to lack of knowledge or poor experimental methods and how much is aleatoric based on joint materials, geometry and loading.
2. In experiments, normal loads, forces and/or amplitudes need to be controlled particularly for sinusoidal force testing to take into account or remove harmonics that naturally occur when forcing a nonlinear system with a sinusoid.
3. Whether testing should be transient, sinusoidal, periodic or random is still somewhat of an open question, at least partially dependent on the final application.
4. Load history can affect experimental joint response.

2.4. Session 3 – Predictability

The discussions on predictability were grouped around questions that related to the ability of predicting system behaviour, mainly through approaches based on computer modeling and numerical forecasting. Three main themes emerged from the discussions.

I. Computer based modeling and simulation techniques have to aim mainly towards design.
The approaches have to be useful and have to be applied in design. What has no impact in design does not need to be predicted. Similarly, the accuracies needed, should be viewed from the design perspective too. This will ultimately decide on needs of computing performance, high performance computing, integration of multi-scale and multi-physics aspects, etc.

II. The development needs to go from component level, or a small or few element view, to full large-scale systems level predictions.
While at component level predictability has reached a satisfactory level, for systems set up of smaller or larger numbers of components, prediction today poses a major challenge. Still, due to higher system integration levels and the general drive towards higher system complexity, most substantial progress will be accomplished only when a full systems perspective, and also a perspective to full operational loads, will be taken.

III. There needs to be a better balance of bottom-up, first principles physics based views and approaches and top-down, complex systems, empirical and data based approaches.
At present the two views seem largely disconnected. On the one hand there is the modeling community that follows geometry and physics based modeling, growing models bottom-up following physical principles, sometimes down to ab-initio ideas. The approach suffers from growing inaccuracies and uncertainties in building up larger models. On the other hand there is the community following complex systems empirical approaches, often starting out with the large-scale whole systems perspective. Balancing the two perspectives better will be a future task.

2.5. Session 4 – Way Forward and Applications

Recap of Key Ideas
Some of the key ideas that came out of these discussions include:

- A need for a better terminology. The terms that we currently use to discuss our systems are developed for linear systems. Thus, their applicability to these nonlinear/real systems is questionable.
- The thought of looking at expanding a series of linked beams, starting with a single joint and scaling up to multiple joints, would be interesting to quantify the effect of a single joint in a complex system, the formulation of a joint continuum model, etc.
- Several new ideas include adapting the complex systems research to joints applications, and investigating the potential of slow dynamics for explaining the variability in joints.
- Consensus for better communication between dynamicsists and contact mechanics researchers is needed, as well as better communication between industries to clearly articulate the needs and potential overlap.

- One new open question: how will additive manufacturing affect joints? Already, we've seen that the removal of joints (ex: F22) can lead to problems. We don't have a valid approach, yet, for designing assemblies using additive manufacturing though…
- Guidelines and concrete structures for round robin exercises are needed: the details need to be flushed out and clear. Additionally, we need to share data (experiments and models).
- There are several applicable industries out there that we aren't engaging, especially when friction-induced vibrations are considered, such as is found in the drilling industry for oil and gas.
- A new organizational model is suggested, in which we define themes, or subgroups, that focus on specific aspects of joints research rather than defining actions and challenges for the community with a sense of obligation.

Detailed Discussion of the Way Forward Session

The Way Forward breakout session was preceded by four talks:

- Matt Allen presented on modal Iwan methods. Some of the key thoughts included
    - There are multiple examples of joints causing a problem when they are *removed* from a system, such as the F22's tail wing, and the cowl in GE turbines. Without the joint, there was insufficient damping in the system despite attempts to engineer in high damping
    - With additive manufacturing, joints will be removed increasingly in the future; this introduces the potential for new problems (such as those mentioned above)
    - The proposed analysis framework is thus: 1) Is a system linear? If so, traditional methods. 2) Is a system quasi-linear? If so, uncoupled modal methods (such as modal Iwan). 3) The system is strongly nonlinear, and new tools are needed still.
- Muzio Gola presented on the capabilities of AERMEC
    - For specific details of the 13 test rigs, see Muzio's slides
    - The AERMEC test rigs are developed to have high temperature (1000˚C) for the samples via induction and multiple unique capabilities.
- Norbert Hoffmann presented on complex system perspectives, and touched on:
    - Friction as a self-excitation process (brakes, well-bores, etc.), which has non-standard bifurcations that warrant more analysis
    - Advancements over the LuGre friction model become very messy due to large numbers of variables or state variables
    - Presentation of an "Intensity Chart," which presented data complexity versus modeling complexity (Norbert will provide more details of this in a separate write-up)
- David Nowell presented on physics-based potential approaches
    - Top down approaches tend to be phenomenological, and usually are efficient but limited in scope
    - Bottom-up approaches tend to be behavior based that is extrapolated to real structures, but are often complex and difficult to scale
    - Recent advances in hysteresis measurements have focused on use of new capabilities such as DIC (digital image correlation)
    - Consensus is needed on the quantity to measure/model

With this motivation in mind, the breakout groups were given the guidance that they need to detail the outcomes of the workshop now. What shape should this take though? Thoughts on a roadmap and context for joints research to outsiders are necessary. The summaries of the three breakout groups are

Group A

- Instead of "Actions and Challenges," why not "Themes"? Defining a set of research groups that collaborate, but reducing obligations as this seems to be a turn-off for many to engage.
- We should define several structural benchmarks to illustrate the range of problems (i.e. aerospace, additive manufacturing, automotive, etc.).
- Regarding costs: thoughts on highlighting recurring versus infrequent costs. Often, it may be the case that a designer could live with a high infrequent cost due to the low probability of occurrence; recurring costs, however, are guaranteed.
- Methods that make a better link between contact mechanics and dynamics are needed
- Guidelines for testing, measurements, data processing, and modeling are needed, including putting data online and sharing experiments/models
- Better strategy for testing needed: standard test plans, communication between analysts and experimentalists, etc.
- Prioritization of end-user needs? Simple things that can be completed sooner might be well received
- What are industrial needs?
  - Simplified models that run faster
  - Education of importance of joints and best practices
  - How to translate from one problem to another
- Potential industries include:

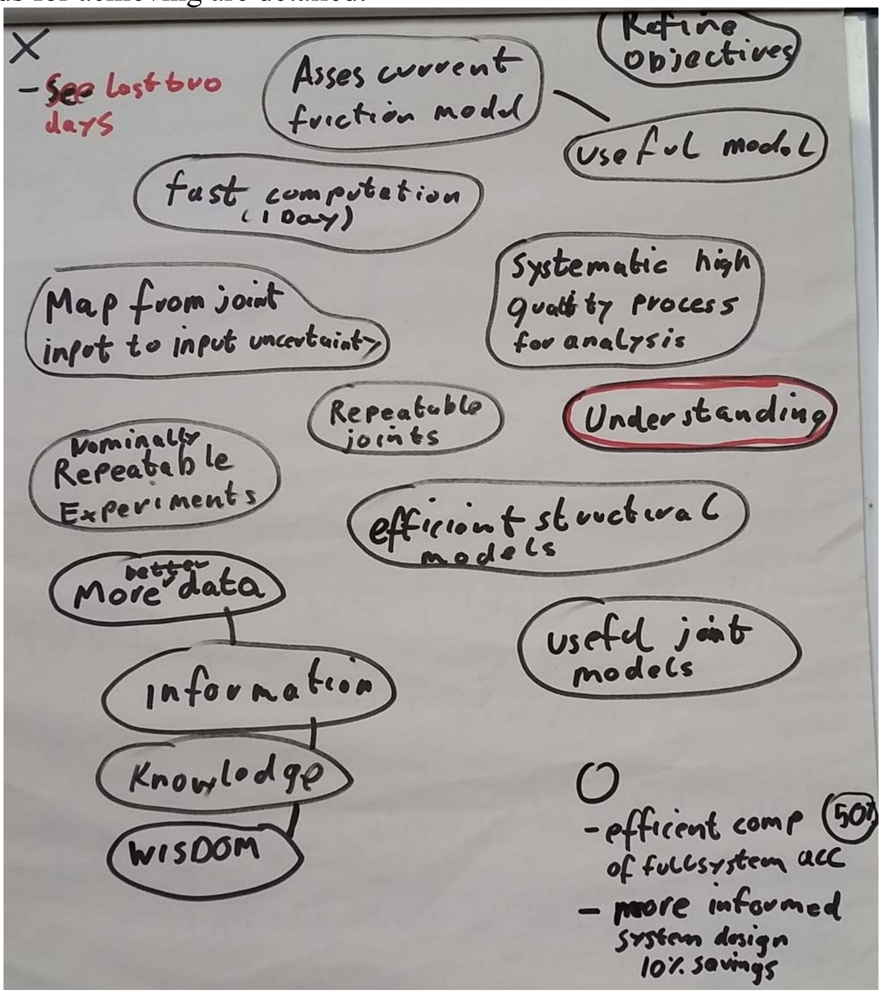
Brake noise	Oil and gas	Machine vibration
Turbines	Tires	Bolted joints
Combustion related joints	Musical instruments	Cables
NVH	Submarines	Automotive applications
Nuclear power	Marine applications	Helicopters
Piping	Control systems/accessories	

- Defining requirements and creating a detailed gap analysis needs another workshop.
- Some potential objectives include:
  - Damping within X%
  - I-Bolt
  - Predicting the onset of macroslip
  - Account for variability in a joint
  - Faster numerical methods for transient analysis (on part with current HBM computational times)

Group B

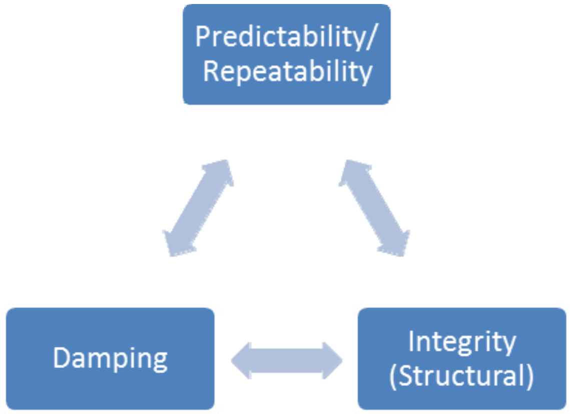
- Focus on funding:
  - Top down perspective beginning with industry and trickling down to workshops, visions, industry-centric roadmaps, more industrial involvement, etc.
- Coherent vision touching on need of potential customers in government and industry
- Statement of how work impacts industry's vision (for perusal by other industries)
- Build a consortium analogous to the GUIDE consortium

- Line of sight within strategy between each sector and research elements most impacted by them
- An attempt at creating a strategy:
    - An example objective: More informed system design that can result in 10% cost savings
    - Methods for achieving are detailed:

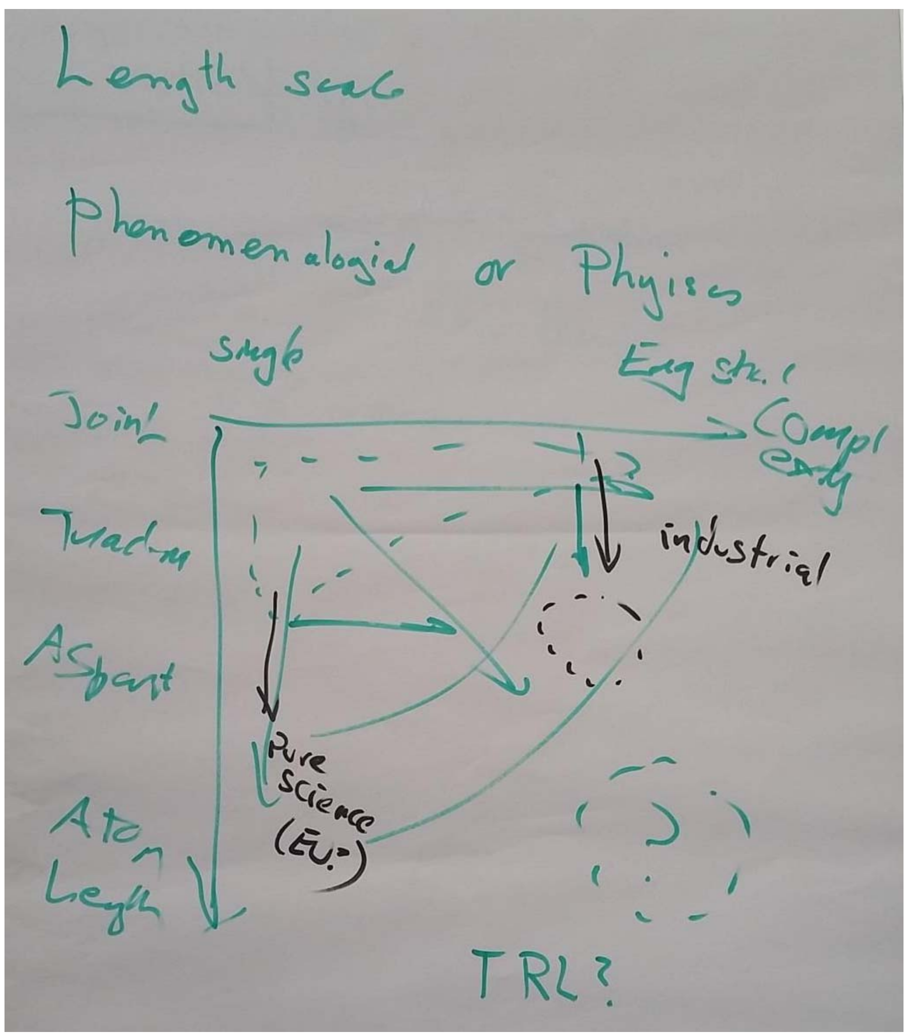


Group C

- Key engineering issues include Predictability/Repeatability, Integrity (Structural), and Damping. These three issues form a triad chart that Christoph will flush out and provide to the group:



- A second chart discussed was length scale versus physics (from a single joint to a system, and from atomistic modeling to structural modeling). David Nowell will flush out this second chart:



- Regarding predictability, what is the best joint? Factors including damping, statics, stiffness…
- Funding: the turbine industry has more clearly defined needs than transient or random applications; dissemination needed to help other industries better formulate their needs
- Regarding the development of the joints group
  - Keep the annual meeting and update at a conference
  - Virtual workshops?
  - Encourage sub-groups
  - More interaction between contact mechanics and dynamicists
  - Use the website more; include email addresses, ping inboxes when updates are published, etc.

Following the discussion of the Way Forward and Applications session, two researchers that are external to the joints community were asked to provide their observations and suggestions to the community after observing the three days of the workshop. Their feedback is summarized here:

Adnan Akay

- We need to be careful regarding our perspective. Our discussions focused on joints, not vibration suppression or failure reduction. Joints are essential elements with primary functional requirements. Joints also have a secondary role in dynamics and damping.
- View joints as complex impedance nodes. They dissipate energy and regulate transmission of energy in the entire structure.
- Uncertainty, variability, repeatability are all related, and they're associated with the secondary aspects of joints
- Regarding physics: it is highly likely that there are important, hidden parameters at the microscale (such as flatness). We tend to integrate the effect of microscale features within the joint, which loses some of the microscale aspects. For example: microslips are unobserved at the macroscale, we can only see the integral of their effect
- Concepts such as loss factor, Coulomb's friction coefficient, and viscosity group many parameters into scalar terms. This is dangerous. We should redefine parameters for our applications. "Words define thoughts."
- What is the source of the nonlinearity in joints?
- We need to consider joints globally. Does the same type of joint have the same influence on a structure regardless of its location?
- A physics based understanding is far off. In the meantime, we can develop new methods for reducing uncertainties and proceeding with uncertainties that we have to deal with.
- A collection of well defined, deterministic sub-models into an assembly constitutes a "complex system." These complex systems exhibit emergent behavior, and there's a whole field of research out there to be applied.

Dan Inman

- Need for a classification of the types of joints, and a classification of application types, including level of importance.
- Focus on damping, define ideal connections as a game-changing application.
- What impact will additive manufacturing have here?

- A simple joint experiment to validate codes against is needed: start with something simple and build up. Define what's good enough: statistical values, bounds, factor of safety, etc. Develop a round robin with protocol, fixtures, parameters being measured, etc. This needs careful thought to ensure that its potential is maximized; otherwise multiple labs can provide varied analyses.
- Can we define a continuum model of joints? i.e. from an expanding series of joints, 1, 2, 5, 10, … , how much does modeling of one joint matter in the entire system?
- Regarding predictability: clarifying accuracy/needs is necessary. We should determine if Coulomb's friction is correct, or quantify how wrong it is.
  - We should organize sub groups within our community.
  - We should quantify the costs of not solving the joints related problems.
  - Often, in funding calls, there's a section for engineering relevance/pathways to impact: how relevant is this work, and how does it affect society?
- Formally have dynamicists, tribologists, and contact mechanicists get together.
- New terms/concepts are needed: instead of coefficient of friction, normalized friction force (instantaneous), hysteresis loop, etc.
- Slow dynamics (e.g. Young's modulus changing over time) may be applicable to joints: how much could this explain some of the sources of nonlinearity?

3. OUTCOMES

3.1. Industrial Importance of Joints

The perspectives in what follows are solicited from select researchers at each company identified. The views do not represent the corporate perspective or opinion on the matter, but rather the opinions and perspectives of the researchers that were engaged.

3.1.1. The Perspective from Select Researchers at Audi

For inclusion in a list of statements and explanations from engineering companies to give explanation to why better understanding of joints is important in engineering.

Short Statement
In the field of noise, vibration, and harshness (NVH) increasing energy dissipation in structures is key to optimize the system's dynamic behaviour. In assembled structures mechanical joints and contact interfaces usually account for most of the dissipation of vibrational energy. Hence, improving the understanding of these coupling elements and establishing guidelines for maximum energy dissipation will allow for the design of more "silent" and robust structures at lower project costs.

Statement
Mechanical joints and contact interfaces are important design elements in the development process of technical systems, especially in the context of noise, vibration, and harshness (NVH) where increasing energy dissipation in structures is key to optimize the system's dynamic behaviour. In assembled structures mechanical joints and contact interfaces usually account for most of the dissipation of vibrational energy. However, design guidelines are missing and a-priori statements about the impact of a specific joint or contact interface on the vibrational behaviour of a structure are difficult or even impossible to make. The reasons for this are manifold. Amongst others, the proper representation of coupling elements in finite-element models and the identification of the corresponding parameters are still challenging. In addition, the underlying effects are not fully understood. However, especially in the field of noise, vibration, and harshness this kind of information could dramatically reduce the development time and enhance the quality of the product. Based on calculated deflection shapes of the system mechanical joints and contact interfaces could be designed to minimize the corresponding vibrations. Thus, starting from the concept phase, i.e. without/with few hardware, the system could be designed in such a way that the vibrational amplitudes during operation remain in the permitted limits. The numerous test runs which obviously always require hardware could be largely replaced by virtual testing. This saves time and reduces costs. Today several iteration loops with multiple hardware modifications are necessary to find a feasible configuration.

The mechanical joints and contact interfaces which are of relevance in the context of NVH problems are rubber bushings, ball joints, frictional interfaces with point, line, or areal contact, and bolted joints.

Summing up, improving the understanding of coupling elements, enhancing their modelling, and establishing guidelines for maximum energy dissipation will allow for the design of more “silent” and robust structures at lower project costs.

3.1.2. The Perspective from Select Researchers at the Atomic Weapons Establishment

For inclusion in a list of statements and explanations from engineering companies to give explanation to why better understanding of joints is important in engineering.

Short Statement
In complex built up structures interface type can have a significant impact on behaviour through dynamic environments. Improved understanding of these interfaces allows for more predictive methods to be developed with less conservatism being required. Interface design based on desired dynamic behaviour rather than just the joints function is desired for the greatest optimisation possible. The test and analysis partnership should most effectively answer specific questions of response and life driving efficiency and confidence.

Statement
The effect of joints within structures remain something that is not fully understood. Currently a structure can be tested and responses measured at selected locations through a variety of technologies, which can build a picture as to the behaviour under specified environments. Test results can be compared against computational models allowing correlation between predictions and test providing evidence for model updating. This process influences the assumptions and level of conservatism applied to the model, but even with an accurate representation of joints being developed for the specific structure, it would be post manufacture.

With an up front detailed understanding of how jointed interfaces affect a structure assumptions can be more correctly made from the concept stage. Allowing the detail of the joint design to be a more comprehensive design feature, the understanding would be based on specific principles rather than the net effect of many things.
Understanding how specific joint types and arrangements impact a structures response can remove the requirement for conservative design features, affect mass considerations, tune for dissipation of energy and ultimately make a model more predictive.

More predictive models give greater understanding of response levels, transfer functions and have significant influences on life predictions. Greater understanding of interfaces in terms of simple relationships and principles allows for representative methods to be developed and translated directly into computational models. Until we know exactly what is happening and why, this level of detail can not be included.

We desire to be in a position where the level of understanding of interface dynamic behaviour allows joints to not only be designed based upon the static behaviour but on control of energy dissipation and response levels. Greater predictive models would aid efficiency of test and considerably increase confidence in the models used alongside them. We require models which predict the severity of response and not just the mode shapes and natural frequencies.

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3.1.3. The Perspective from Select Researchers at Sandia National Laboratories

Overview

From the highest level, the defense systems/aerospace structures that are designed at Sandia National Laboratories primarily are single use systems. Inherent in this single use, however, are a large number of different types of missions that the systems must be designed to survive. For instance, a normal operating environment requires testing of random vibration (at multiple levels), sine sweeps, and large amplitude vibration that represents typical loads encountered during a mission. As a result, the joints within a system designed at Sandia National Laboratories must survive a number of transient load cases. A second issue associated with "surviving" is associated with aging effects: many systems will be, effectively, in storage for decades. During this time, some of these systems may be affected by environmental effects (such as large temperature changes), and it is paramount that environmental aging effects not affect the performance of the system.

Common Joints

Within the systems designed at Sandia National Laboratories there are a number of commonly used joints:

- Bolted Joints
- Tape Joints
- Compression/Press Fit

Bolted Joints

These are joints in the traditional sense that two or more pieces are held together with a squeezing force applied by a bolt or similar fastener. Ultimately the friction force arising from the contact pressure holds the components together, however if large motion is encountered, pinning of the fastener occurs. A sample of a Sandia bolted connection is shown in Figure 1. Many versions of bolted connections are employed, including lap joints, radial joints, and connections such as shown in Figure 1 and discussed in (Segalman et al)

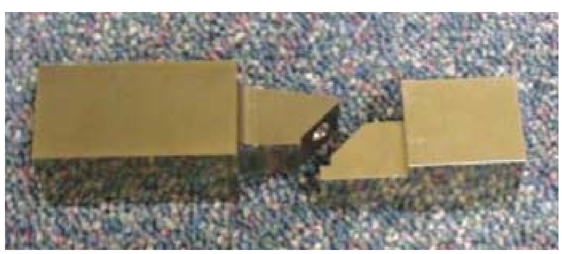


Figure 3.1.3.1 – Traditional Bolted Connection

Typically, the primary function of bolted connections is well understood and well designed for: hold multiple components together. Good design practices have long been established that aid designers in creating bolted connections with the correct stiffness and load properties. However, the consideration of the joint's secondary effects is often neglected during their design. This secondary effect, however, is crucial to predicting how an external force is transmitted to the (relatively sensitive) internal components. As an example, consider a jointed structure which houses an electrical component. When subjected to a large impulse load the joint will undergo macro-slip which drastically reduces the system's stiffness and in turn the load on the component. Were the structure monolithic, more of the load would reach the electronics requiring a more robust design (Hopkins and Heitmann). Without a predictive capability for the secondary effects (i.e. how damping changes as a function of excitation amplitude), multiple costly experiments are necessitated to characterize the response of the system at a number of different load levels. Additionally, possibly catastrophic anomalies, such as fretting fatigue (despite the "single use" nature of systems designed at Sandia National Laboratories), also arise when the overall frictional interface is not considered. A more sound understanding of jointed structures would allow the analyst and designer to work together to design bolted connections that meet all form and functional requirements of the product. Additionally, the connections could be designed such that they fell within the envelope of what is predictable with the state-of-the-art analytical techniques. See (Brake) for more information.

Tape Joints

This type of joint connects components by driving multiple tapered metal bars into a groove formed when the components are brought together (see Figure 2 and (Starr and Segalman)). As the bars slide passed each other in the groove, the taper forces them tightly against the groove walls, hence holding them together. The pressure arising from the tapers holds the components together, but friction plays a large role in the amount of preload achieved.

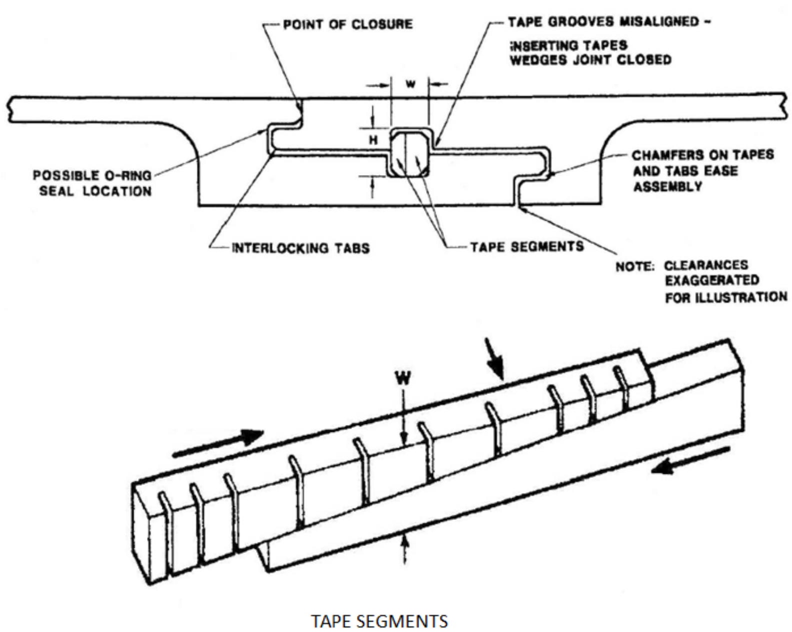


Figure 3.1.3.2 – Schematic of a Typical Tape Joint (Starr and Segalman)

Tape joints have been an important connection mechanism in nuclear weapons since there invention in 1969. These joints rely on the binding of 'tapes' with the pieces in which they are holding together, as shown in Figure 2. This joint type exhibits a bi-linear behavior in axial modes and a cubic nonlinearity when the interface is subjected to bending (similar to a Duffing oscillator). The nonlinearities arise from the contact interface and the effect of loading on the contact pressure distribution within the joint (Starr and Segalman). These joints are important to Sandia products because they allow internal connections to be made without directly accessing the internal space. Additionally, the exterior surface of the components which they connect remains smooth. A lack of understanding of the nonlinear damping and stiffnesses associated with these joints currently prevents leveraging these nonlinearities to the designs advantage.

Compression / Press Fit

This type of joint relies only on the friction force arising from the parts as they are preloaded together. A good example of this is loading D-Cell batteries into a flashlight. The batteries are placed into the cylindrical portion of the flashlight; a cap with a spring is then applied to preload the batteries. Friction between adjacent cells and between the cells and the wall keep the components in place during vibration. These joints often introduce a large, distributed frictional interface that cannot accurately be modeled using Coulomb friction. An example of a press fit is shown in Figure 3. A key difference between the system of Figure 3 and a flashlight is that applications at Sandia National Laboratories tend to have foam encapsulation filling gaps to ensure tight fits, that components are held in place, and that potentially damaging amounts of energy are absorbed by the foam instead of the components.

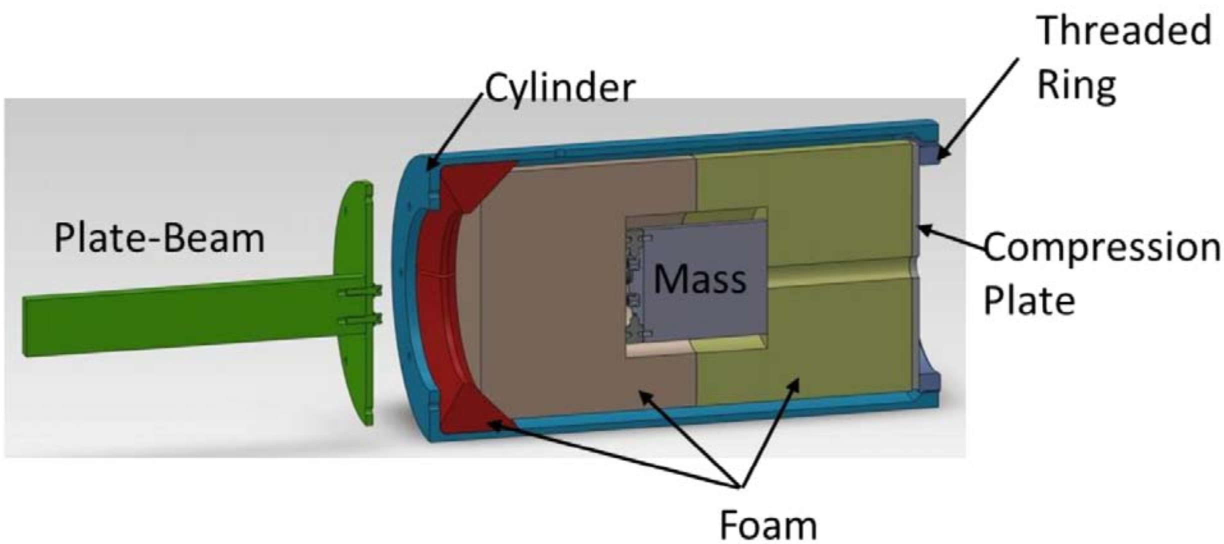


Figure 3.1.3.3 – Press Fit Joint Example (Mayes, Pacini and Roettgen)

Compression and press fit connections are designed into systems where traditional bolted connections are not possible. These types of interfaces introduce nonlinear stiffness as well as nonlinear damping into the system. They differ from bolted connections in that they often have larger contact areas and have no direct kinematic constraints holding the constitutive components together (such as a bolt). They rely solely upon contact pressure and friction to hold the components together. These types of connections not only have micro-slip and macro-slip, as the traditional bolted connection, but suffer more from the components banging into adjacent components and dissipating energy. Sometimes, these types of connections tend to be less sensitive to manufacturing tolerances because elastomer pads placed between the components lessens the tolerances effect; however, in other instances these types of connections are very sensitive to manufacturing conditions due to the potting process. In all cases, these connections tend to be sensitive to preloading, as varying preload levels changes the contact pressure distribution and therefore changes load paths.

References for Further Reading

Brake, Matthew R. W. (Ed.) The Mechanics of Jointed Structures, Springer, 2017.

Hopkins, Ron and Lil Heitmann. "A Method to Capture Macroslip at Bolted Interfaces." IMAC XXXIV. Orlando, FL.: SEM, 2016.

Mayes, Randall L., Benjamin R. Pacini and Daniel R. Roettgen. "A Modal Method to Simulate Typical Structural Dynamic Nonlinearity." IMAC XXXIV. Orlando, FL.: SEM, 2016.

Segalman, Daniel J., et al. Handbook of Dynamics of Jointed Structures, SAND2009-4164. Albuquerque, NM: Sandia National Laboratories, 2009.

Starr, Michael J. and Daniel J. Segalman. On the Nonlinear Dynamics and Quasi-Statics of Tape Joined Structures, SAND2012-6527. Albuquerque, NM: Sandia National Laboratories, 2012.

3.2. Industrial Requirements of Joints

The following lists of requirements are solicited from select researchers at each company identified. The views do not represent the corporate perspective or opinion on the matter, but rather the opinions and perspectives of the researchers that were engaged. Due to the high amount of overlap between the requirements of different agencies, the lists have been combined into a single set of industrial requirements.

Short term requirements

- Prediction of the onset of macroslip in assembled structures with complex loading (accuracy within a factor of two)
- Reduction of variability/unrepeatability observed in experiments
- Extension of current theories to large contact area interfaces (e.g. compression pads, etc.)
- Maturation of existing methods/theories to high TRL level (e.g. modal Iwan, experimentally derived CB, etc.)
- Refinement of models such that once calibrated, are predictive for other environments (excitation amplitudes) to within 5% for stiffness and 20% for damping
- New solver methods and ROM techniques to more efficiently predict the response of nonlinear structures
- Maturation of techniques for quantifying and propagating uncertainties (e.g. SROMs, Soize’s method, etc.)
- Nonlinear system identification techniques for UQ, calibration, and validation
- To know what amplitude predictions are either consistently conservative or under predicted

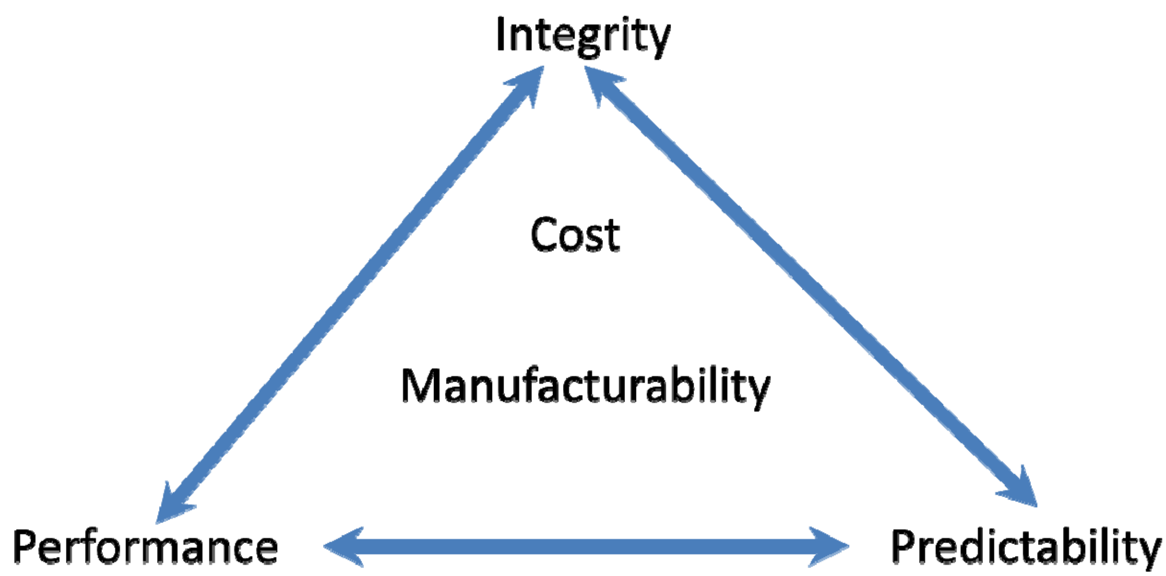
Long term requirements

- In situ techniques for measuring joint properties/behaviors/forces
- Development of joints that are less sensitive to manufacturing variabilities
- Development of models that can predict the amplitude dependent stiffness and damping of an assembled system
- Development of models for jointed structures under random loading
- To predict amplitude and frequency within a jointed structure to the confidence seen within unit-to-unit variability
- The ability to influence the selection of joint type and arrangement based on desired system level responses and damping characteristics

3.3. Visualization Tools and Their Descriptions

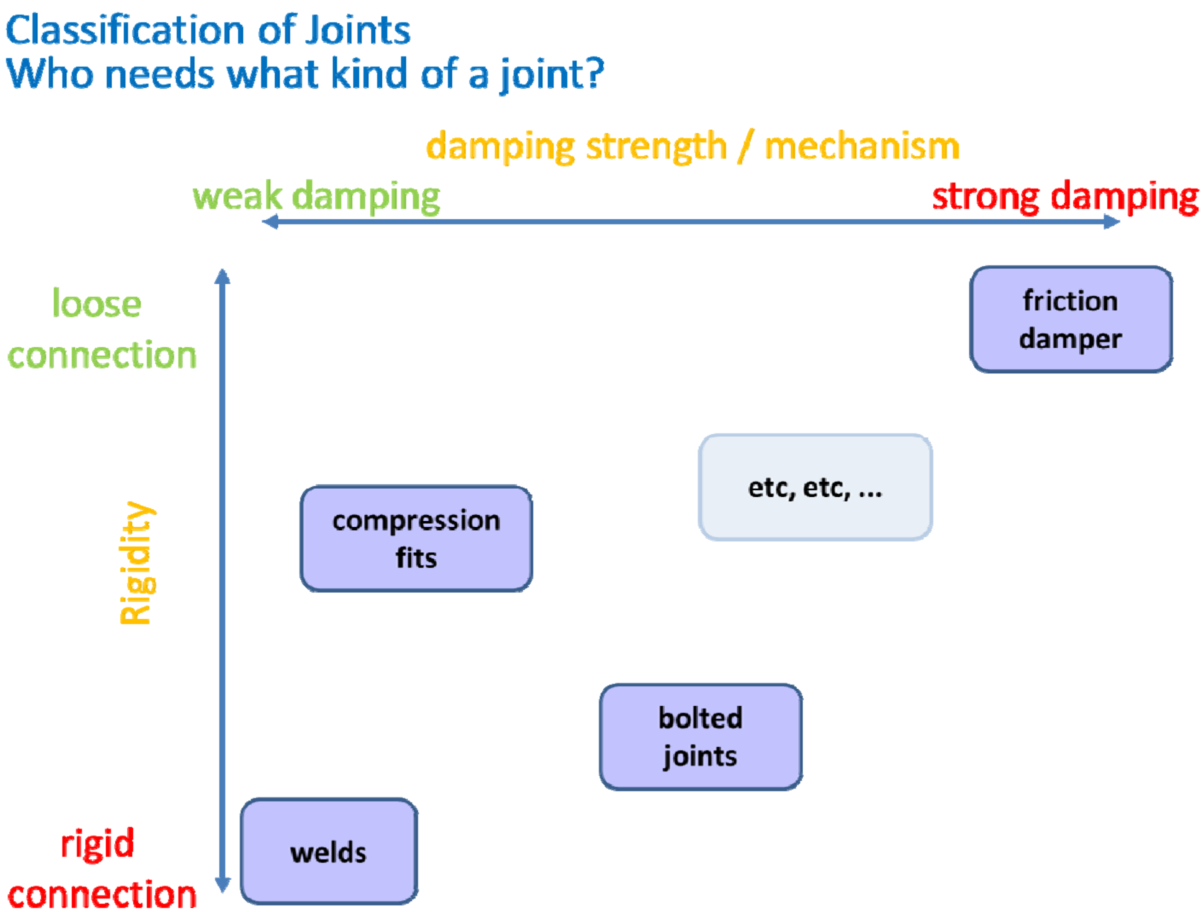
During the presentations and subsequent break-out sessions, a number of useful diagrams were developed that merited further attention. A select number of these diagrams are reproduced below.

3.3.1. Venn Diagram of Requirements from a Bolted Joint



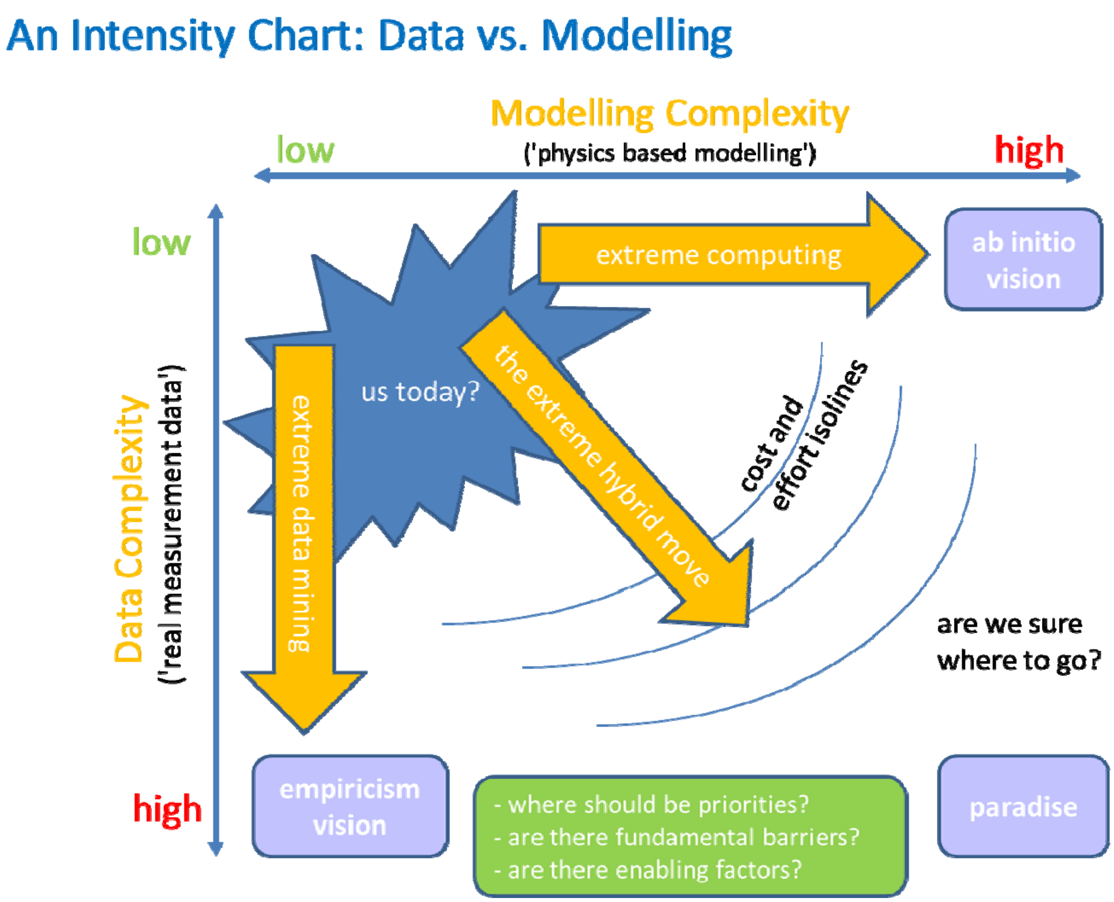
Integrity, performance and predictability are the three requirements for a bolted joint. The static design of joints currently provides good static integrity. For dynamic performance we additionally require beneficial damping as well as dynamic integrity. Dynamic integrity includes other capabilities such as fatigue life and tolerance to transient events. Good predictability implies that the joint will be repeatable when it is replicated many times. Dynamic performance may require compromises between integrity, performance and predictability. Overall issues of manufacturability and cost must also be considered and these may also drive compromises.

3.3.2. Categorization of Damping versus Rigidity in Joints



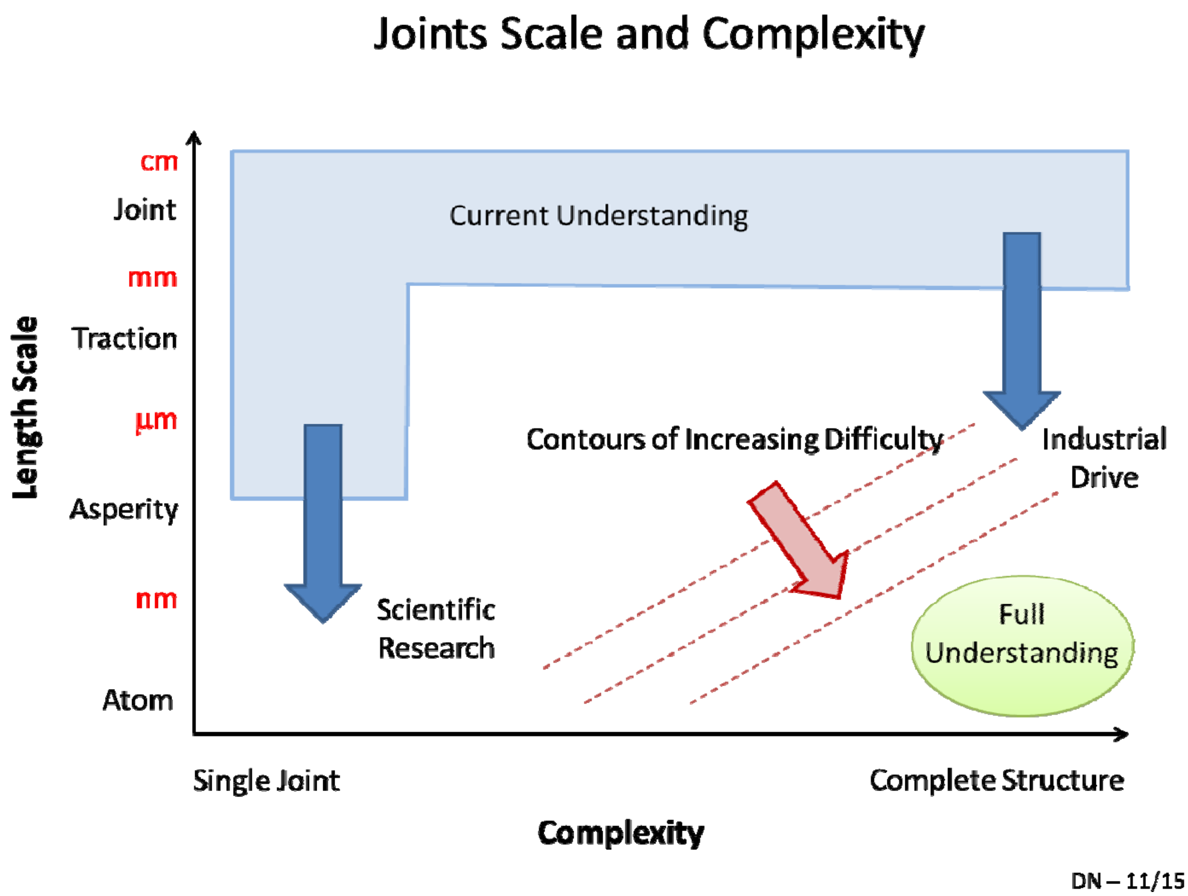
This classification originated in answering the question of "who needs what kind of joints?" In flushing out this chart, a better view of the role of each type of joint is able to be discerned.

3.3.3. Intensity Chart of Data versus Modeling



This chart, originally from Norbert Hoffmann's presentation "Structural Dynamics from a Complex Systems Perspective: Are We Missing Something Out Here?", illustrates the trade-offs in data and modeling complexity. In the extreme of high data complexity and low modeling complexity are data mining applications such as geosciences. It is possible that future empirical joint models will reside here. In the other extreme of high modeling complexity and low data complexity, is extreme computing applications, such as weather forecasting. It is possible that first principles based models for joints will reside here. The ultimate goal is a hybrid approach that is rich in both data and modeling complexity. The open question, though, is where to go from our current state?

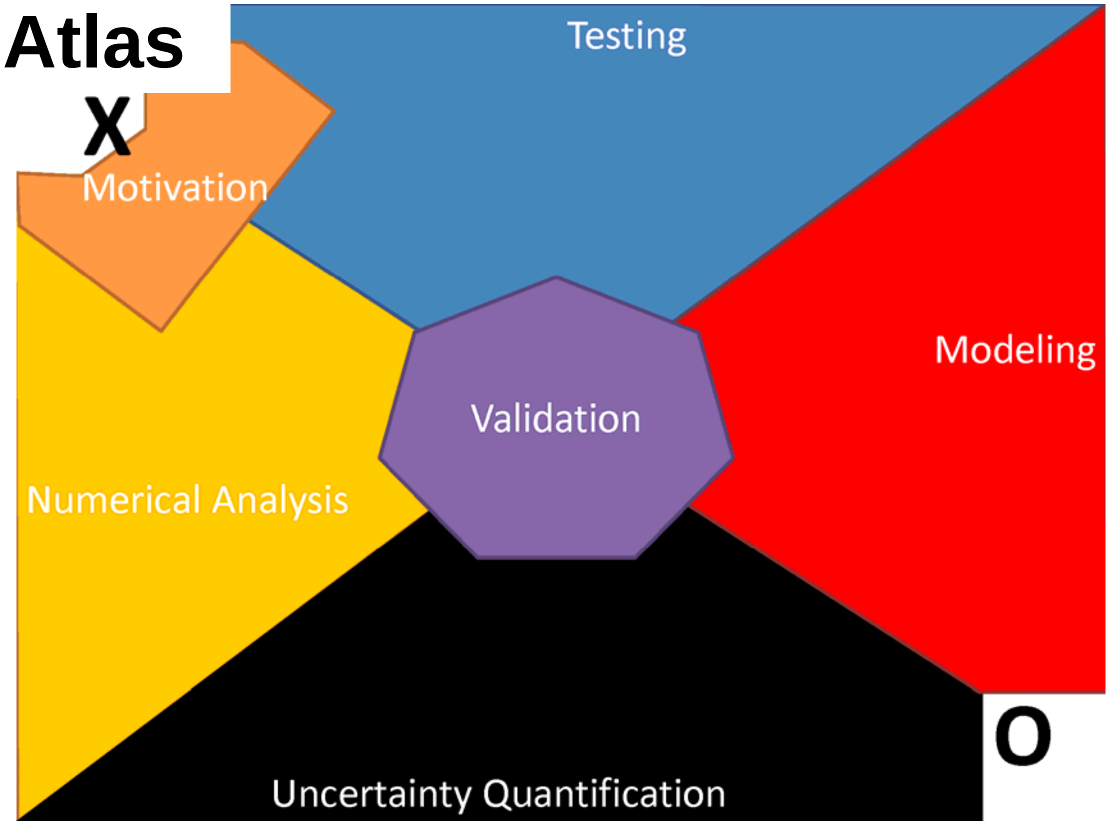
3.3.4. Joint Scale and Complexity Relationships



The diagram presents the frictional joint space plotted against two axes. On the vertical axis is length scale, ranging from nm up to cm. Also labelled on this axis are length scales at which analyses may be carried out (atomic/asperity/continuum traction/whole joint). The horizontal axis plots 'complexity', ranging from a single joint to a complex structure incorporating many joints. The objective (plotted in green) is full understanding (where a complete structure could be analysed correctly at an atomic level). Current understanding is plotted in blue: Complete structures can be analysed at the joint level, and single joints down to asperity scale. The drivers of pure scientific research (to smaller length scales for a single joint) and industrial need (more understanding for a complete structure) are shown with blue arrows. Also shown in red are contours of increasing difficulty (in addressing the problem).

3.4. Draft of the New Roadmap

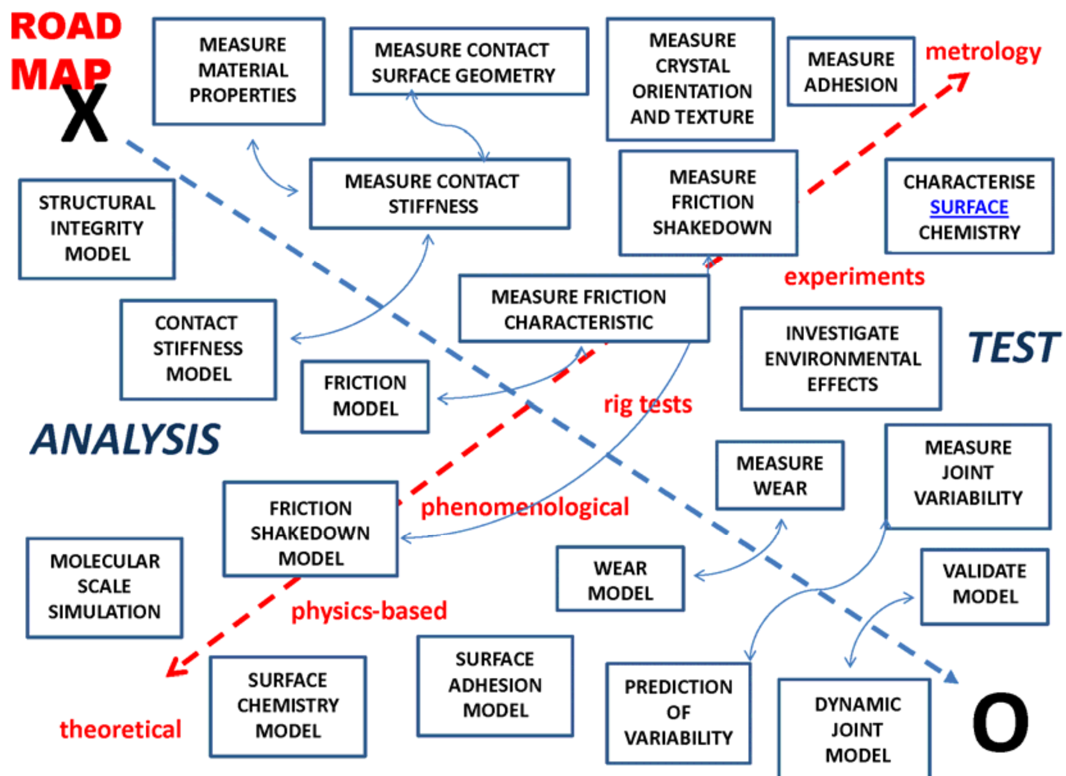
As of publication, the new roadmap (and associated declaration) is still a work in progress. The current version of the roadmap focuses on three different levels: a high level understanding, termed an atlas; the roadmap itself outlining all of the major goals and challenges, and a street map for each goal/challenge with sub-goals detailed. These three levels are illustrated in order:



As outlined in [5], a series of seven themes are suggested for the Atlas:

- Building external consensus for support;
- Experimental investigation of repeatability and variability;
- Techniques to characterize/identify nonlinearities;
- Constitutive model development;
- Numerical methods for nonlinear dynamics;
- Multiscale investigation of interfacial physics; and
- Uncertainty-based strategies for modeling and experiments.

These seven themes are intentionally chosen to be broader than joint mechanics in order to help tie in related veins of research to the efforts of the present community. By making the themes broad, the hope is that the challenges highlighted by the Atlas will be more approachable by a wider community than is currently engaged.



JOINTS MODELLING ROADMAP 2015: V3.2 7-12-15

The roadmap is still a work in progress. Two potential views of it are that it could either contain all of the sub-goals of the focus areas listed in the Atlas, or it could be developed for each specific goal listed in the Atlas. The version above is developed to contain many of the sub-goals for all of the focus areas listed in the Atlas. Highlighted by this roadmap are the central elements of a strategy, discussed by David Ewins during his workshop opening presentation. A strategy is defined to have four components:

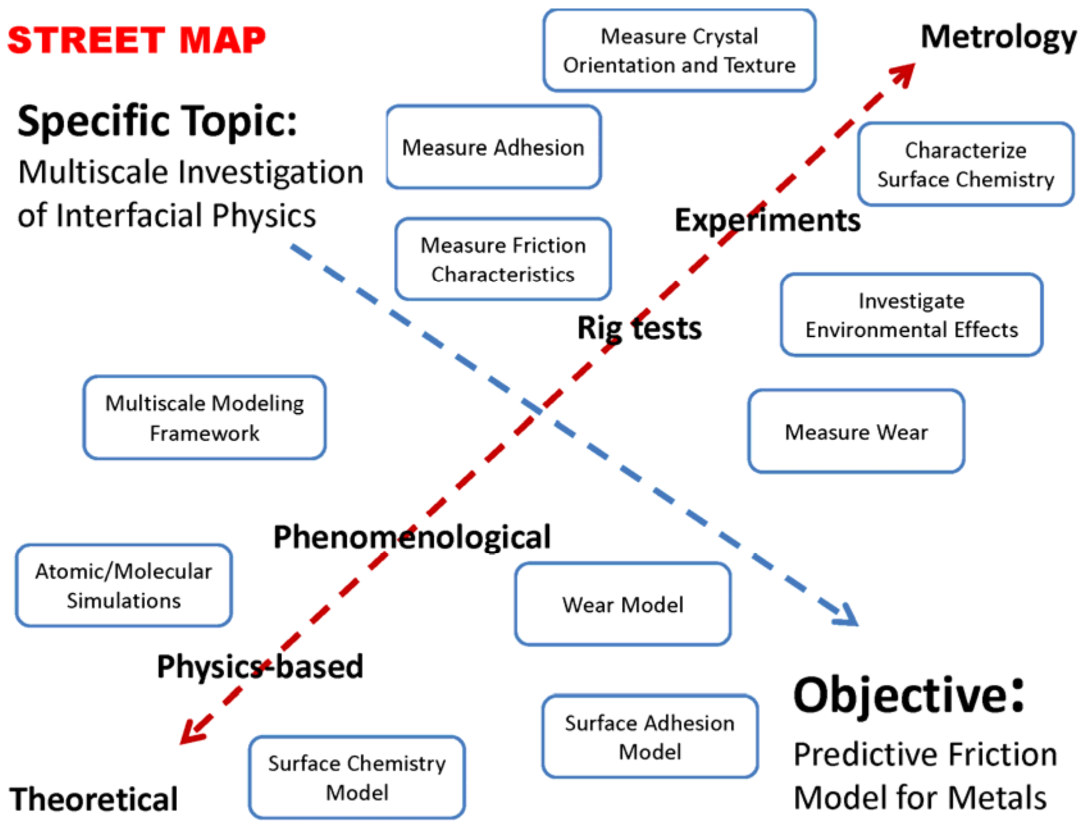
1. A comprehensive understanding of the current state;
2. A clearly defined objective;
3. Possible tools to lead from the current state to the objective; and
4. A plan for how to use those tools.

The first component, a comprehensive understanding of the current state, is summarized by [5]. The objective of joints modeling could be:

> ***A validated method for the Design and Analysis of dynamically loaded structures with frictional joints.***

The roadmap, is thus, the list of possible tools to lead from the current state to the objective. "X" in the above diagram is the current state, "O" is the objective. While there are many tools and paths possible for ultimately achieving the objective, it is not year clear what the possible plan will be. This is to be further flushed out in the final outcome of the workshop, the Dartington Declaration, when it is published.

Lastly, the third layer of the roadmap is the streetmap, which highlights specific tools to achieve a specific sub-goal. For illustrative purposes, the sub-goal of multiscale investigation of interfacial physics is highlilghted:



3.5. Action Items

Following the conclusion of the workshop, a series of 13 action items were issued and assigned to various participants. The complete list follows:

1. Flush out graphs and provide a one paragraph description of them:
Hugh Goyder (the Venn Diagram),
Norbert Hoffmann (both damping versus rigidity and the graph from his presentation),
Christoph Schwingshackl (the stiffness-damping-integrity graph),
David Nowell (the physics/length scale graph)
This is due no later than November 15th.

2. Detailed information/brief write-up of the economics of fasteners under dynamic duty
Hugh Goyder

3. 1-2 Page summary of the break-out session overviews
Dane Quinn: Session 1, Applicability
Randy Mayes: Session 2, Repeatability
Norbert Hoffmann: Session 3, Predictability
Matthew Brake: Session 4, Way Forward
This is due no later than November 15th

4. List of Industrial Requirements
Ed Green

Dan Brown (complete)
Matthew Brake
Merten Tiedemann
This is due no later than November 15th

5. Global Roadmap
David Ewins
David Nowell

6. Explore dessimination through NAFEM
Hugh Goyder
David Ewins

7. Update website capabilities to include automatic updates and journal paper lists
Pablo Tarazaga

8. Provide Industrial presentations to M. Brake and D. Ewins
Dan Brown
Ed Green
Merten Tiedemann

9. Industrial description of why joints are important
Dan Brown
Ed Green
Merten Tiedemann
Matthew Brake

10. Images for the website (i.e. research/application related visuals)
All.

11. Dartington Declaration (several pages)
Executive+ committee

12. Update to entire community
Matthew Brake

13. Photos of the workshop to M. Brake
All.

4. ATTENDEES

The workshop was attended by 45 researchers from Europe and the United States. These researchers included a mixture of professors, industrial researchers (including government labs), and graduate students. The complete list is:

Name	Affiliation	Email
Akay, Adnan	Bilkent University	akay@bilkent.edu.tr
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Tiedemann, Merten	Audi	merten.tiedemann@tuhh.de
Vakakis, Alex	University of Illinois, Urbana-Champaign	avakakis@illinois.edu
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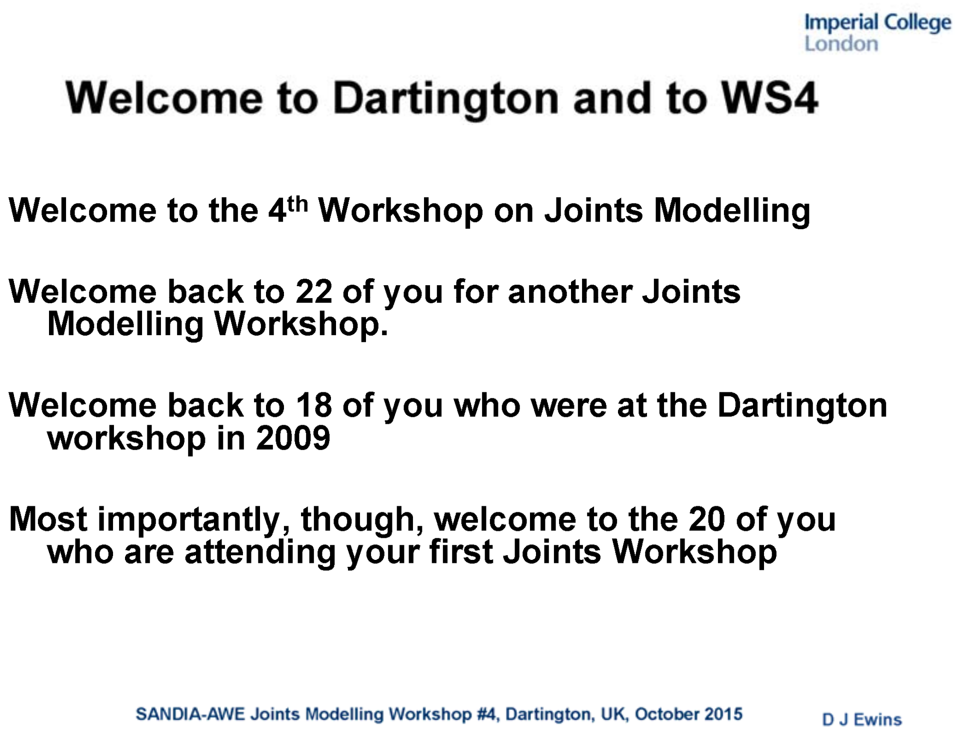
5. PRESENTATIONS

5.1. Workshop Introduction and Opening Perspectives

D. J. Ewins: Introduction to the 4th Workshop on Joints Modelling, 2015



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Introduction to the 4th Workshop on Joints Modelling, 2015

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

Imperial College London

Welcome to Dartington and to WS4

Let me begin by introducing the principal actors in staging WS4:

Larry Bergman and Dan Segalman were 2 of the founding fathers of this group

Dane Quinn and Matt Brake have become the other 2 leading players in the organisation of the ASME Research Committee which has become our ‘home’

Phil Ind and Randy Mayes represent the sponsoring organisations who have funded the entire workshop

From the UK, David Nowell, Hugh Goyder, Norbert Hoffmann and Christoph Schwingshackl have helped with local organisation

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

Imperial College London

PURPOSE

A reminder of why we are all here …. and what our task is…

This workshop is part of an ongoing initiative which seeks to develop a capability to understand, predict and then control the influence that joints have on the dynamics of critical structures.

Our task here is not only to review progress and new ideas in joints modelling technology, but also to explore the application of these methods to real engineering structures in order to gain technical and economic benefits.

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

Imperial College London

This 4th Workshop is fully sponsored by Sandia National Labs, NSF and AWE with the broad remit 0f improving our ability to understand, predict and then to control the dynamics of structures which contain joints

Our 3 days here will be made up of a combination of:

- **Reviewing progress since the last workshop in 2012**
- **Short talks by established experts in key areas,**
- **Parallel breakout groups to extract promising developments**
- **Short talks and posters reporting current research in key topics**

and will lead to the preparation of:

- **A review report of the current state of the art and the prospects for practical applications based on an improved joints modelling capability**
- **A 'road map' presentation of the key challenges and methods for addressing them**

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

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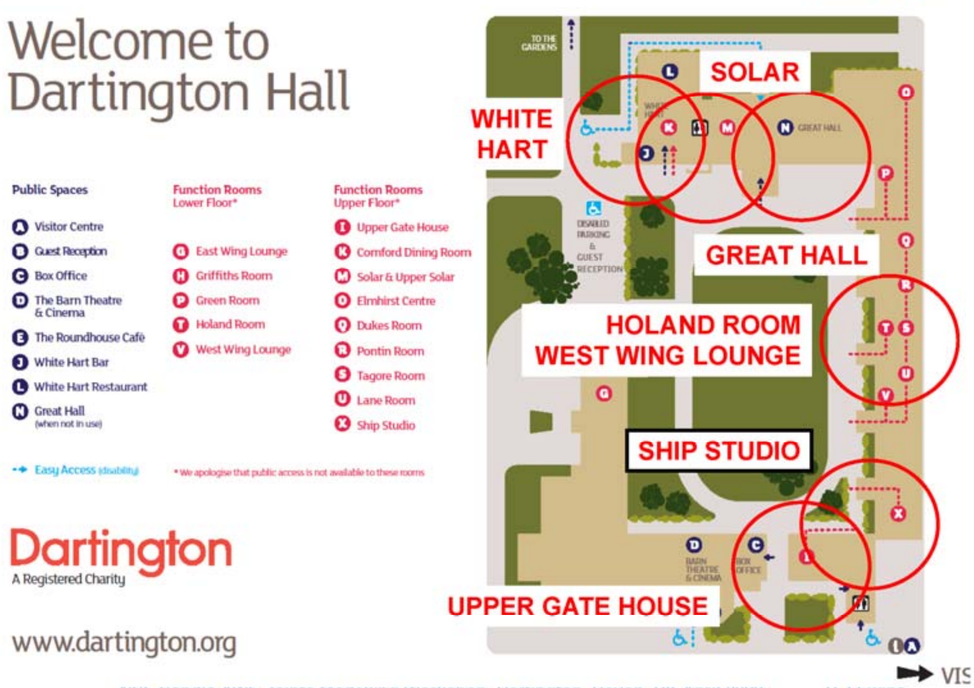
London

Welcome to Dartington and to WS4

And we shall do all this in these magnificent surroundings of Dartington Hall …

- **Main meeting room - Upper Gatehouse**
- **Breakout Rooms – Upper Gatehouse, Ship Studio, Solar**
- **Coffee Breaks and Posters - West Wing Lounge and Holand Room**
- **Dinner - Great Hall (Sunday and Tuesday), Solar (Tuesday)**
- **Lunch – Solar**
- **Breakfast – White Hart**
- **Bar – White Hart**

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins



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

Schedule for Day 1

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Coffee Breaks at appropriate times ***West Wing Lounge & Holand Room* (Posters*)***

0900 Workshop Day 1 ***Upper Gate House***

Chair Ewins

- Introduction, Objectives and Structure of Workshop

Chair Brake

- Review of progress on previous Actions & Challenges

1230 Lunch ***Solar***

1345 1st Themed Session: **APPLICABILITY**

Chairs Segalman, Ind

- Introduction (Invited Short Talks) *Upper Gate House*
- *(Speakers - **Green; Mayes; Ind; Tiedemann**)*
- Breakout (3 parallel sessions) *Upper Gate House; Ship Studio; Upper Solar*
- Summaries and Compilation of Collected Objectives

1730 Submitted Short Talks and Posters ***Upper Gate House***

Chair Goyder

1945 Dinner ***Solar***

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

Structure of the Workshop

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The Workshop is built around 4 Themed Sessions – with the participants split into 3 parallel groups each addressing the same issues.

Each themed session has three stages:

Short Talks by invited established experts in the relevant theme

Breakout into 3 groups to address the high level question posed for that theme

Compilation of an agreed set of common issues to be proposed as the response to the question

These lists will represent the main part of the outcome of the workshop and will form the basis of the final reports, covering both the key areas for further developments of the analytical and experimental technologies AND the prospects for specific applications in industrial application.

D J Ewins

Imperial College
London

Inputs and Outputs

You should have a folder with 6 items in it –

- Schedule
- List of Delegates
- Biographical notes
- List of the Short Talks and Posters, both Invited and Contributed
- Proposed Roles of all participants
- Some Thoughts on THE QUESTIONS

The essential function of the workshop is to address The Questions and to produce – in whatever format seems most appropriate – a critical assessment of the state of the art of the subject of Joints Modelling for structural dynamic analysis, and our specific proposals for a way forward in both developing and applying these methods in practice.

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

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Structure of the Workshop

The first session prepares the ground for the rest of the workshop. It presents the aspirations and the objectives of the joints modelling community, and progress since the last workshop in 2012

0900 Workshop Day 1 ***Upper Gate House***

Chair Ewins

- Introduction, Objectives and Structure of Workshop

Chair Brake

- Review of progress on previous Actions & Challenges

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

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Background

Joints have long been a problem for the structural dynamicist and, increasingly, the joints are becoming the weakest link in many design analyses. They are widely regarded as being 'unrepeatable' and 'unpredictable'

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 15 years…

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

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Previous Activities – 1

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

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

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

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

2006 Workshop on Joint Mechanics
Sponsored by NSF, SNL

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

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Previous Activities – 2

2009 Workshop 2 on Joint Mechanics - Dartington UK
Sponsored by AWE, NSF, SNL

2010 Formation of ASME Research Committee on Joint Mechanics

2012 Workshop 3 on Joint Mechanics - Chicago
Sponsored by SNL

2013 Update meeting and Sessions at IDETC, Portland
2014 Update meeting and Sessions at IDETC, Buffalo,
2015 Update meeting and Sessions at IDETC, Boston

2015 Workshop 4 on Joint Mechanics
Sponsored by AWE, NSF, SNL

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The Outcomes of Previous Workshops

We started with…..

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

as at 30/6/06

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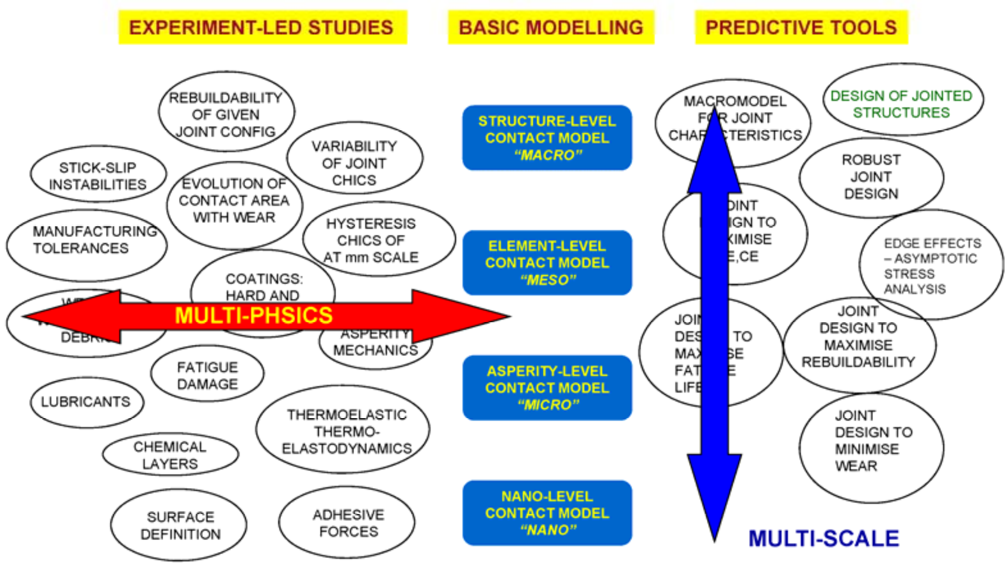
RESEARCH ROADMAP FOR FRICTION CONTACT AND WEAR IN STRUCTURES



Imperial College London

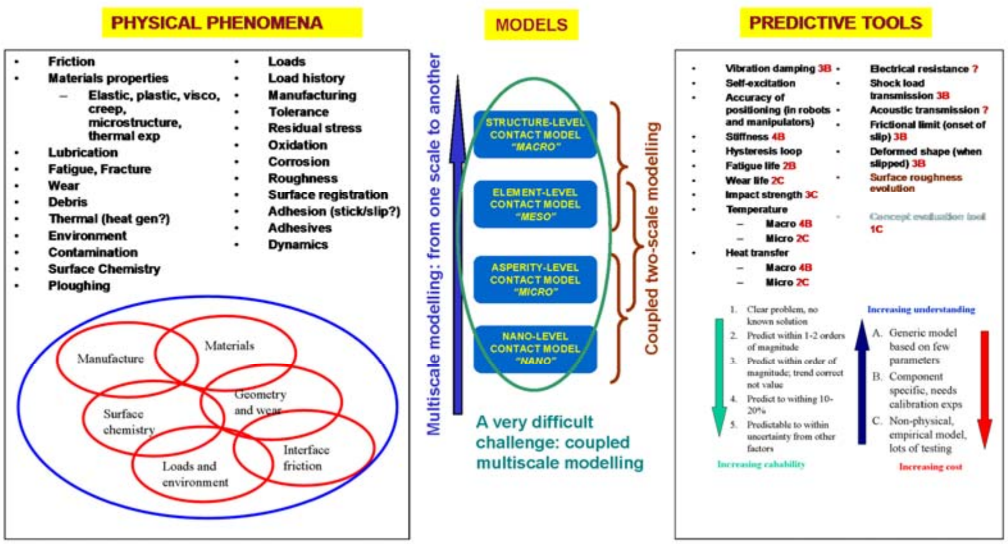
and ended up with…..

SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

as at 09/2006

Imperial College London

RESEARCH ROADMAP FOR FRICTION CONTACT AND WEAR IN STRUCTURES



as at 30/4/09

Imperial College London

Actions and Challenges, 2009

During the previous Dartington Workshop, we switched from the Road Map format for our output, and constructed a set of 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 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.

as at 30/4/09

Imperial College London

JOINTS MODELLING WORKSHOP #2

SUMMARY OF FINAL SESSION DISCUSSIONS ON 29/4/09

AGREED ACTIONS/CHALLENGES, INTERESTED PARTIES AND SUGGESTED LEADERS (in bold)

ACTIONS

1. Terminology & Vocabulary (***Segalman***; *Bergman)*
2. Develop Hills Chart *(**Dini**; Berger)*
3. Classification of Standard Joint Types *(Hills; Vakakis; **Starr**)*
4. Classification/Cataloguing of
   4.1 Non Linearity ID Methods (***Vakakis***)
   4.2 Modelling approaches (***Polycarpou***; *Quinn)*
   4.3 Measurement methods *(**Nowell**; Bergman; Akay)*
5. Benchmark current computation multiscale methods against analytic solutions *(**Masud**; Laurson; Quinn)*
6. Create a formal Joints Modelling Network (or Community) with more frequent and regular contacts (!); 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.

   University group on basic joints contact
   i. mechanics science

   University/industry group on measurements of
   ii. friction properties required for industrial
   iii. applications

   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 (meast-meast) and Variability (unit – 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 MultiScale 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

DJE 6/5/09

as at 11/2010

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A International Research Community
MECHANICS OF JOINTED STRUCTURES
Based on an ASME Research Committee

The Research Committee on Mechanics of Jointed Structures, established in 2010, investigates a broad spectrum of issues associated with the theoretical, experimental, and computational aspects of mechanics of joints and the mechanics of jointed structures. Its activities include the generation of new knowledge and development of guidelines for use by engineers and scientists in measurement, analysis, prediction, and design of mechanical joints and jointed structures

as at 30/8/13

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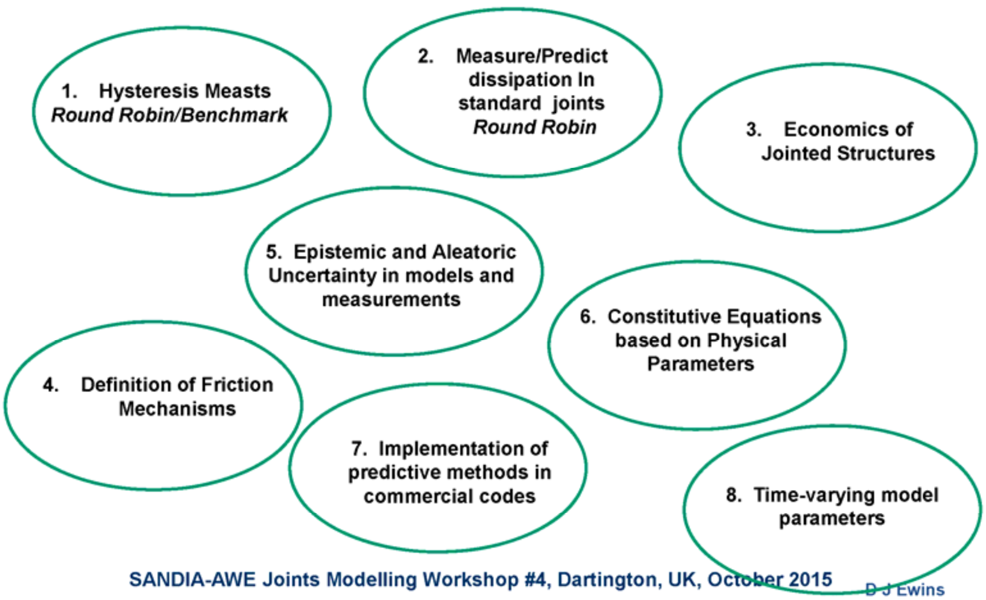
Challenges and Activities in Joints Modelling for Dynamics as at August 2013



SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015 D J Ewins

as at 30/8/14

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Current State of Challenges

Challenge	Participants
1. Round Robin – Hysteresis Measurements	David Ewins, David Nowell, Muzio Gola, Christoph Schwingshackl
2. Round Robin – Measurement and Prediction in Standard Hardware	Hugh Goyder, Matt Allen, Lothar Gaul, Norbert Hoffman, Kai Willner, Christoph Schwingshackl, Laura Jacobs, and Gael Chevallier
3. Economics of Jointed Structures	Matt Brake, David Ewins, Hugh Goyder, Pascal Reuss, Christoph Schwingshackl, and Matt Allen
4. Defining Mechanisms of Friction	David Nowell, Melih Eriten, Matthew Brake, George Ostermeyer, and Somuri Prasad
5. Epistemic and Aleatoric Uncertainty	Marc Mignolet, Dan Segalman, Kai Wilner, Matthew Brake, Mike Starr, Alex Vakakis, Lothar Gaul, and Larry Bergman
6. Derivation of Constitutive Equations	Lothar Gaul, Randy Mays, Norbert Hoffman, and Mike Starr
7. Numerical Implementation	Matthew Brake, Melih Eriten, Dan Brown, Hugh Goyder, and George Ostermeyer
8. Surface Chemistry Issues	Daniele Dini?, Simon Medina?, Christoph Schwingshackl?, and Melih Eriten?

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This Workshop

We need to re-group and move ahead…..
and for this will focus on 4 major themes:

1 APPLICABILITY
2 REPEATABILITY
3 PREDICTABILITY
4 WAY FORWARD

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THEME 1: APPLICABILITY

WHAT ARE INDUSTRIAL NEEDS AND POTENTIAL BENEFITS – TECHNICAL AND ECONOMIC - FROM A NEW JOINTS MODELLING CAPABILITY?

THEME 2: REPEATABILITY

WHAT ARE THE IMPLICATIONS OF MOST EXPERIMENTAL STUDIES REVEALING SIGNIFICANT UNREPEATABILITY IN JOINT DYNAMICS BEHAVIOUR?

THEME 3: PREDICTABILITY

TO WHAT EXTENT ARE CURRENT MODELING METHODS CAPABLE/ADEQUATE FOR FUTURE NEEDS?

THEME 4: APPLICATIONS AND WAY FORWARD

WHAT ARE THE PRIORITISED APPLICATION AREAS AND CAPABILITIES NECESSARY FOR TECHNICAL AND ECONOMIC SUCCESS?

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

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This Workshop

We need to re-group and move ahead…..
and for this will focus on 4 major themes:

1 APPLICABILITY
2 REPEATABILITY
3 PREDICTABILITY
4 WAY FORWARD

and propose taking a strategic approach…..

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What is a Strategy?

A Strategy has 4 parts:

1. An objective to be achieved (**O**)
2. A clear statement of the current position (**X**)
3. A comprehensive list of possible ways ahead ($\mathbf{m_i}$)
4. A specific plan of action to achieve #1 ($\mathbf{P_j}$)

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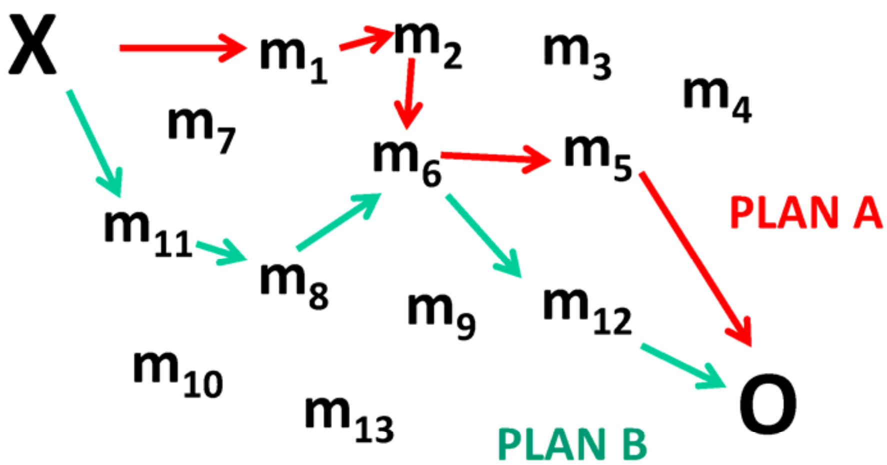
A STRATEGY



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The 4 themes of this workshop

1 APPLICABILITY
2 REPEATABILITY
3 PREDICTABILITY
4 WAY FORWARD

can fit into this structure as follows….

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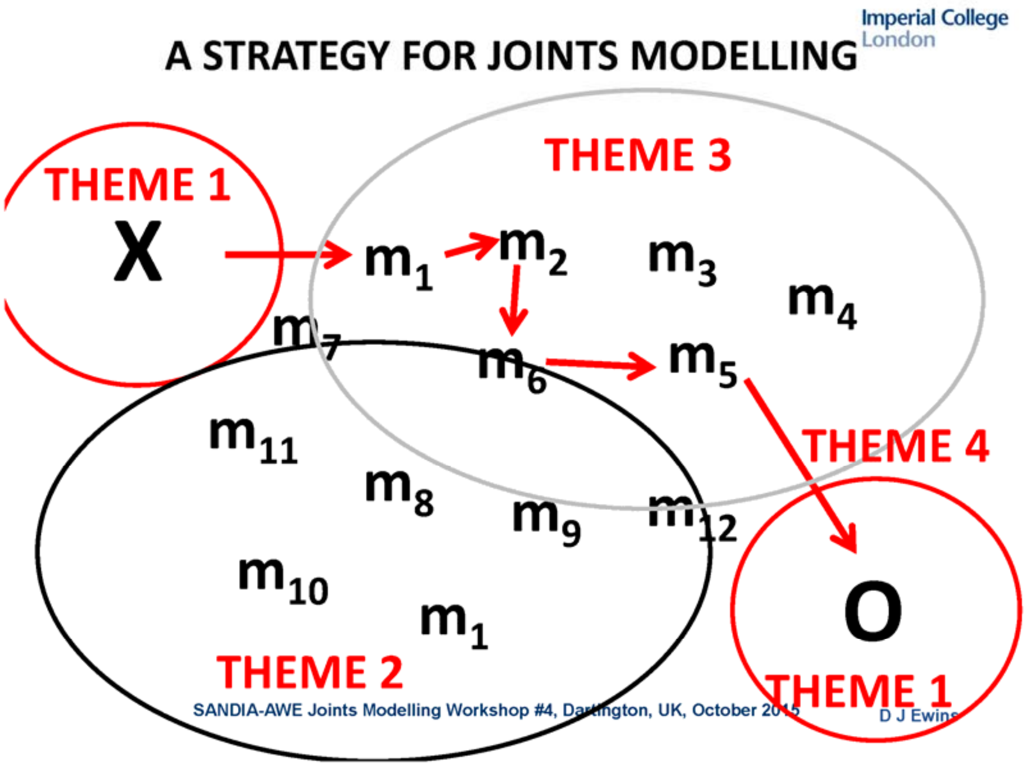
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London
A STRATEGY FOR JOINTS MODELLING
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X
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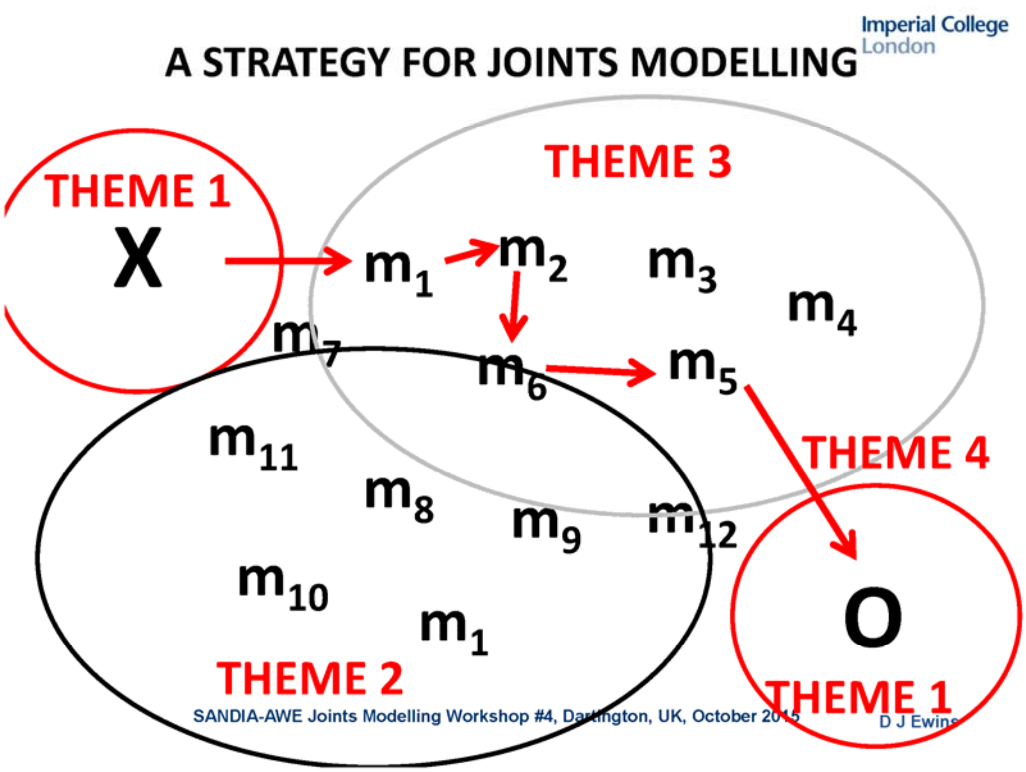
THEME 1
SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015
D J Ewins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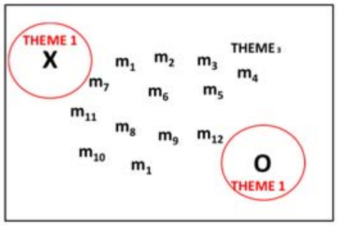
A STRATEGY FOR JOINTS MODELLING



THEME 1: APPLICABILITY

WHAT ARE INDUSTRIAL NEEDS AND POTENTIAL BENEFITS – TECHNICAL AND ECONOMIC - FROM A NEW JOINTS MODELLING CAPABILITY?

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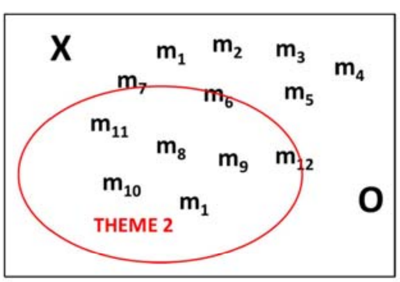
A STRATEGY FOR JOINTS MODELLING

THEME 2: REPEATABILITY

WHAT ARE THE IMPLICATIONS OF MOST EXPERIMENTAL STUDIES REVEALING SIGNIFICANT UNREPEATABILITY IN JOINT DYNAMICS BEHAVIOUR?



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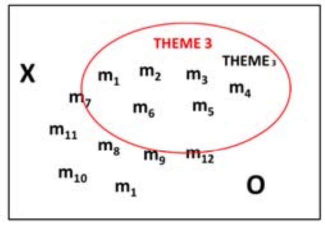

THEME 3: PREDICTABILITY

TO WHAT EXTENT ARE CURRENT MODELING METHODS CAPABLE/ADEQUATE FOR FUTURE NEEDS?

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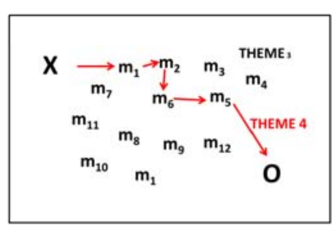
THEME 4: APPLICATIONS AND WAY FORWARD

WHAT ARE THE PRIORITISED APPLICATION AREAS AND CAPABILITIES NECESSARY FOR TECHNICAL AND ECONOMIC SUCCESS?



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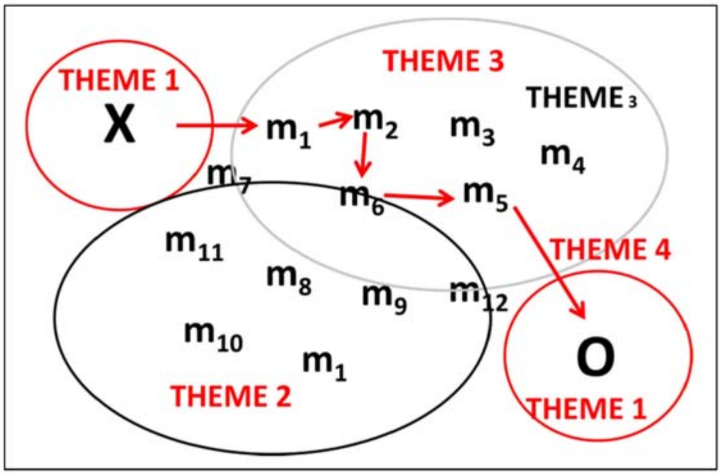

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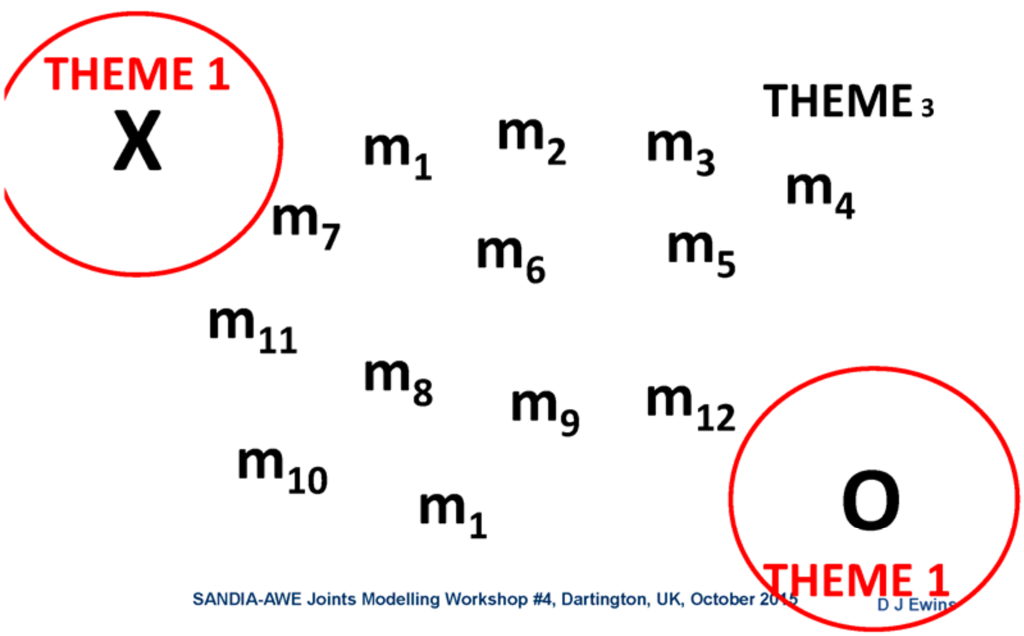

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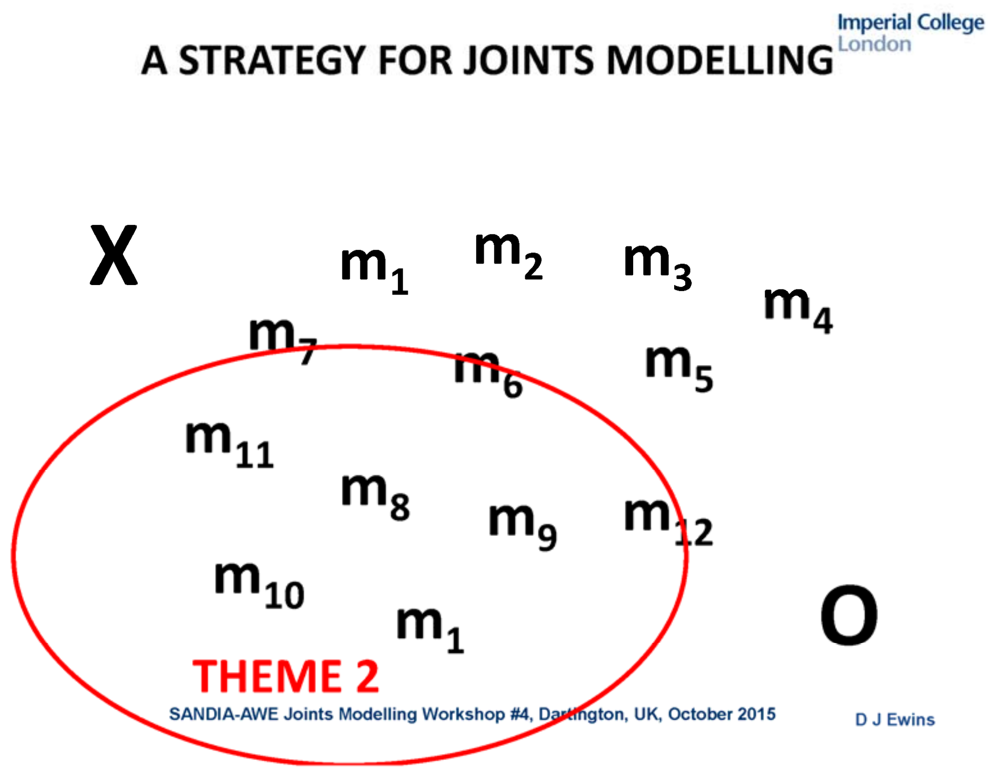
A STRATEGY FOR JOINTS MODELLING
Imperial College London
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THEME 2
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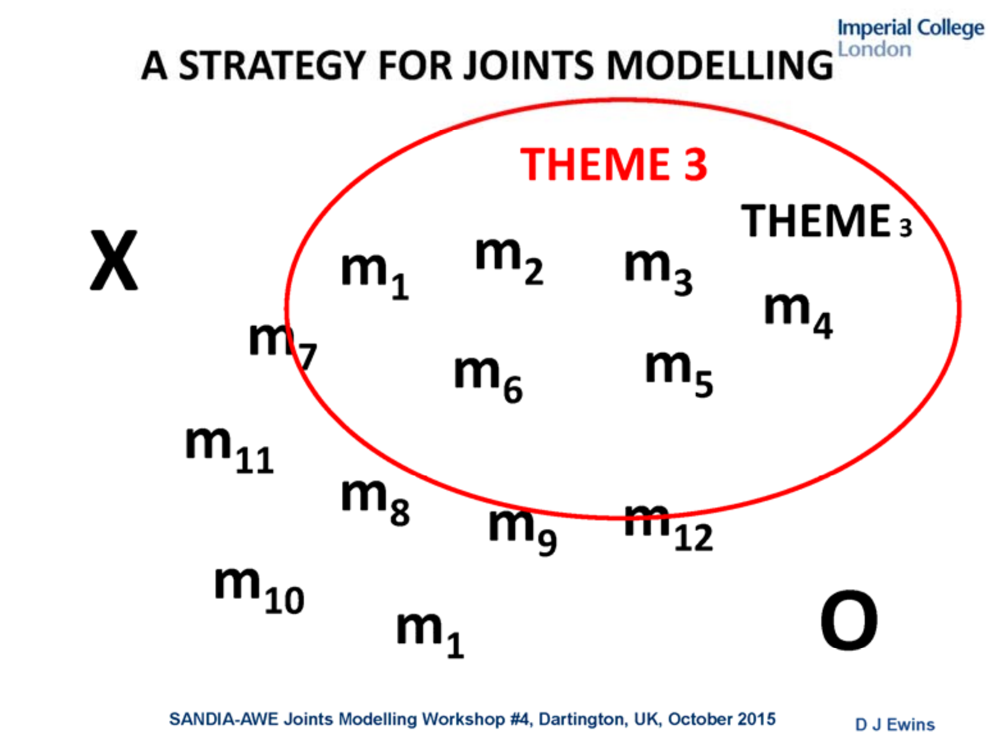
A STRATEGY FOR JOINTS MODELLING
Imperial College London
THEME 3
THEME 3
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SANDIA-AWE Joints Modelling Workshop #4, Dartington, UK, October 2015
D J Ewins

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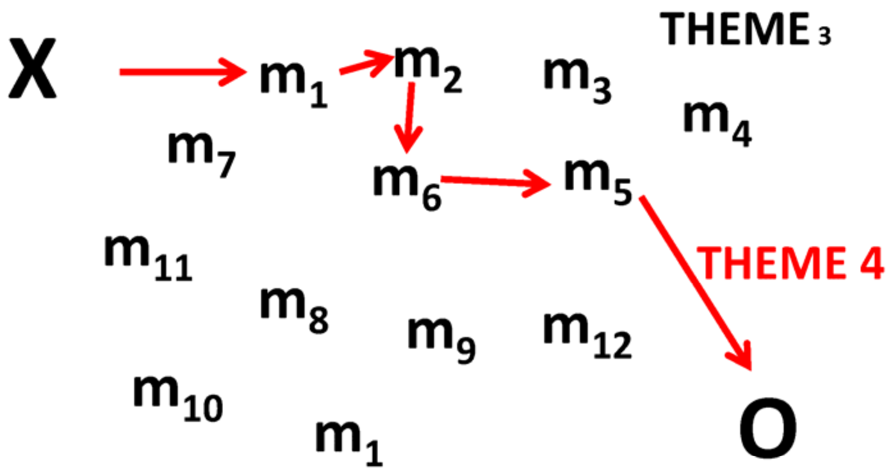
A STRATEGY FOR JOINTS MODELLING



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The Outcome of This Workshop

A New Road Map?

New list of Actions & Challenges?

…….?

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Structure of the Workshop

.

Before embarking on these activities, the 4 officers of the Research Committee – some of whom are near retirement – will give a brief of their perspective and aspirations for this WS4

0900 Workshop Day 1 ***Upper Gate House***

Chair Ewins

- Introduction, Objectives and Structure of Workshop

Chair Brake

- Review of progress on previous Actions & Challenges

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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 and current 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.

D J Ewins

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Welcome to Dartington and to WS4

We shall do this firstly by reporting on progress since we last met in 2012, and then by conducting systematic reviews of the crucial elements in a joints modelling capability.

These review comprise short talks by established specialists in the various key areas, followed by parallel breakout groups listing key issues and developments and finally by combining these separate efforts into an agreed prioritised list.

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Welcome to Dartington and to WS4

We shall use similar tools to carry out the task as have been used in previous workshops, although in a slightly different context.

The workshop is centred around 4 half-day Themed Sessions, each of which comprises:

(i) 3 or 4 short invited talks to provide some deep insight and background to the theme in question, followed by

(ii) 3 parallel breakout groups all with the same remit of compiling a list of issues, or topics, that relate to the Theme,

(iii) Followed by a reassembly of all delegates to compile an agreed and prioritised common evaluation of the current theme

In addition, there will be 'optional' short talks in the early evening between the afternoon session and dinner.

Finally, there will be the important task of assigning leadership and participation in capturing the output of these discussions and planning the future developments to progress the subject.

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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?

D J Ewins

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Structure of the Workshop

The Workshop is built around 4 Themed Sessions – with the participants split into 3 parallel groups each addressing the same issues.

1345 1st Themed Session: **APPLICABILITY**

Chairs Segalman, Ind

- Introduction (Invited Short Talks) ***Upper Gate House***
  *(Speakers - **Green; Mayes; Ind; Tiedemann**)*
- Breakout (3 parallel sessions) ***Upper Gate House; Ship Studio; Upper Solar***
- Summaries and Compilation of Collected Objectives

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

D J Ewins

Larry Bergman: Introductory Remarks

Introductory Remarks

Larry Bergman
University of Illinois
lbergman@illinois.edu

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1

Introductory Remarks

- Problems of dynamics of complex, jointed structures continue to perplex the analyst and experimentalist despite the best efforts of this group and others.
- The many uncertainties associated with joints and jointed structures are, no doubt, responsible.
- Many sources of uncertainty exist, which we try to categorize as either epistemic or aleatory.
- Briefly, epistemic uncertainties are those that can be reduced by refining models and/or gathering more data, while aleatoric uncertainties are those that are viewed as irreducible.
- The latter are best modeled in a probabilistic framework; the former not necessarily so.

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\

Introductory Remarks

- Identify major epistemic sources of uncertainty and try to focus R&D in those directions for the immediate future.
- Will likely require significantly more experimental effort over a wide range of scales to resolve.
- Direction from this community should drive this activity.

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3

4^{th} Workshop on Joints Modeling

D. Dane Quinn

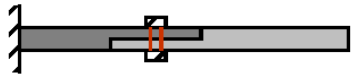
Department of Mechanical Engineering
The University of Akron
Akron, OH 44325–3903 USA

Dartington Hall
October 2015

D. D. Quinn (U. Akron) | Joints Modeling | Joints Modeling 1 / 4

What Do We Hope To Obtain?

Structural systems are often composed of multiple components joined together at localized interfaces



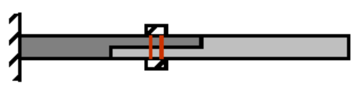
- added mass due to the joint
- little influence on the load carrying capability

Why are we here?

- Joints play a significant role in the dynamics of structures
- Better modeling and prediction can allow for better design of jointed structures

What Do We Hope To Obtain?

Structural systems are often composed of multiple components joined together at localized interfaces



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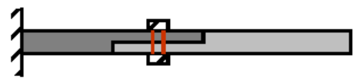

- added mass due to the joint
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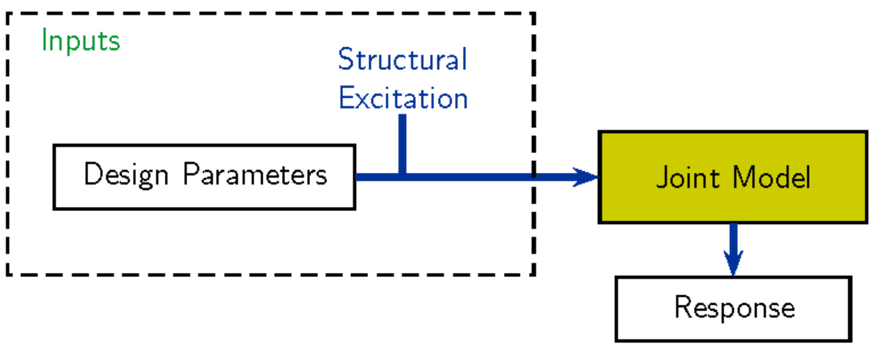
Why are we here?

- Joints play a significant role in the dynamics of structures
- Better modeling and prediction can allow for better design of jointed structures

What is the Problem?

Jointed structures suffer from issues of

- Repeatability
- Predictability



This assumes that the remainder of the structure can be well-modeled **independent of the joint**

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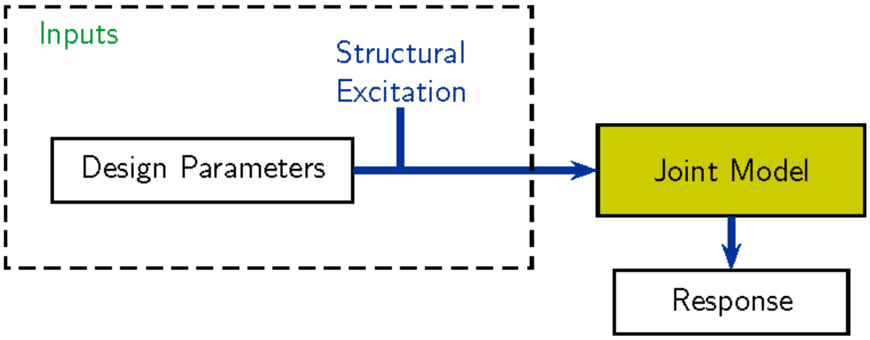


This assumes that the remainder of the structure can be well-modeled **independent of the joint**

What is the Answer?

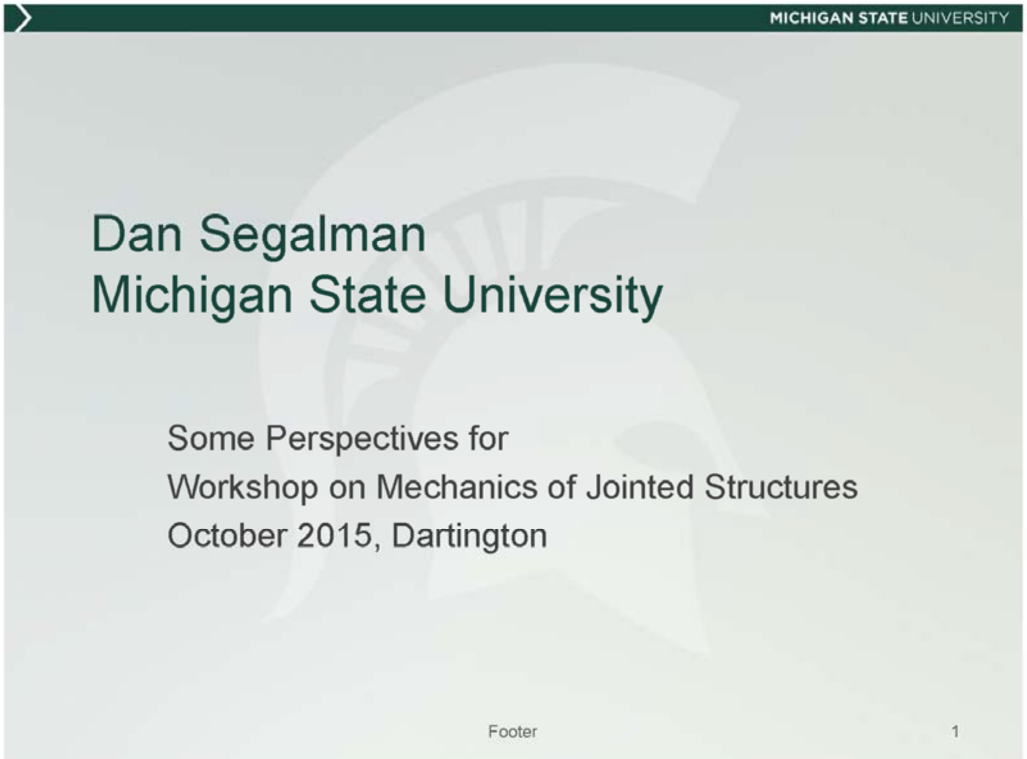
If I knew, we would not be here...

Desired Qualities

- Predictive
- Repeatable
- Computationally Efficient
- Easily Incorporated into Existing Approaches

Go!

Dan Segalman: Some Perspectives for the Workshop on Mechanics of Jointed Structures



Where do we stand on these issues?

- Applicability
- Repeatability of Joints
- Ability to Model

In each case it depends on context.

Consider 0^{th} order design and analysis

Traditional Concerns in Joint Design

- Strength (tolerable loads)
- Fatigue resistance
- Thermal stability
- Cost
- .
- .

Traditional Design/Analysis Process

1. Design with tradition and low order models
2. FE models to analyze design
3. Test Prototypes at multiple stages
    1. Revise design as necessary
    2. Revise FE models as necessary
    3. Retest as necessary

Relies on long experience and large factors of safety

- Shigley-level analysis is sufficient for design and assessment
- Measurable variability among nominally identical joints is acceptable

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1^{st} order analysis and design of joints (Lower factors of safety and fewer tests)

More Current Design Process: more integration of FE analysis

1. Design subsystems with FE and 0^{th} order physics
2. FE models used to analyze design
3. Design iteration via FE rather than frequent test
4. Occasional subsystem tests
5. Admirals test

Sophistication

- Demands on underlying physics are slightly higher – there are fewer subsystem tests.
- Joint models still fairly simple. High fidelity ability to predict role of joint in system dynamics not required.
- Most applicable where individual joint predictability and uniformity are not critical.
- This usually means dealing with only perturbations on old designs.

Footer 4

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Next order analysis and design of jointed structures

- Dynamics of structure over very distinct amplitude regimes is important.
- Designs of very different character are contemplated. One cannot count on the new design behaving similarly to old ones.
- Very expensive or critical systems are contemplated.
- Very fast analysis turn around is expected.
- There is very little opportunity for intermediate testing.

Footer 5

Levels and sophistication of joint models

- Can parameters be found to cause the model to fit the data for a given system?
- Does the model with those parameters predict other experiments?
- Can we pull parameters for other geometries, materials, load regimes from some dimensionless master charts?
- Can these useful models and their parameters be derived quantitatively from some underlying physics?

Sophistication

A question about repeatability/variability

- Granted that among nominally identical joints stiffness may vary by 50% and dissipation by 300%
- What does this mean for response of structures where loads go through MANY joints?
- Can we expect system to system variability to be substantially less than joint to joint variability?

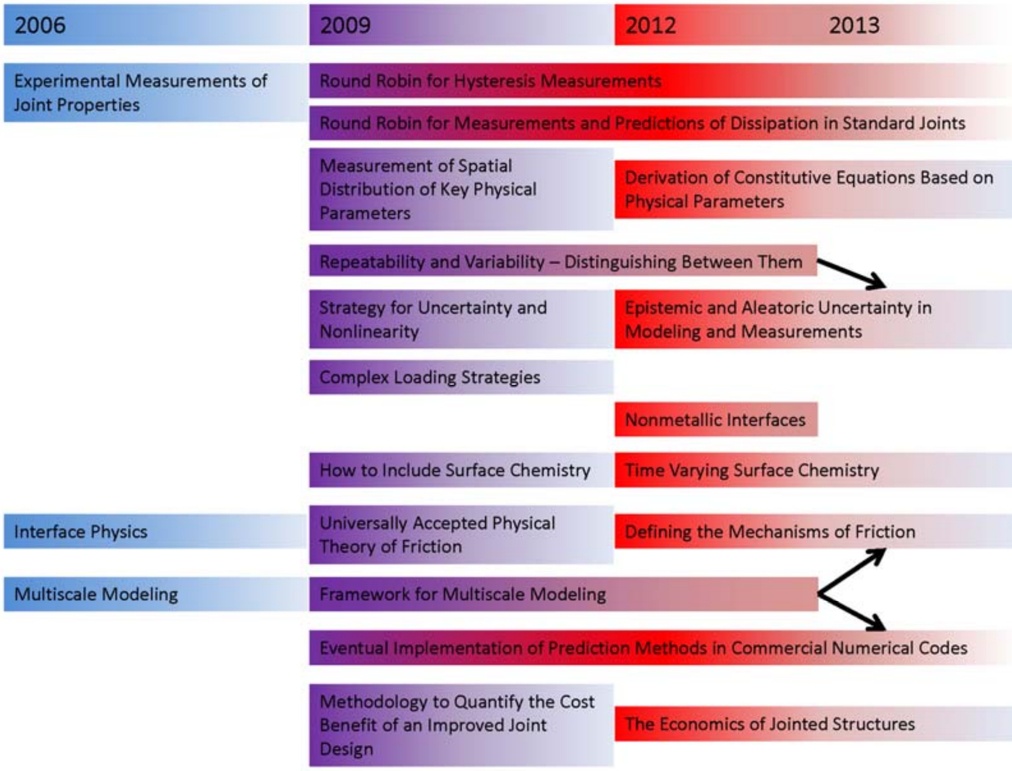
Matthew Brake: Actions and Challenges from the 3rd Joints Workshop: Progress as of October 2015

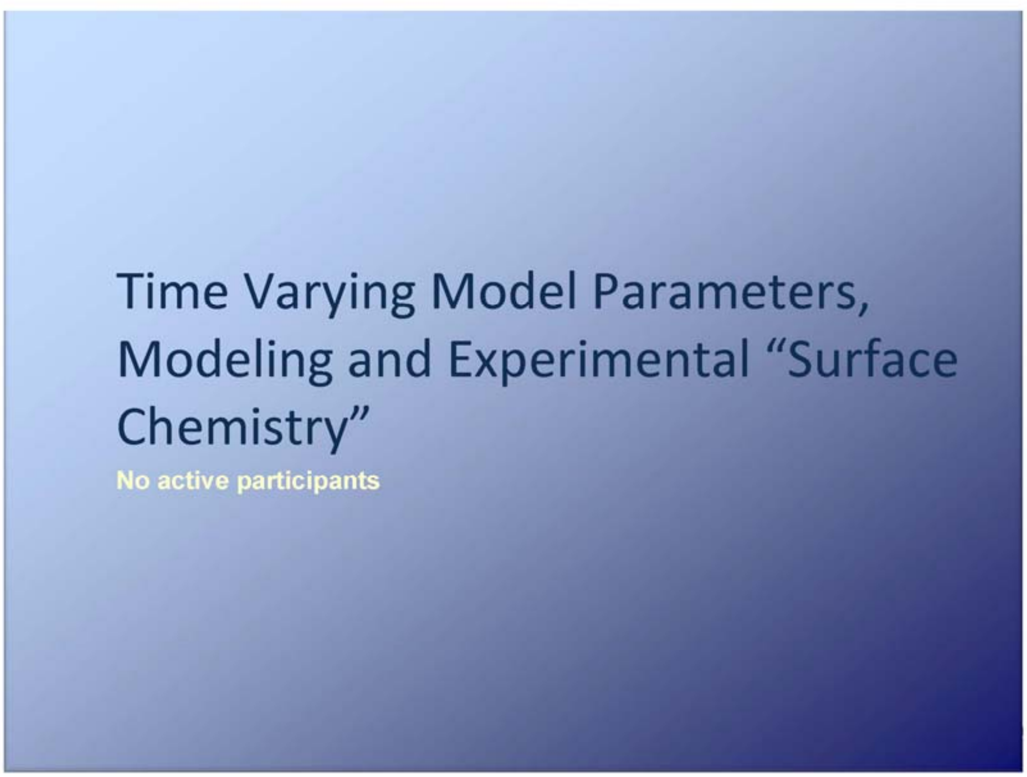
Actions After the Third Workshop

- Website – Pablo Tarazaga
  - http://asmercmjs.weebly.com
- Documentation – Starr and Brake
  - *SAND2013-6655*
- Grow committee and add new blood
  - 50 committee "members" in 2012, 140 today
- Investigate multinational collaborations for funding
  - David Ewins volunteered for this
  - Stuttgart/Imperial/Torino/others submitted one applications…
- Organize 4th Workshop
  - Done.
- Executive summary of the challenges for the website
  - Incomplete
- Updates at key conferences, not just IDETC
  - Expanded to IMAC; perhaps ISMA or TurboExpo in the future?

Challenges After the Third Workshop

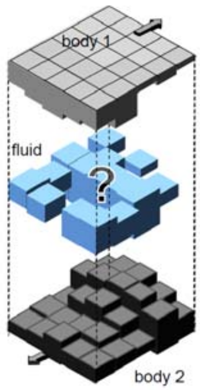
- Challenges redefined after the mini-workshop in Portland:
  1. Round Robin/Benchmark Exercise for Hysteresis Measurements
  2. Round Robin/Benchmark Exercise for Measurement and Prediction of Dissipation in Standard Joints
  3. The Economics of Jointed Structures
  4. Defining the Mechanisms of Friction
  5. Epistemic and Aleatoric Uncertainty in Modeling and Measurements
  6. Derivation of Constitutive Equations Based on Physical Parameters
  7. Eventual Implementation of Prediction Methods in Commercial Numerical Codes
  8. Time Varying Model Parameters, Modeling and Experimental "Surface Chemistry"

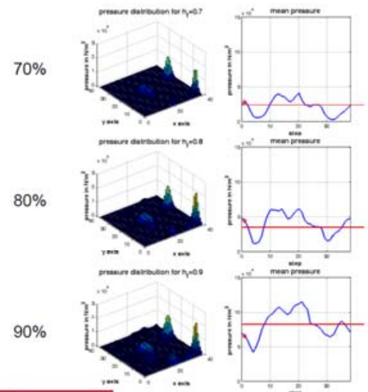

2006
2009
2012
2013
Experimental Measurements of Joint Properties
Round Robin for Hysteresis Measurements
Round Robin for Measurements and Predictions of Dissipation in Standard Joints
Measurement of Spatial Distribution of Key Physical Parameters
Derivation of Constitutive Equations Based on Physical Parameters
Repeatability and Variability – Distinguishing Between Them
Strategy for Uncertainty and Nonlinearity
Epistemic and Aleatoric Uncertainty in Modeling and Measurements
Complex Loading Strategies
Nonmetallic Interfaces
How to Include Surface Chemistry
Time Varying Surface Chemistry
Interface Physics
Universally Accepted Physical Theory of Friction
Defining the Mechanisms of Friction
Multiscale Modeling
Framework for Multiscale Modeling
Eventual Implementation of Prediction Methods in Commercial Numerical Codes
Methodology to Quantify the Cost Benefit of an Improved Joint Design
The Economics of Jointed Structures


Time Varying Model Parameters, Modeling and Experimental “Surface Chemistry”
No active participants

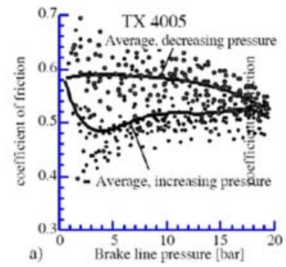
Progress Since 2012

- Time varying local friction models deduced from a cellular automata computational approach for dry friction extended to lubricated and partially lubricated conditions (Ostermeyer and Muller)





$$\dot{\mu} = \alpha\left(\mu - \mu_{stat}(N, v, T)\right)$$
$$\dot{T} = \beta\left(T - T_{stat}(N, v, \mu)\right)$$



Technische Universität Braunschweig

19.08.2014 | Müller, Butser, Ostermeyer | IDETC 2014 / VIB , Buffalo

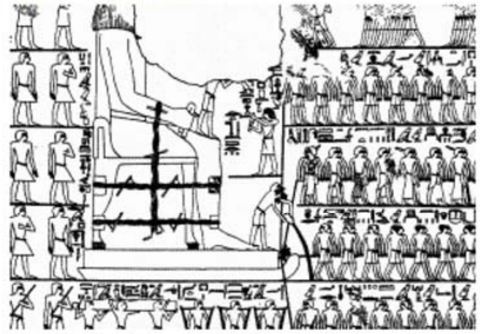
Defining the Mechanisms of Friction

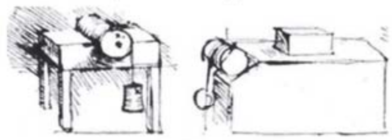
David Nowell (lead, Oxford), Matthew Brake and Somuri Prasad (Sandia), Melih Eriten (Wisconsin), and George Ostermeyer (Braunschweig)

Defining the Mechanisms of Friction

Centuries-long Quest

Ancient Egypt (~2700 BC)
Da Vinci's drawings (15th century)
Amontons (17th century)
Euler (18th century)
Coulomb (18-19th century)
Bowden & Tabor (20th century)





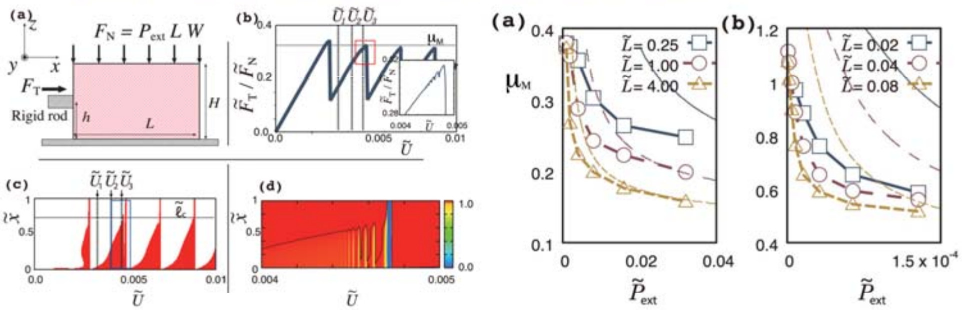
Classical Laws of Friction

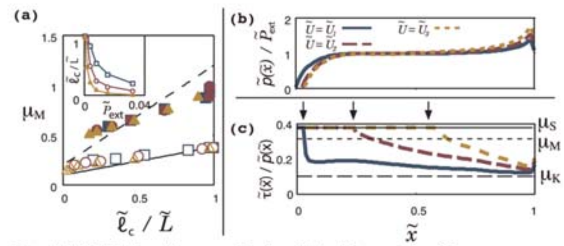
1. The area in contact has no effect on friction (apparently not, but in reality?)
2. If the load of an object is doubled, friction will also be doubled. (constant friction coefficient)
3. Friction is independent of sliding velocity (velocity weakening?)

Progress Since 2012

- Preliminary experimental design developed at Sandia
  - Combined tribology/structural dynamics approach
  - Lack of progress due to lack of funding
  - Alternative proposals have included in situ nano-/micro-scale testing

Friction Coefficient: Not So Constant





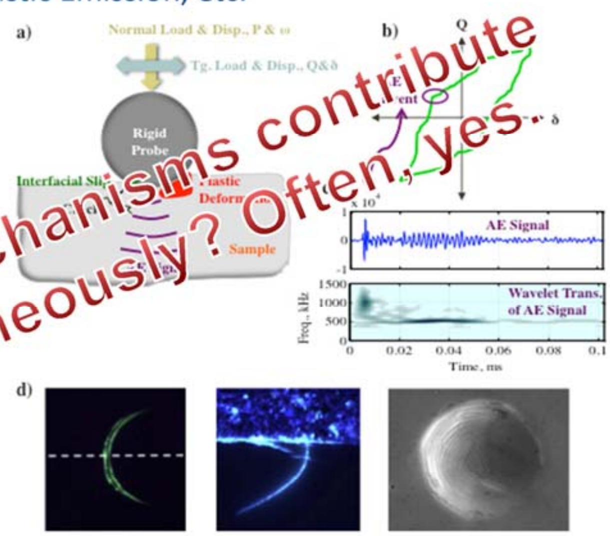
	Kinetic	Static
Rubber on concrete (dry)	0.68	0.90
Rubber on concrete (wet)	0.58	
Rubber on asphalt (dry)	0.67	0.85
Rubber on asphalt (wet)	0.53	
Rubber on ice	0.15	
Waxed ski on snow	0.05	0.14
Wood on wood	0.30	0.42
Steel on steel	0.57	0.74
Copper on steel	0.36	0.53
Teflon on Teflon	0.04	

Otsuki & Matsukawa, Scientific Reports, 3, 2013.

Where Does Energy go in Frictional Interfaces?

An Interplay of Elasticity, Plasticity, Fracture, Interfacial Slip, Adhesion, Impacts, Acoustic Emission, etc.

- Elasticity → bond properties, atomistic properties
- Plasticity → crystal orientation, microstructure, dislocations
- Fracture → surface energy, microstructure, bond strength
- Interfacial slip → elastic mismatch, interfacial bonds
- Adhesion → hysteretic separation/pull-off behavior
- Impacts → geometry, roughness
- AE → coupling with acoustic modes
- Thermal → heat generation



Do all mechanisms contribute simultaneously? Often, yes.

Matrix for New Generation of Ashby Maps

Input/Controls	Output/Measures	Capabilities To Be Developed
Materials: metals, single phase, … Plasticity index, elastic properties, melting/phase transition temperature	**Sliding Force**: principle measurement of friction tests	**Flash Temperature**: intermediate approach via thermocouples
Grain Orientation: grains aligned in-plane, orthogonal to the plane, …	**Topography Changes**: way to infer physical changes to asperities	**Plastic Strain Mapping**: capability needs to be rejuvenated at SNL
Grain Alignment: angle that the measurements are made with respect to the in-plane grain orientations	**Microstructure and Grain Size/Orientation**: electron diffraction and focused ion beam	**Vacuum Chamber**: for friction tests specifically
Surface Roughness: polishing…	**Acoustic Emission**: microphones	
Strain Rate: experimentally controlled		
Normal Force: experimentally controlled		
Temperature: method of determining heat generation when combined with melting/phase transitions		
Humidity: experimentally controlled; lubrication effect		
Atmosphere: inert gas versus reactivity with the samples		

Test Plan

1. Determine Materials
2. Baseline Test Definition
3. Standard Measurements
4. Single Variable Test Cases
5. Multivariate Test Cases

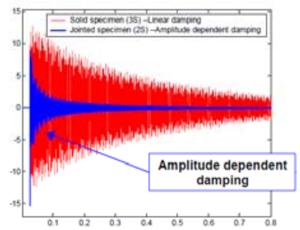
The Economics of Jointed Structures

Matthew Brake (lead, Sandia), David Ewins (lead, Imperial),
Hugh Goyder (Cranfield), Pascal Reuss (Stuttgart),
Christoph Schwingshackl (Imperial), and Matt Allen (Wisconsin)

What are The Economics of Designing Structures With and Without Joints?

Before we can answer, there are several things we need:

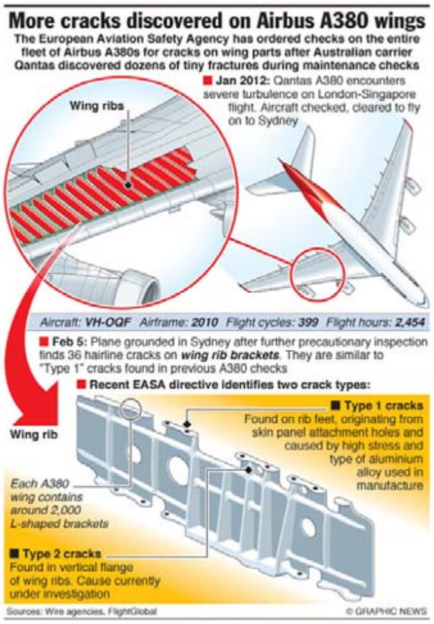
- Should we even have joints? Why not monolithic structures?
- Predictive models
  - What would these buy us?
- Motivation for why we care about joints
- Quantification of what joints will do for us
  - In terms of response
  - In terms of savings
  - Cost/Benefit analysis
    - Cost of Failure
    - Cost of Testing for Uncertainties
    - Benefit of Saving Weight
    - Benefit of Using Joints as Design Tools
    - Benefit of Using Joints to Monitor Structures



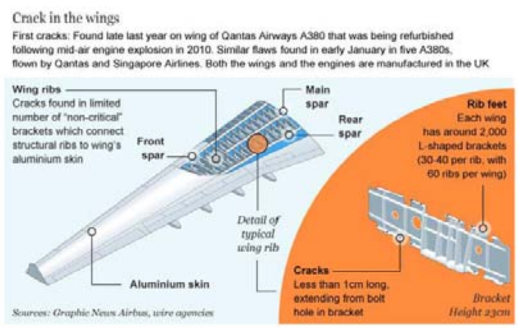
Progress Since 2012

- Two Cost/Benefit Analyses
  - Both preliminary
  - One at Sandia by M. Brake et al. looking at publicly available information
  - One at Rolls Royce by D. Ewins looking at a specific platform
  - Both analyses incomplete for now

Cost of Failure

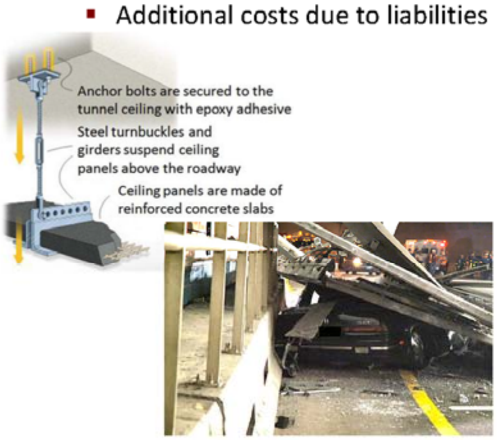


- An example: the Airbus 380
  - Each A380 costs $414 million to buy
  - Total development cost of $15.5 billion
  - Cracks found adjacent to joints
  - Fleet grounded for several months
  - $630 million cost to repair
  - $1.2 billion in lost profit for airlines
  - Additional costs for redesigning

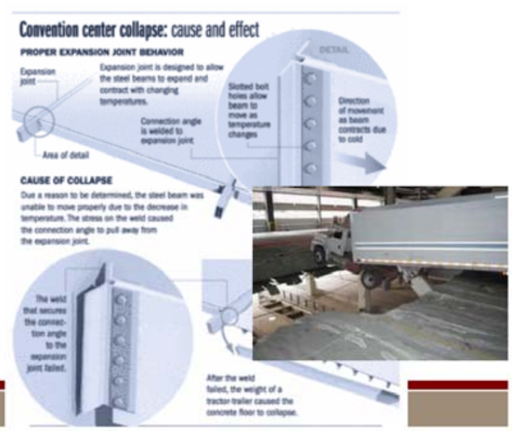


Cost of Failure

- Big Dig ceiling failure
  - Bolted connections holding ceiling panels to tunnel failed
  - $54 million cost to repair
  - Additional costs due to liabilities



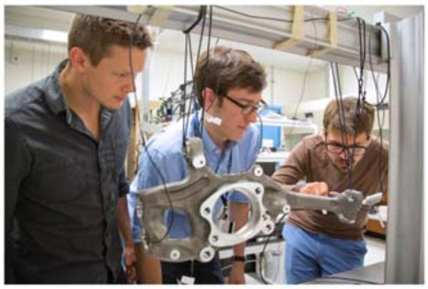
- Quick search of recent failures due to bolts:
  - I-35 Bridge, Minneapolis ($600 million)
  - Centergy Parking Deck, Atlanta
  - The San Antonio Parking Garage
  - Sayano-Shushnkaya Hydroelectric Power Station ($2.2 billion)
  - David L. Lawrence Convention Center, Pittsburgh

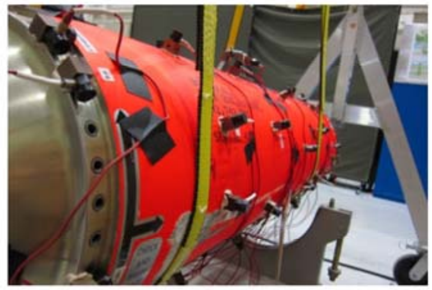


Source: failures.wikispaces.com/

Cost of Testing for Uncertainties

- Dynamic tests are often necessitated for qualification of structures and calibration of models for systems with joints
- In the US alone, on the order of $2.5 Billion[1] is spent on dynamic testing per year.

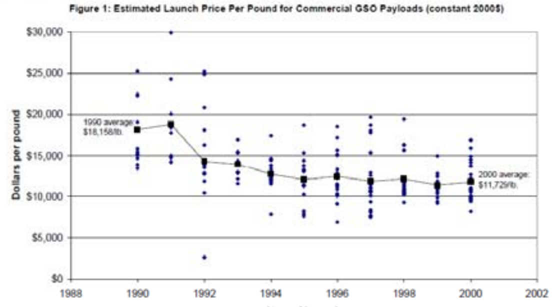




1: Elephant Tech Consulting, "Part 2: The Sound and Vibration Market for Investors – Summary"

Benefit of Saving Weight

Figure 1: Estimated Launch Price Per Pound for Commercial GSO Payloads (constant 2000$)



- Most savings is in fuel efficiency (automotive, aerospace, turbines, etc.)
- Example 1: To launch a payload into low orbit: $4,000/pound; for a geosynchronous orbit: $16,000/pound. ($12,000/pound on average)
  - Reducing weight of joints by X pounds in a satellite directly saves $12,000*X per launch.
- Example 2: Total mass of joints in a Rolls Royce turbine ~100s of kgs; if you can reduce the mass by 2%, that will save 10x the cost of development over a turbine's lifetime...

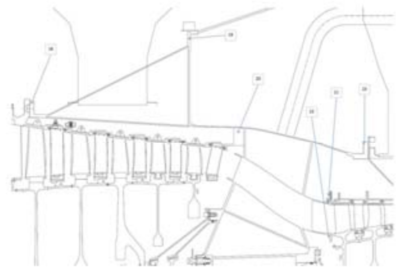
Sources:
www.worldbank.org/
www.futron.com/upload/wysiwyg/Resources/Whitepapers/Space_Transportation_Costs_Trends_0902.pdf

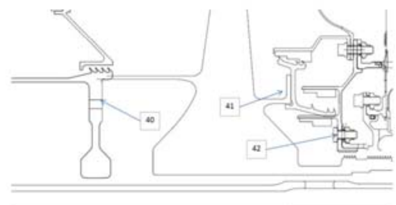
Aero Gas Turbines – Joint Investigation
20

- 91 Flange Joints
- Total Mass = 150kg
- 3796 nuts/bolts
- Total Mass = 40kg

- Engine Mass = 7400kg

- Flanges & Fixings 2.5% of Total Engine Mass



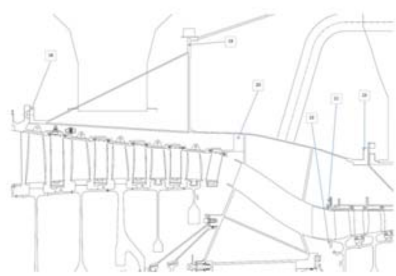


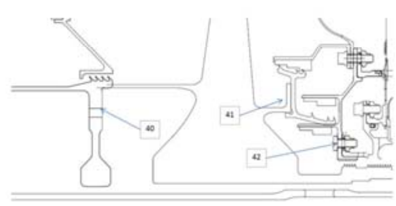
Trusted to deliver excellence

Rolls-Royce

Aero Gas Turbines – Joint Investigation
21

- **Benefits**
  - 20% saving in mass of flanges & fixings = 40kg
  - Benefit from cost reduction ~£100's pre kg per Engine
  - Benefits from reduced total cost of ownership ~£1000's per kg over the life of an Engine
  - Benefits are more significant if improvements can be built in at the design stage, with a high confidence of success.





Trusted to deliver excellence

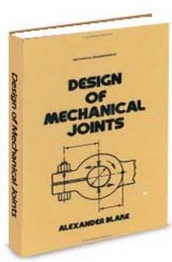
Rolls-Royce

Benefit of Using Joints for Structural Health Monitoring

- Key idea: structural health monitoring built into joints
- Opportunity to optimally plan a repair cycle for a structure
- Early warning sign to avoid structural failures
- Many potential applications have catastrophic consequences associated with failures
- Cost benefit expected to be deduced from insurance company estimates

I-35 in Minnesota, August 1st, 2007

Benefit of Joints as a Design Tool



- Ultimate goal: predictive model of joints
  - Pre-built model of joints with known performance
  - Handbook with easily understood metrics for how a specific joint performed
  - Ability to condition structural response by design of joints
- Impact on direct cost of design time, development cycle, product testing, and production
- If we had X capability from a better knowledge of joints, could we cut out a step in the design cycle?
- Notion of having a repeatable joints design for analysts to use

Epistemic and Aleatoric Uncertainty in Modeling and Measurements

Marc Mignolet (lead, ASU), Dan Segalman (lead, Michigan State), Matthew Brake and Mike Starr (Sandia), Kai Willner (Erlanger/Imperial), Alex Vakakis (Illinois), Lothar Gaul (Stuttgart), and Larry Bergman (Illinois)

Usually Uncertainty is Categorized into Two Sorts

- Aleatoric Uncertainty: uncertainty due to intrinsic variability. This includes parametric uncertainty.
- There is a lot of this in mechanical joints!

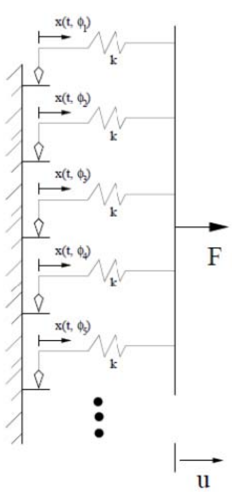
- Epistemic Uncertainty: uncertainty which is due to things we could in principle know but don't in practice. This includes model form error.
- This includes things that we are unlikely ever to know in practice.

Progress Since 2012

- Considered extensively the Sandia data and only that data (so far!) with 4 and 5 parameter Iwan models
  - Efforts made at NOMAD 2015 to create new data sets for standard joints for future UQ analysis
- 5 parameter Iwan model provides a close fit of all 9 samples, thus only aleatoric uncertainty introduced. A modeling based on max entropy distribution was proposed and seem to give appropriate 5th-95th percentile band of uncertainty – includes the data as expected.
- Identification of the parameters of the distribution (hyperparameters) achieved by approximate maximum likelihood
  - See work based on Soize's method
- Modeling using the 4 parameter Iwan would require significant epistemic uncertainty. Several options to introduce this uncertainty were proposed based on treating the distribution of the distribution f as random. Preliminary validation of these options was carried out but no final modeling adopted because of the 5 parameter model discovery.
- The possibility of using the 5 parameter model to generate more data to serve for the development of an epistemic uncertainty modeling for the 4 parameter model has been thought of but not worked on!
- Improved cultural awareness of what validation actually is

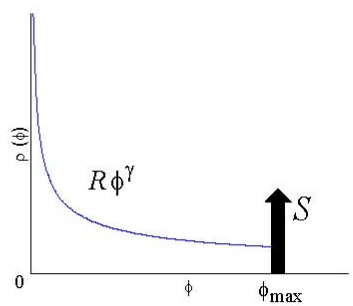
Deterministic 5-Parameter Iwan Model

Proposed Mean (Deterministic) Model: 5-parameter Iwan model



$$F(t) = \theta \int_0^{\phi} \rho(\phi)\left[u(t) - x(t,\phi)\right] d\phi + \int_{\phi}^{\infty} \rho(\phi)\left[u(t) - x(t,\phi)\right] d\phi$$

$$\dot{x}(t,\phi) = \dot{u}(t) \text{ if } \left\| u(t) - x(t,\phi) \right\| = \phi \text{ and } \dot{u}\left(u(t) - x(t,\phi)\right) > 0$$



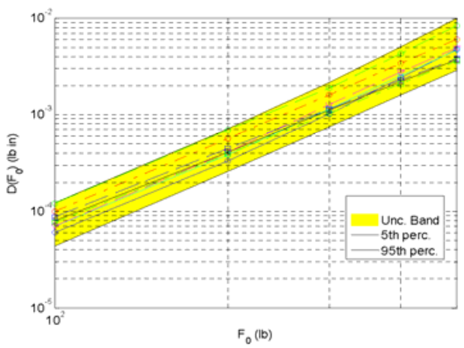
$$\rho(\phi) = S\,\delta(\phi - \phi_{max}) + R\,\phi^{\gamma}\left[H(\phi) - H(\phi - \phi_{max})\right]$$

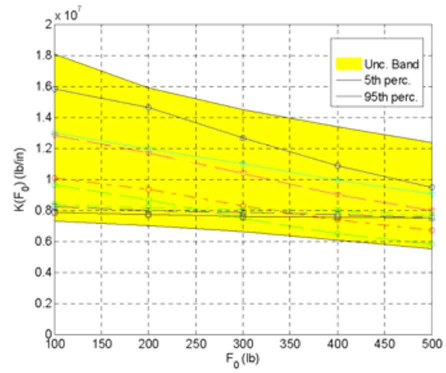
Introduction of kinetic versus static friction to the model…

Mignolet et al.

Uncertain 5-Parameter Iwan Model Validation

Predicted uncertainty bands:





Mignolet et al.

Using Maximum Entropy to Characterize System Matrices in Reduced Order Models

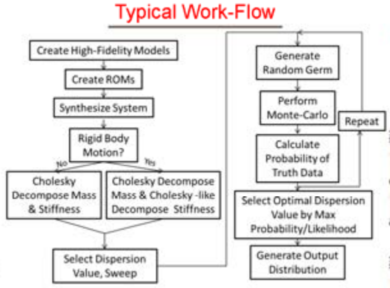
Status Quo

- System characterized on mode-by-mode basis
- Only parametric uncertainty can be quantified
- Analysis performed as afterthought
- ROMs used to make Monte-Carlo feasible for uncertainty propagation

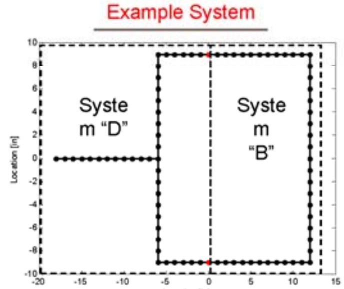
Maximum Entropy

- Technique developed by Soize
- Able to characterize randomness of a semi-positive definite matrix such as mass or stiffness with a single dispersion parameter
- Separates parameter and model form uncertainty
- Generate stochastic matrices that maintain definiteness
- Use Cholesky decomposition and add a random germ
- Monte-Carlo analysis to determine output distribution
- Sweep through dispersion parameter to optimize agreement with truth data

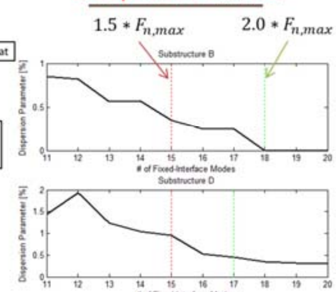
Typical Work-Flow



Example System



Model Form Uncertainty Preliminary Results- Convergence of Dispersion Parameter



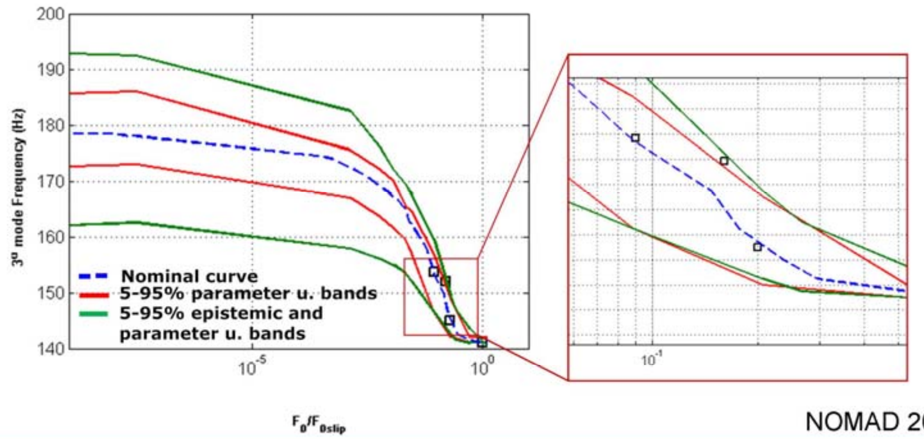
Next Steps

- Expand to other ROMs
- Correlate substructure to full system
- Use Kernel estimators instead of tolerance band
- Use other truth data
- Expand substructure to synthesized system uncertainty

Bonney & Kammer

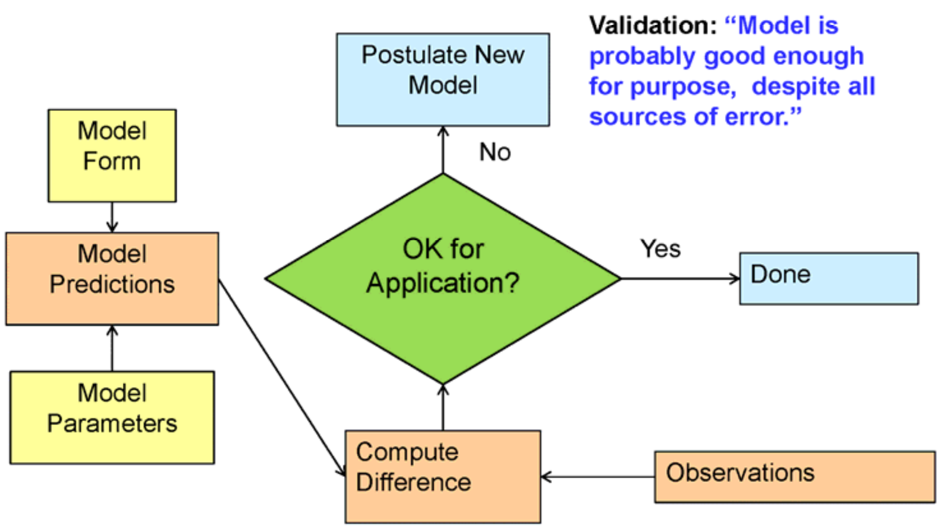
Accounting for epistemic uncertainty

- Even using calibrated models, uncertainty bands only considering aleatoric uncertainty are non-conservative
- Introducing epistemic uncertainty (see Soize, 2011), generally does better in predicting ranges that include data *not used to calibrate* the model



NOMAD 2014

The Validation Process



Segalman

Eventual Implementation of Prediction Methods in Commercial Numerical Codes

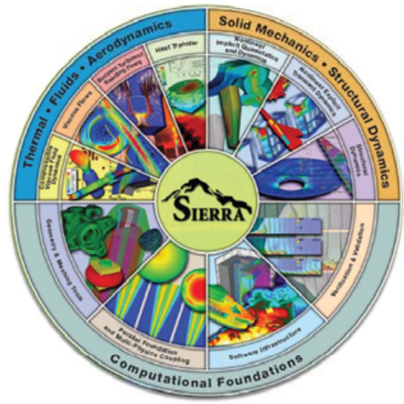
Matthew Brake (lead, Sandia), Melih Eriten (lead, Wisconsin), Dan Brown (AWE), Hugh Goyder (Cranfield), and George Ostermeyer (Braunschweig)

Progress Since 2012

- Development of new/more mature ROM analysis packages
  - Force, ROMAC, ROMULIS, SUPMECA's toolbox…
- Numerical round robins to assess/determine joint modeling best practices
- Next generation of Iwan models
  - Modal Iwan, RIPP joint
- Further work on nonlinear normal modes as intransient properties of a system
  - Including analysis of amplitude dependent properties of stiffness and damping

Case Study: Sierra

- In house code developed at Sandia
- Designed to be massively parallel
- Dedicated development teams
- Iwan models incorporated
- Issue: the joint models aren't used by analysts
  - Too computationally expensive
  - Mystery as to how to specify parameters



Primary Issues

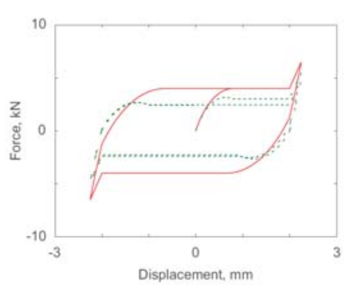
- Efficiency
  - Without an efficient implementation, joint models are unlikely to be adopted by analysts
- Accuracy
  - The Iwan/RIPP joint model, or its future successors, is an improvement over existing techniques (linear springs), but its still not predictive
- Usability
  - In order to be widely adopted, the model must require parameters that are easily found (contrast a Prony series with a Kelvin-Voigt model)

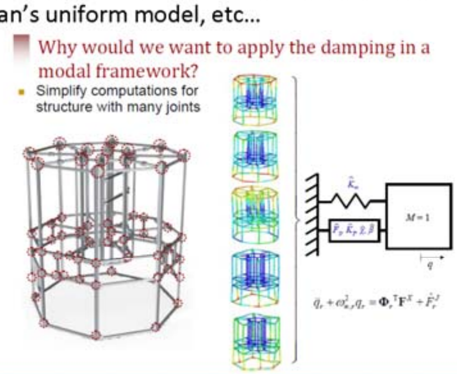
Existing Research on Efficiency

- Model reduction techniques incorporating nonlinearities (a non-exhaustive list)
  - Frequency based substructuring (Reuss et al., 2012; de Klerk et al., 2008)
    - ROMAC @ Stuttgart
  - FRF based model reduction (Petrov, 2010; Popp and Maagnus, 2002)
    - FORSE @ Imperial
  - Other harmonic balance methods (Firrone et al., 2011; Tangpong et al., 2008; Dion et al., 2010-present)
    - Nonlinear Dynamics toolbox @ SUPMECA
  - Non-smooth basis functions (Brake and Segalman, 2013; Milman and Chu, 1994)
    - ROMULIS @ Sandia
- Many approaches, but little consensus
- Collaborations directly comparing methodologies are necessary
  - **Outcome of last workshop** – collaboration between Sandia, Stuttgart, Imperial College London, and Wisconsin to assess frequency based substructuring, harmonic balance techniques, and non-smooth basis function methods; culminated in founding of the NOMAD Research Institute

Next Generation of Iwan Models

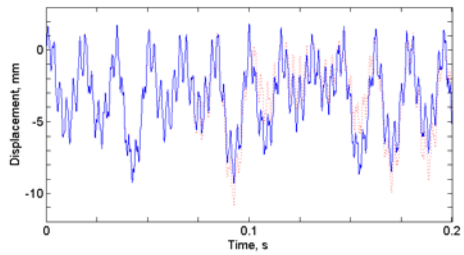
- Modal Iwan
  - If a structure does not change due to the joints, why not model dissipation in a modal sense, with an Iwan element attached to each mode; allow for modeling at a structural level instead of at discrete locations
- Analytical Iwan (RIPP joint)
  - Solution for numerical stability issues observed in discretized Iwan, demonstrated to be ~3x more computationally efficient, extended to include Mignolet's 5 parameter model, Iwan's uniform model, etc…

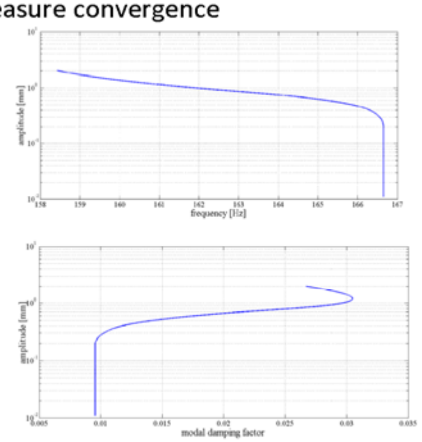




Assessment of Accuracy

- How do you compare two different models of the same nonlinear system?
  - Time histories, dissipation, strain energy, L_2 norm, etc.
  - Use of nonlinear normal modes to measure convergence
  - Intransient model properties…



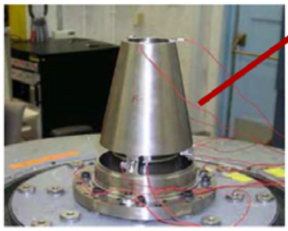


Hilbert Transforms
Short Time Fourier Transforms
Morlet Wavelet Analyses

NOMAD 2015

Usability

- Example of the Iwan model
- Long history of development: Baushinger, 1886; Masing, 1926; Prandtl, 1928; Ishlinskii, 1944; and Iwan, 1966 and 1967
- Four parameter Iwan model: Segalman, 2005
  - **Usability issue**: determining those four parameters (β, χ, K_T, F_S)
  - Still not predictive…



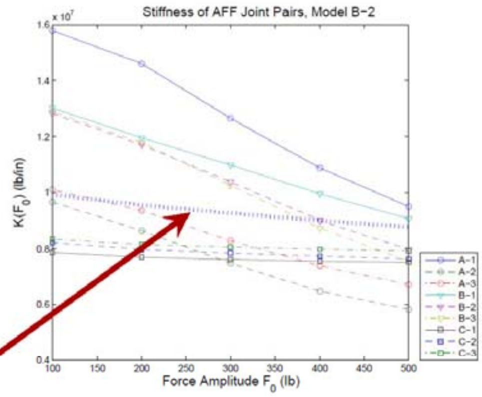


Figure 12.15. Stiffness of AOS Joint Pairs.

The thick dotted line is the stiffness of the four-parameter Iwan model, calibrated to reproduce the dissipation curve with fidelity and to match the stiffness of a load of 400 lb.

From the joint handbook (SAND2009-4164)

Conclusions: Not ready yet to talk about implementation on a large scale, but many lessons learned for studying things at the research level…

Derivation of Constitutive Equations Based on Physical Parameters

Lothar Gaul (lead, Stuttgart), Randy Mayes (lead, Sandia),
Norbert Hoffmann (Hamburg), and Mike Starr (Sandia)

Two Sets of Problems

1. How can we make in-situ measurements of quantities at/near the contact interface?

2. How can we use experimental knowledge to postulate an improved friction model?

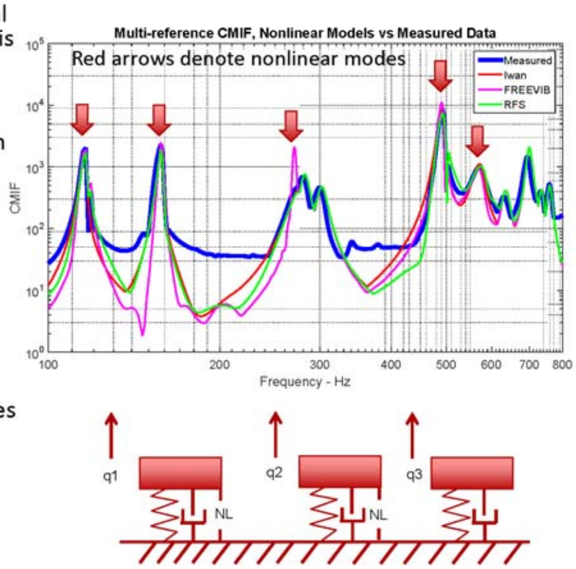
Progress Since 2012

- Mignolet's 5 parameter Iwan model in which static and dynamic coefficients of friction are different. 5th parameter is ratio between the two coefficients.
  - Fitting of the Sandia joints handbook data with the 5th parameter model was very successful – close fit for all 9 samples
- Assessment of modal models using measured data by Mayes et al.

Nonlinear Modal Model Identification – Mayes/Pacini/Roettgen

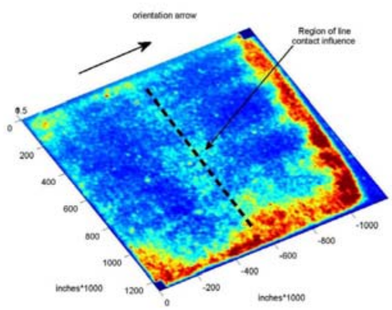
- We follow the suggestion of M. Mesh and D. Segalman to develop a modal model with nonlinear elements. This was first fleshed out by Deaner and Allen.
- A low level impact test was implemented on hardware to obtain a linear modal model for 17 modes.
- High level impacts showed 5 modes that were significantly nonlinear.
- Impact data were modal filtered.
- 3 different nonlinear model forms were fit: 1. Cubic spring/damper 2. Feldman Freevibe 3. Iwan
- All 3 nonlinear models produced better results than linear model which over predicted multiple modes by almost 100%

$$\overline{\mathbf{x}} = \mathbf{\Phi}\overline{\mathbf{q}}$$



The Contact Patch Process Zone is 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.

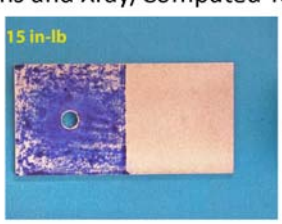


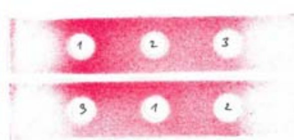
- The digitized film shows an apparent assembly misfit, periodic machining marks, and local surface roughness characteristics.

Methods to Determine Contact Area of Two Plates with Bolt Through Center

- Prussian blue ink
- Pressure sensitive film
- Ultrasonic measurement
- Implanted Ions and Xray/Computed Tomography

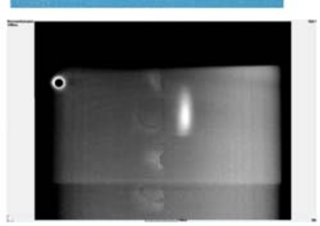
Prussian Blue

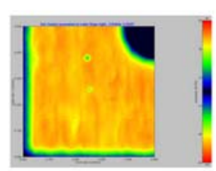




Fuji Pressure Sensitive Film

CT Slice





Ultrasonic

Round Robin/Benchmark for Hysteresis Measurements

David Ewins (lead, Imperial), David Nowell (Oxford), Muzio Gola (Torino), Christoph Schwingshackl (Imperial)

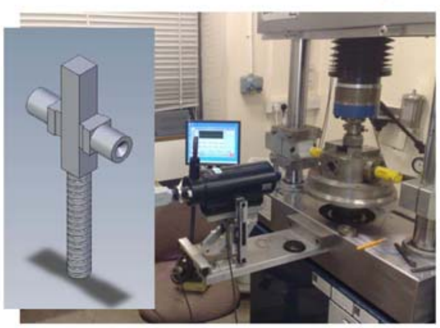
Difficulties in Modelling Contacts

- In general, the normal and tangential stiffnesses of a joint need to be experimentally measured, along with the friction coefficient
- These properties may change with time (e.g. as the contact wears, with position, and with load)
- Progress is needed towards a model of interface behaviour, which is based on more fundamental properties (material properties, surface geometry etc).
  - We also need to understand how to incorporate the interface behaviour into global (FE) models of the system

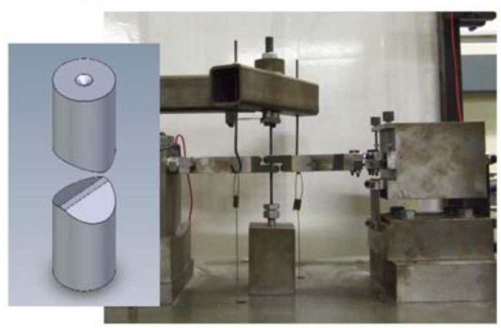
Progress Since 2012

- Identification of another group researching this challenge at Cambridge
- Several new test rigs for *in situ* damping and hysteresis measurements developed at Torino

Measurement of Contact behaviour – Oxford and Imperial rigs

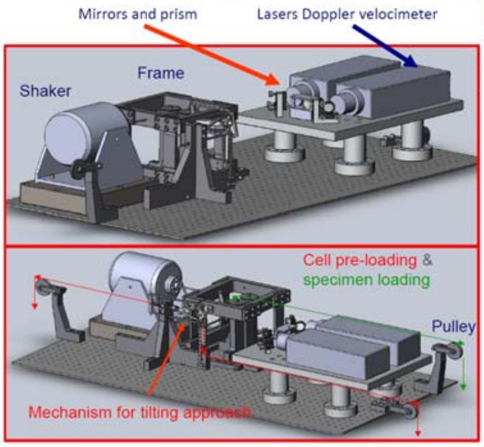


- 80 mm^2 flat and rounded contact
- 1Hz Frequency
- 0.6mm sliding distance
- Displacement measurement by remote LVDT or digital image correlation

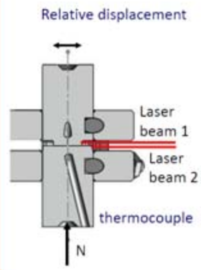


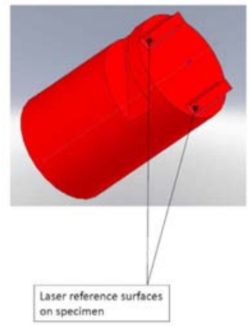
- 1 mm^2 flat on flat contact
- ~100Hz Frequency
- 30μm sliding distance
- Displacement measurement integration of LDV measurements

New Test Rigs Developed at Torino



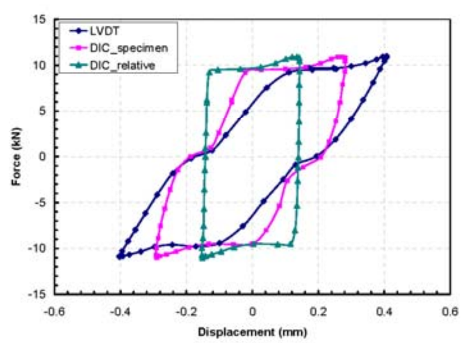
Max temperature: 1000°C
Temperature control: ± 1°C
Displacement: 2-80 μm
Displ. control: ± 0.5 μm
Normal force: max 300 N

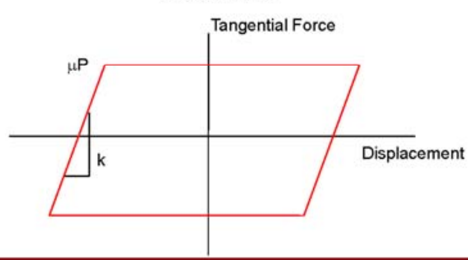




- Developed for studying nominally "flat" contact surfaces
- Additional new rigs for studying dampers, resonant tests of dampers on blades, and tip timing measurements on a spinning rig

Measured and Idealised Hysteresis Loops





- Idealised loop is characterised by contact stiffness, k and friction coefficient, μ
- These can be reasonably representative of real loops

Round Robin/Benchmark for Measurements and Predictions of Dissipation in Standard Joints

Hugh Goyder (lead, Cranfield), Matt Allen (lead, Wisconsin), Lothar Gaul (Stuttgart), Laura Jacobs, Randy Mayes, and Matthew Brake (Sandia), Gael Chevallier (Supmeca), Norbert Hoffmann (Hamburg), Kai Willner (Erlanger/Imperial), Christoph Schwingshackl (Imperial)

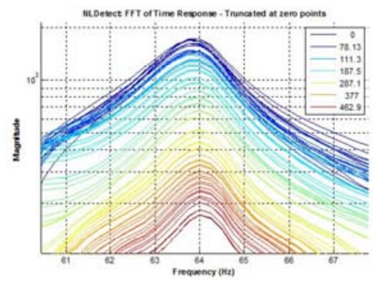
Progress Since 2012

- Multiple benchmark systems proposed/investigated
  - Ampair 600 Wind Turbine
  - Catalytic Converters
  - Square 4 Bolt Plate
  - SUPMECA/FEMTO-ST's system
  - Resonator Structures
  - Brake-Reuß Beam

  - Sumali beam proposed, but not enough damping
  - Goyder proposed several systems, but similar challenges

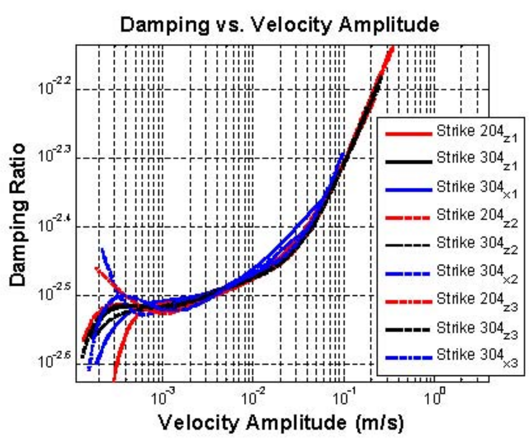
Ampair 600 Wind Turbine

- Model available on Substructuring Wiki page
  - substructure.engr.wisc.edu
- Website contains both experimental data along with FE data
- Contributors from 8 different universities and organizations



Photos curtesy of substructure wiki

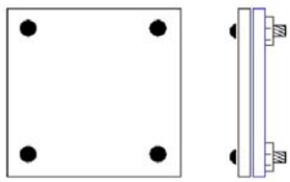
Possible Testing Round Robin Structure

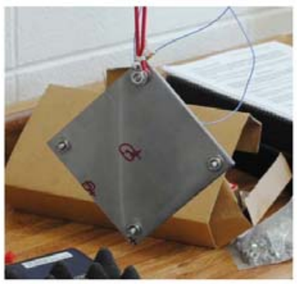


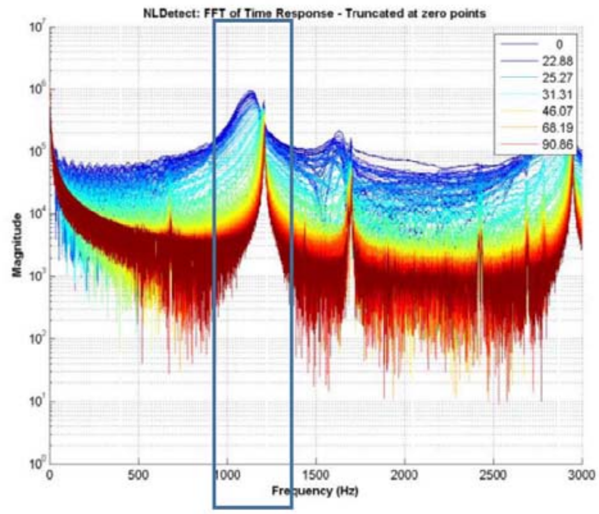
- Industrial structure assembled from two catalytic converters.
- Tests to date show clear nonlinear behavior due to the joint, even at normal clamping loads and with factory (metal) gaskets.
  - We have 3 copies of this hardware that we could possibly share. (It can be purchased at automotive parts stores, but each assembly costs $3k or more.)
  - Disadvantage: this will be quite challenging to model in FEA!

Square Four-Bolt Plate

- See Dan Segalman's demonstration...

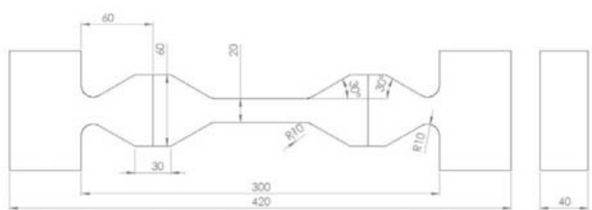


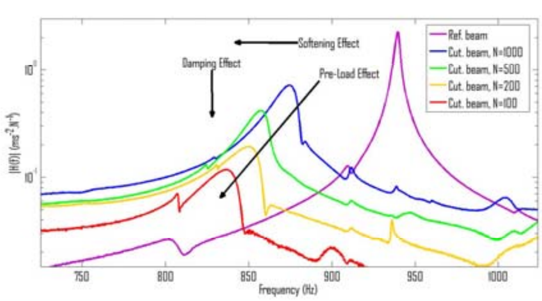




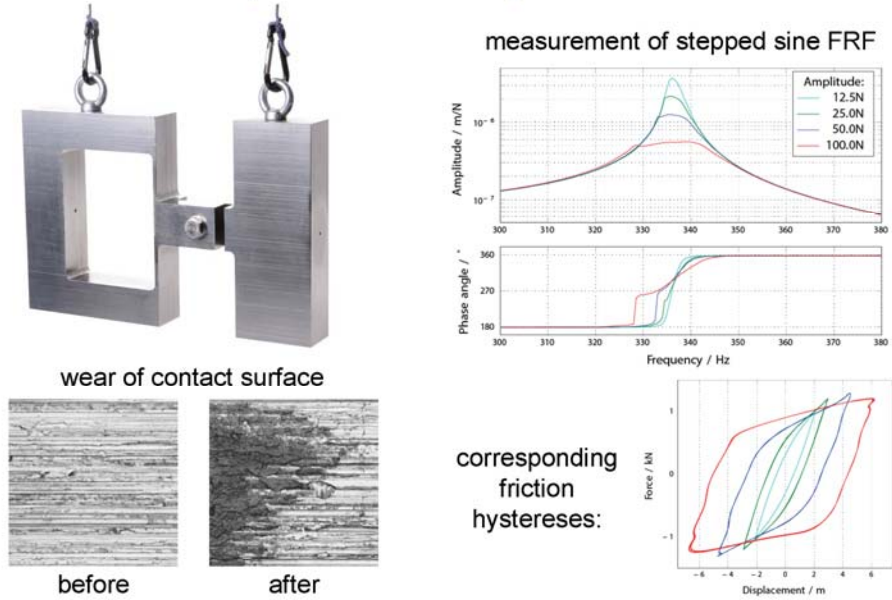
SUPMECA/FEMTO-ST

- Beam has large damping by having large interface dimensions
- Designed to avoid coupling between normal stresses at the interface and vibration



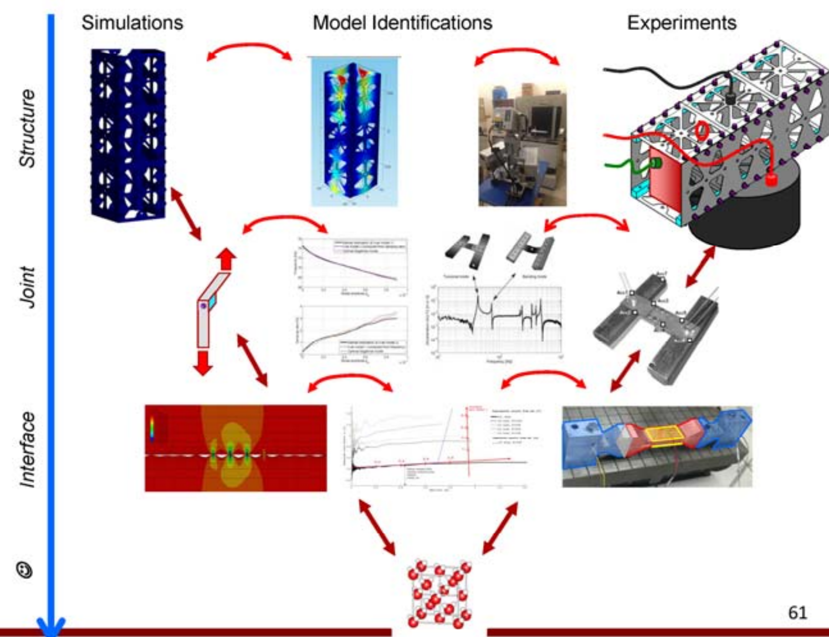


Measurement (and calculation) of resonator structure



Courtesy of K. Willner, Erlangen-Nurnberg

A multi scales/domains benchmark



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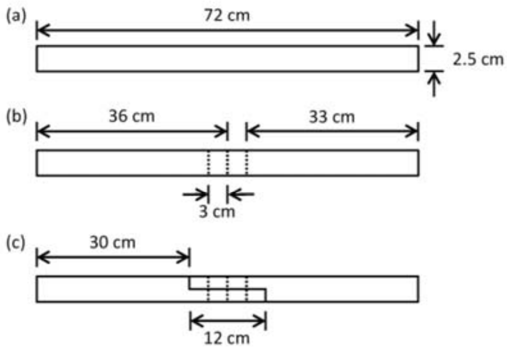
ASME IDETC

Brake-Reuß Beam

- Brake-Reuß beam: 72 cm long beam with a lap joint
- Multiple versions to assess contribution of joint to dynamics



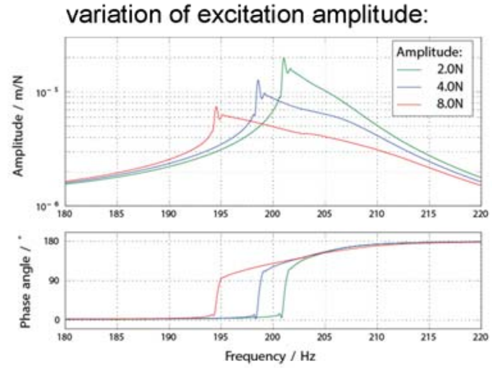
Measurement (and calculation) of beam structure

beam equal to [Brake et. al., 2014]

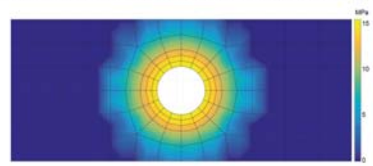
modified beam with only one bolt

tangential and normal investigation

variation of excitation amplitude:

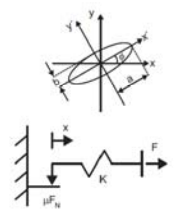
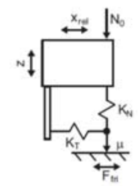
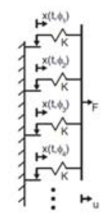


FE-simulation of contact normal stress



Courtesy of K. Willner, Erlangen-Nurnberg

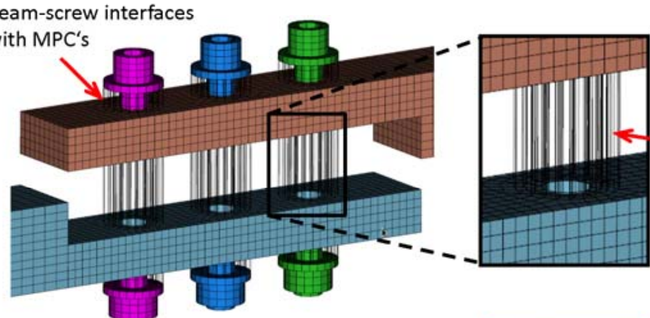
Numerical Round Robin at NOMAD 2015

	Stuttgart Approach	Imperial Approach	Sandia Approach
FE Tool	CalculiX	NASTRAN	SIERRA/SD
Model Fidelity	Craig-Bampton ROM	Hybrid ROM	Craig-Bampton ROM
Nonlinear Element	2D Jenkins Element 	3D Contact Element 	Iwan Element, RIPP 
Nonlinear Solver	ROCMAN	FORSE	ROMULIS
Solver Type	Harmonic Balance	Multi-Harmonic Balance	Transient Integration

- Additional modeling conducted in Abaqus and Hyperworks

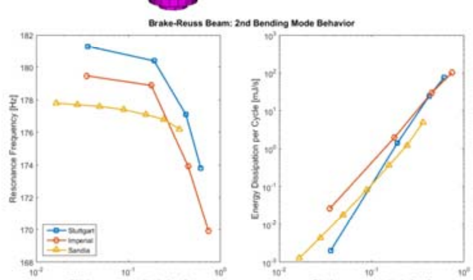
Numerical Round Robin at NOMAD 2015

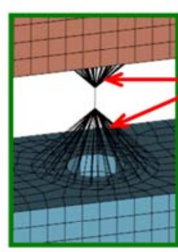
Tie coincident nodes on beam-screw interfaces with MPC's



Stuttgart/Imperial Approach

Tie coincident nodes on the friction interface with Jenkins/3D contact elements.





Sandia Approach

Connect interface nodes to a virtual node with NASTRAN RBE3 element spider.

Tie virtual nodes with an Iwan element.

Status of Challenges

Significant Progress

- Round Robin/Benchmark Exercise for Measurement and Prediction of Dissipation in Standard Joints
- Epistemic and Aleatoric Uncertainty in Modeling and Measurements
- Eventual Implementation of Prediction Methods in Commercial Numerical Codes

Some Progress

- The Economics of Jointed Structures
- Derivation of Constitutive Equations Based on Physical Parameters

Little Progress

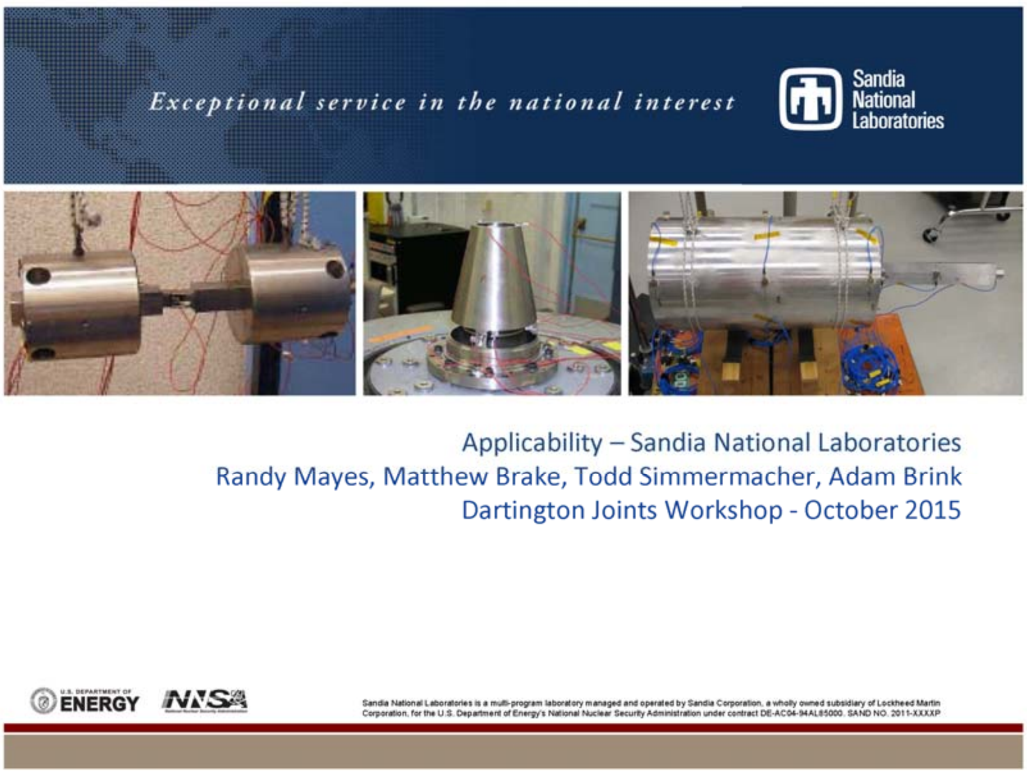
- Round Robin/Benchmark Exercise for Hysteresis Measurements
- Defining the Mechanisms of Friction
- Time Varying Model Parameters, Modeling and Experimental "Surface Chemistry"

5.2. Session 1: Applicability

Ed Green: Perspectives from the Aeroturbine Community

As of publication, these slides were not yet available due to review and approval issues internal to Rolls Royce.

Randy Mayes: Applicability – Sandia National Laboratories



Applicability – Sandia National Laboratories

Technical and Economic Aspects of Mechanical Joints for SNL Requirements - Categories

- Sandia dynamic response requirements and economics
- Social challenges of implementing technology
- Major physics challenges
- Major model implementation challenges
- Major design challenges
- Major experimental challenges

Applicability – Sandia National Laboratories

SNL Dynamic Response Requirements and Economics

- Define dynamic specifications for component designs
- Design components to successfully meet functional requirements and survive dynamic specifications

Applicability – Sandia National Laboratories

SNL Social Challenges to Implementing Technology

- To implement technology on a production basis, it moves serially from Research ➡ Development ➡ Application
- SNL struggles to get through the Development phase.
  - The "new" research must get socialized on the way to Application.
  - Has the research approach really addressed all the technical problems?
  - Is it "really" the way to go?
  - Does the relevant community know about the research, understand it and endorse it?
  - Does it require a change in the standard approach to "doing business" .
    - The approach may require new code, and not fit into the old code.
    - The testing community may need to learn new testing techniques and acquire new equipment.
    - The project groups have to be educated to ask for the new approach instead of the status quo approach.
  - The Development phase requires several stakeholders to implement the approach which may be painful (change/investment/boring/frustrating).

Applicability – Sandia National Laboratories

SNL Major Physics Challenges

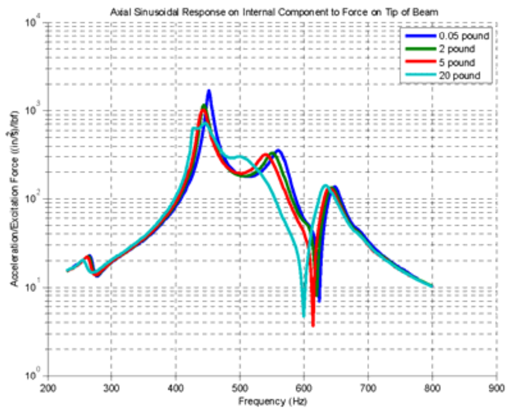
- The Application state of the art for dynamic modeling at SNL is updating FE model stiffness parameters from modal test data and inserting modal damping derived directly from the low level modal test. Sometimes damping is calibrated from operational shock test data.
- Deriving specifications from linear models for components that actually experience macroslip in an environment typically is unacceptable. Models overpredict >> than a factor of 2.
  - Full system testing that demonstrates this problem comes too late in the program development to impact the design phase.
  - If we had a predictive approach that would get within a factor of 2 it would probably be sufficient.
  - Probably, in some cases, we don't even recognize that macroslip has occurred in full system or subsystem ground tests.
- We need simplified nonlinear models of a component analogous to a "wrist watch" to perform uncertainty quantification (UQ) analyses to see which design parameters are important to keeping the "wrist watch" functional in anticipated environments.
  - Most of these nonlinearities are associated with interfaces between parts. The need is for simplified models so that many computational runs can be performed in a short time to identify the important parameters.

Applicability – Sandia National Laboratories

Sandia National Laboratories

SNL Physics Challenges

- A lesser, but significant problem, is predicting (or even calibrating) damping (energy dissipation). Amplitudes of specific modes can be off by a factor of 2 or 3 in operational response if we utilize modal damping derived from a low level modal test.



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SNL Major Model Implementation Challenges

- The 4 parameter Iwan nonlinear elements we can implement sometimes do not capture the physics to our satisfaction.
  - One analysis required addition of a 5th parameter[2].
  - Iwan not typically used for hardening springs or softening springs where damping remains constant.
  - Implementing Iwan nonlinear elements at many degrees of freedom can be overwhelming.
  - Analytical model researched, but not developed [3].

[1]:

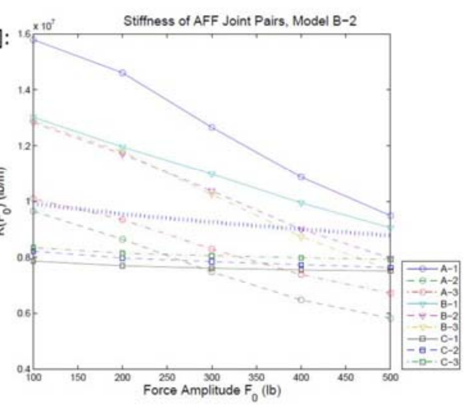


Figure 12.15. Stiffness of AOS Joint Pairs.

The thick dotted line is the stiffness of the four-parameter Iwan model, calibrated to reproduce the dissipation curve with fidelity and to match the stiffness of a load of 400 lb.

Applicability – Sandia National Laboratories

SNL Major Model Implementation Challenges

- Predictive models are necessary
  - Currently no means to predict dynamic properties *a priori*
  - Onset of macroslip in complicated systems/loading scenarios
  - Prediction of damping to within a factor of two for microslip events
  - Even submodels or 90% solutions could work
- Implementing solvers that can run nonlinear problems efficiently in time domain.
  - Transient solvers necessary for studying environmental specifications
  - Orders of magnitude more computational time required than harmonic balance (HB) methods
- Implementing nonlinear frequency domain solvers.
  - Partly a cultural challenge as HB methods work well, but aren't robust for constitutive modeling
  - Issue still remains in deducing nonlinear damping parameters from HB methods
  - Alternative methods being investigated
- Simplified (but adequate) nonlinear mechanism models are desired, but not easily derived.

Applicability – Sandia National Laboratories

SNL Major Design Challenges

- Notion that we should design systems to have the damping that we want, rather than to calibrate models after the fact
  - Need for repeatable or predictable standard jointed interfaces
- Repeatability and variability still large issues
  - Manufacturing tolerances unavoidable
  - Ability to specify which tolerances are more important and which are less important could lead to moderate savings in costs
- Benefits of improved design ability:
  - Reduced cost
  - Reduced weight
  - Reduced number of iterations required for design qualification (largest source of savings)

Sandia National Laboratories

SNL Major Experimental Challenges

- In regards to full system and subsystem nonlinear experiments for nonlinear parameter identification:
  - Operational levels are needed to identify nonlinear parameters
    - Standard modal testing equipment may not be sufficient for obtaining operational levels
    - Standard shaker table vibration testing introduces other sources of damping besides test article physics which can overwhelm the desired identification of nonlinear parameters
  - Systems and subsystems have many degrees of freedom which can inhibit isolation of nonlinear parameters for identification
- In regards to first principles experiments:
  - Most techniques are focused on measuring a hysteresis loop response
  - What is really needed is measurement of distributed joint interface forces and displacements to provide validation data for a proposed constitutive model

Sandia National Laboratories

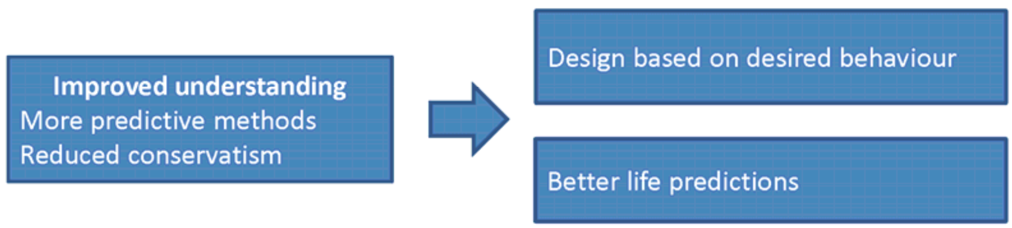
References

1. D.J. Segalman, D.L. Gregory, M.J. Starr, B.R. Resor, M.D. Jew, J.P. Lauffer, and N.M. Ames, "Handbook on Dynamics of Jointed Structures," Technical Report SAND2009-4164. Sandia National Laboratories, Albuquerque, NM. 2009.
2. X.Q. Wang and M.P. Mignolet, "Stochastic Iwan-Type Model of a Bolted Joint: Formulation and Identification," 32nd International Modal Analysis Conference, 2014.
3. M.R.W. Brake "A Reduced Iwan Model that Includes Pinning for Bolted Joint Mechanics," 34th International Modal Analysis Conference, 2016.

Why improved understanding of jointed interfaces is important to AWE

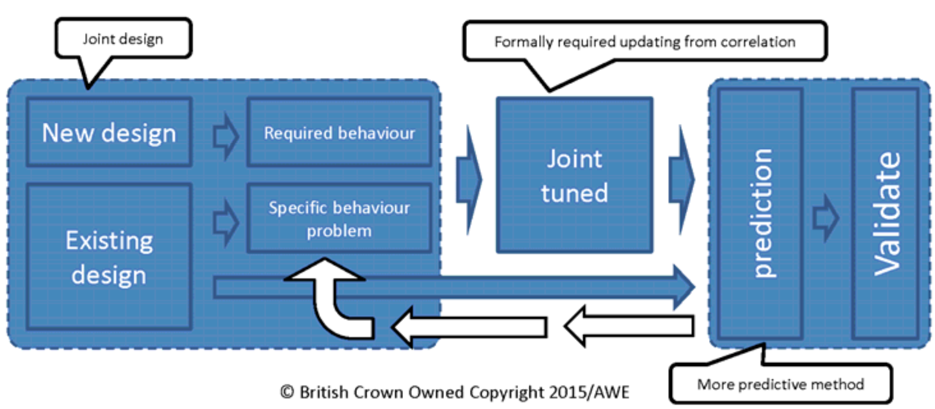
What better joints modelling means to AWE

- Interfaces can have a significant effect on dynamic behaviour
  - Stiffness distribution
  - Energy dissipation



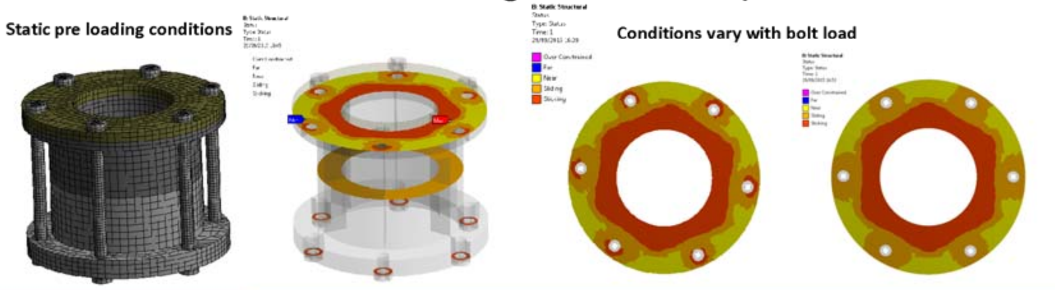
Desired position

- Selection of joint design based on behaviour
  - At concept stage
  - How adjusting joint design would tune behaviour
  - Tune for dissipation of energy
- Predictive model at an early stage
  - Validation for evidence, less updating required



Our position now

We believe significant progress in predicting stiffness has been made and demonstrated through commercially available codes:



We are concerned with energy loss

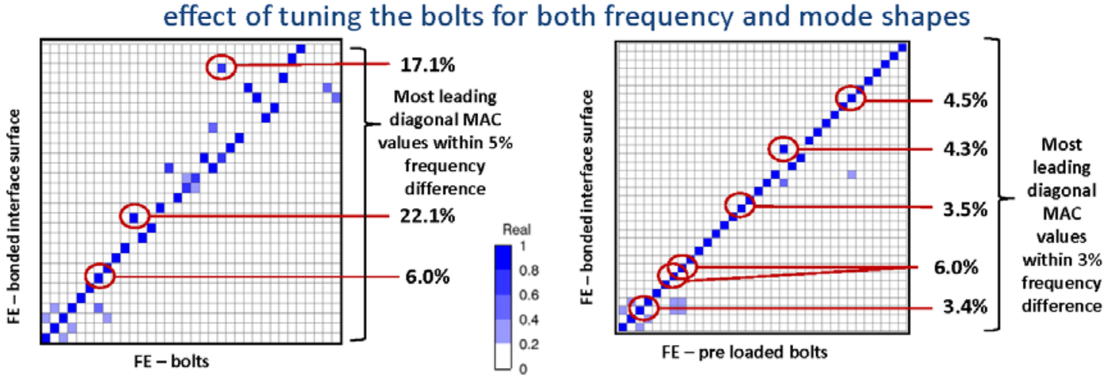
- Joints effects on damping is not fully understood

We can perform model updating based on test
- Influence initial assumptions
- Improved model post manufacture

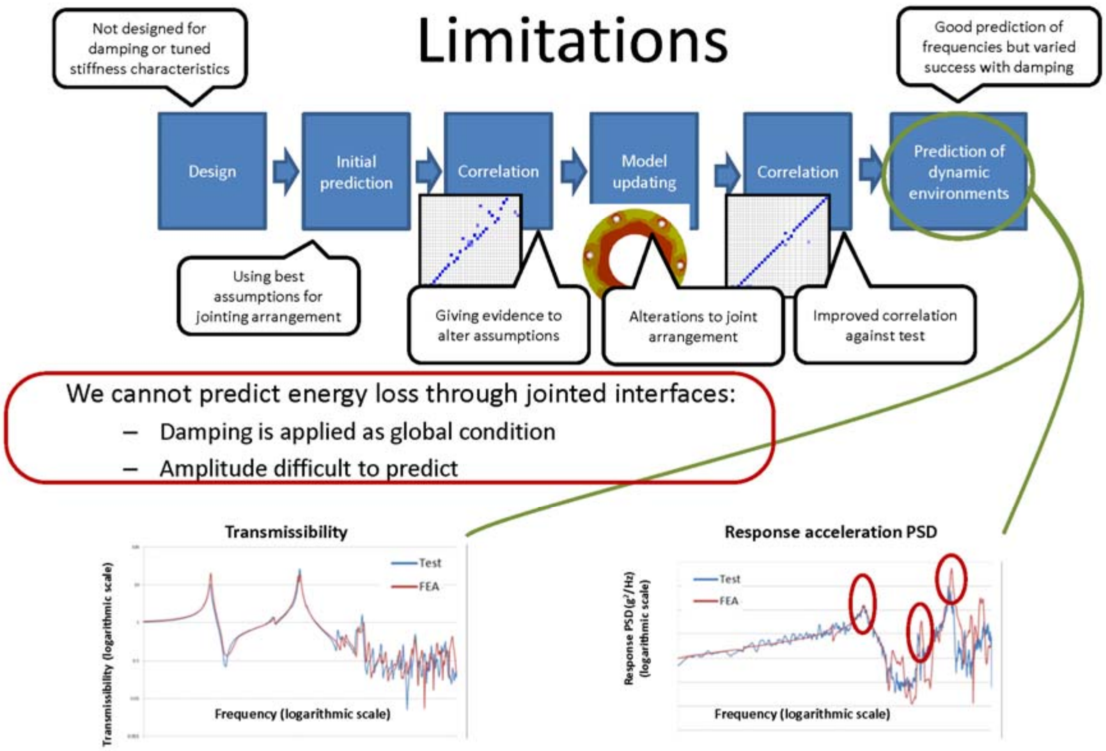
What this looks like

- Make good predictions of system level frequencies:
  - Within a few percent
- Tune stiffness of bolts affecting contact area and condition (near, sliding, bonded)
  - Using rigid elements and coincident nodes
- Pre stressed solid elements
  - Frictional contact (during static solution)
  - Penalty overlap based on known torque

In a bolted assembly structure made up of metallic component parts we observe the effect of tuning the bolts for both frequency and mode shapes



Limitations



Moving forward

- **Characterisation of joint types**
  - **Bolted arrangements seemed like a reasonable starting point**
- What influences the dissipation of energy
  - How can this be tuned
  - How can this be represented within numerical models – simple examples
- Can we emulate amplitude and frequency for research rigs
- Could we apply this to a larger model of a known configuration with a variation in arrangement
- How would predictions of a joint designed for specific behaviour compare with manufacture of it

- **Prediction of joint behaviour**
  - **The better the assumptions at the static level of the analysis (preloading) the greater the level of correlation appears**
- Explore emerging static analysis options within FE codes to exploit within dynamic analysis
  - Options for wear may be used for the geometric updating for bedding in of interfaces
  - Bolt thread representations may help to define the contact regions more precisely using geometric modification

Reference: AWE/CL/20151005-brown daniel-01 Date: 27/10/15

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Signed by: ....
The Author
Date: 27-10-15
Name: Daniel Brown
Position: Senior Structural Dynamicist

Signed by: ....
Date: 5/11/15
Name: CARL SMITH
Position: Commercial Manager
(On behalf of MOD)

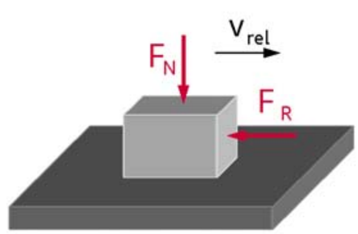
Merten Tiedemann: The Relevance of Joints in Friction Brake NVH; Friction-Induced Vibrations in Nonlinear Multi-Component Systems

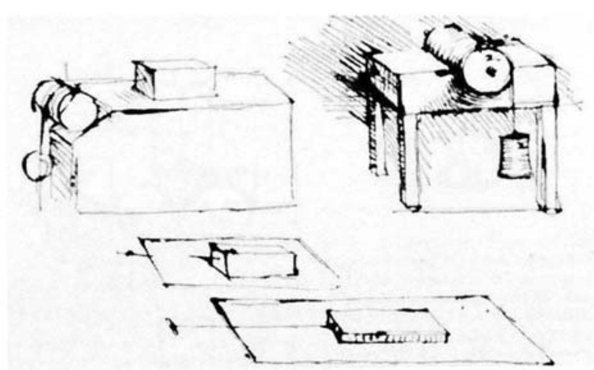
Agenda

- Introduction
  - Friction Brakes
  - Noise, Vibration, Harshness
- NVH Development of Friction Brakes
  - Workflow
  - Noise Countermeasures
- Noise Reduction by Smart Design of Joints
  - Preconsiderations
  - Case Study
- Summary

Introduction
Friction Brakes

- **Vehicle Brakes:**
  - High performance products
  - Relying on **"ancient" mechanism**



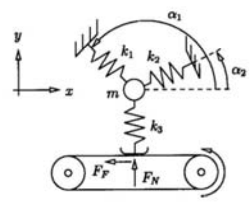


DA VINCI (1452–1519): first systematic studies on friction

Audi Vorsprung durch Technik

Introduction
Noise, Vibration, and Harshness

- **Friction brakes: tendency to generate noise**
  - creep groan
  - moan
  - judder
  - **squeal**

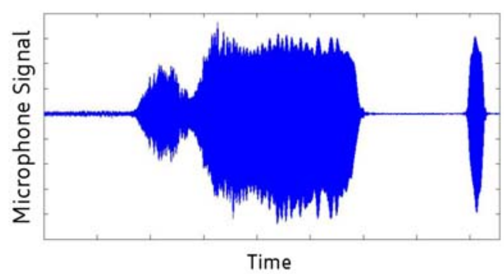


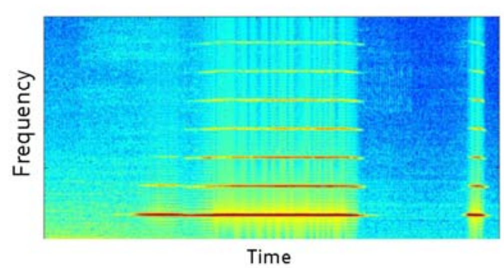
Most critical...
- ...highly customer relevant
- ...no predictive tool available, yet
- ...influence of axle components, environment etc.

Energy feed-in into the system **by friction** (→ onset of periodic vibrations)

Introduction
Squeal in a Nutshell

- Classification:
  - Nonlinear oscillation with characteristic frequency and deflection shape
- Characteristics:
  - f = [1 – 20] kHz
  - SPL >130 dB(A)
  - Brake disc = speaker
- Occurence:
  - „Stop at red lights."

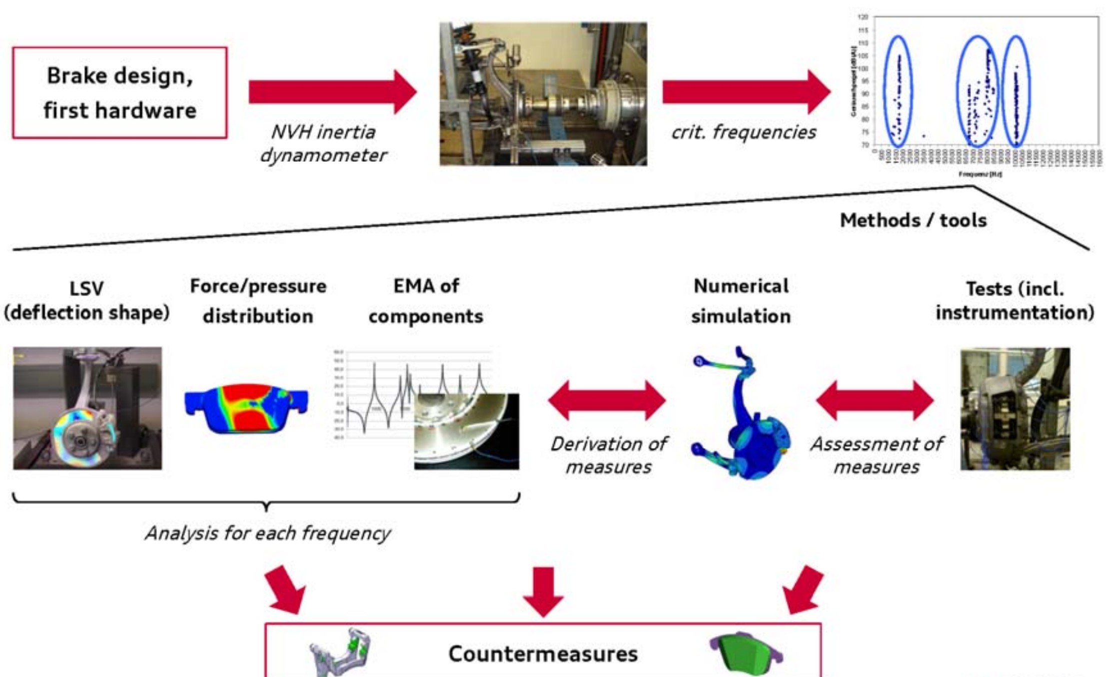




Agenda

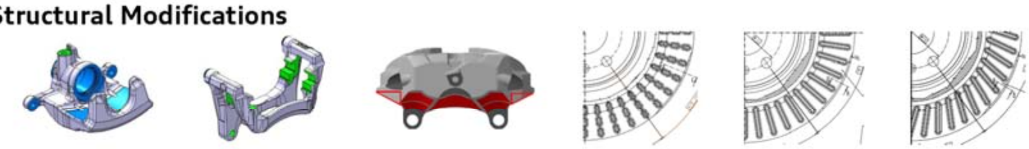
- Introduction
  - Friction Brakes
  - Noise, Vibration, Harshness
- **NVH Development of Friction Brakes**
  - **Workflow**
  - **Noise Countermeasures**
- Noise Reduction by Smart Design of Joints
  - Preconsiderations
  - Case Study
- Summary

NVH Development of Friction Brakes
Workflow

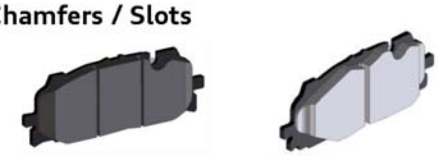


NVH Development of Friction Brakes
Noise Countermeasures

Structural Modifications



Chamfers / Slots



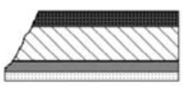
Friction Material Modifications

- Material mixture
- Stiffness
- Damping properties (underlay
- Thermal treatment („Scorchin

Joint Design / Manipulation

Next slides

Damping Shims



Agenda

- Introduction
  - Friction Brakes
  - Noise, Vibration, Harshness
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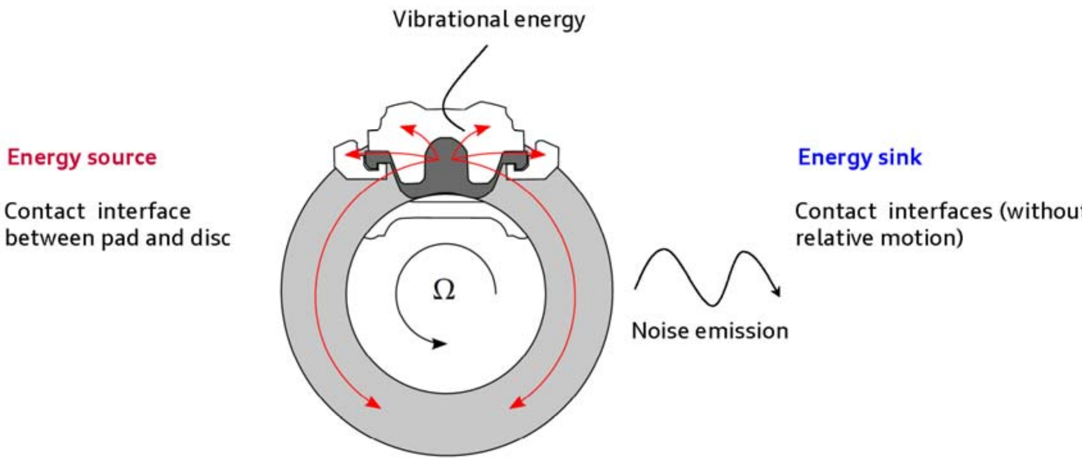
  - Workflow
  - Noise Countermeasures
- **Noise Reduction by Smart Design of Joints**
  - **Preconsiderations**
  - **Case Study**
- Summary

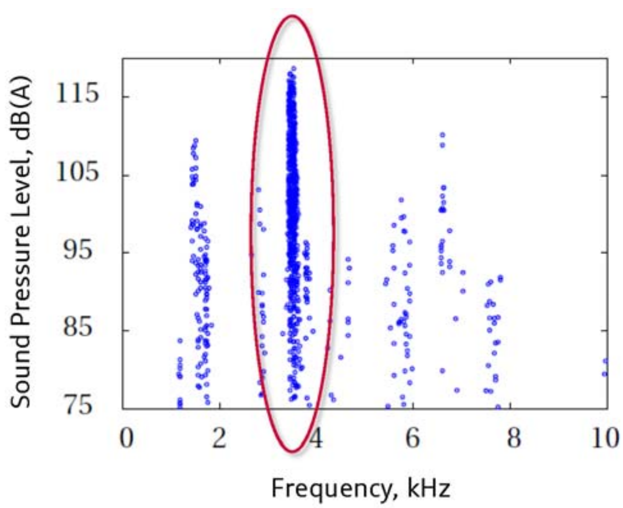
Noise Reduction by Smart Design of Joints
Energy Sources and Sinks

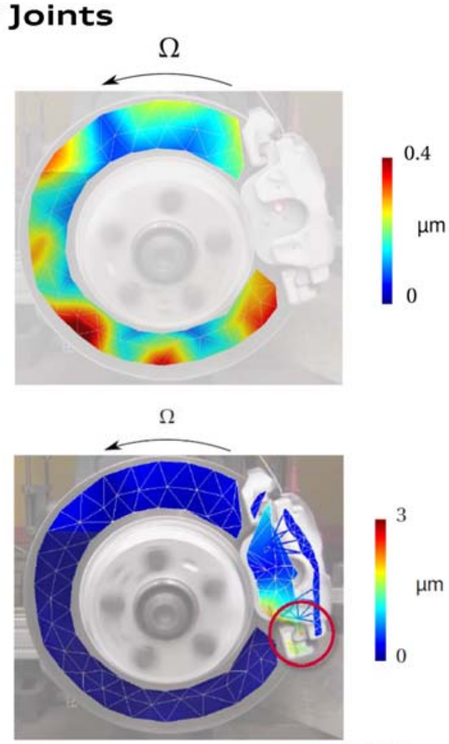


- **Aim:** Reduction of noise in friction brakes by smart design of joints
- **Hypothesis:** Manipulation of surfaces at "active" joints influences noise propensity

Noise Reduction by Smart Design of Joints
Vibration Behaviour

- Brake dynamometer results: significant no. of ev

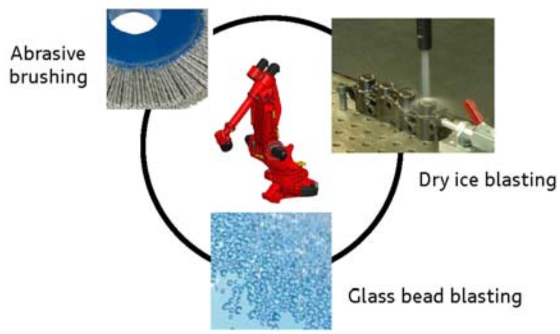




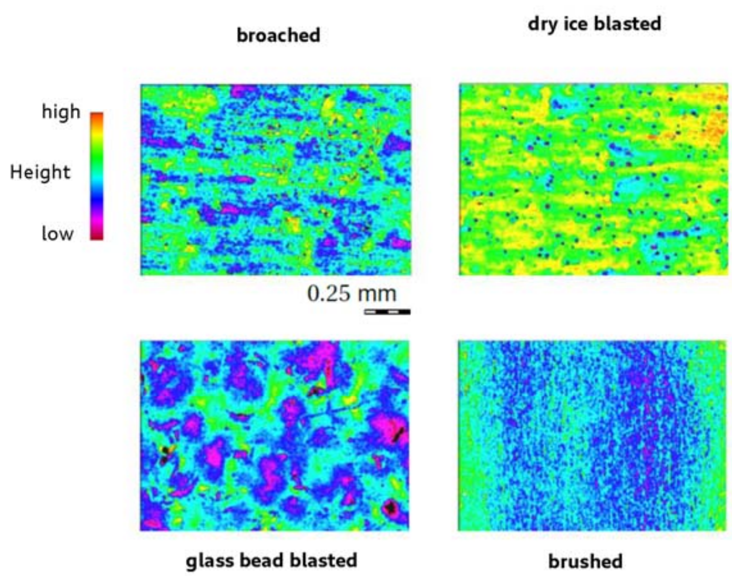
Noise Reduction by Smart Design of Joints
Selection of Machining Processes

- Requirements:
  - Difference in machining results in terms of surface integrity
    - Surface texture (material share, roughness, topography)
    - Subsurface structure (cracks and stresses)

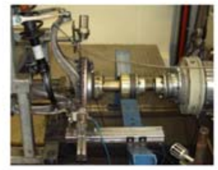
    } Surfaces with different mechanical properties
  - Suitability for integration in existing production lines in automotive industry
  - Low process times and costs
- Chosen processes:
  - Dry ice blasting
  - Glass bead blasting
  - Abrasive brushing

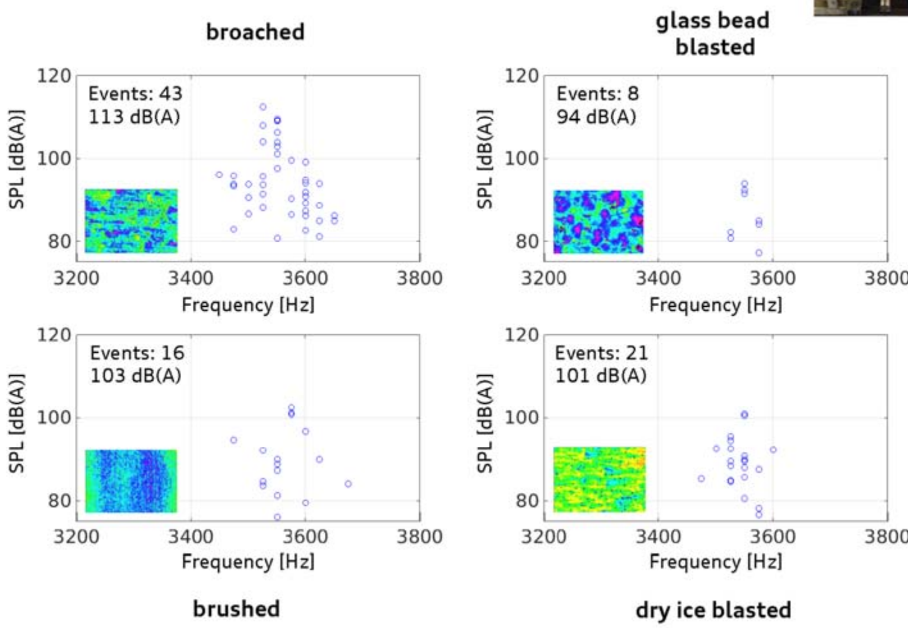


Noise Reduction by Smart Design of Joints
Qualitative Height Maps of the Surfaces



Noise Reduction by Smart Design of Joints
Results of Dynamometer Testing





14 Merten Tiedemann, I/EF-51, The Relevance of Joints in Friction Brake NVH

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Agenda

- Introduction
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- **Summary**

15 Merten Tiedemann, I/EF-51, The Relevance of Joints in Friction Brake NVH

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Summary

- **Brake noise is a challenge throughout the whole brake industry**
  - → System-inherent in friction brakes
- **Most critical in the field: brake squeal (1...20 kHz)**
  - → Dynamic instability of equilibrium solution with pad to disc friction as energy source

Challenges of Brake NVH Development

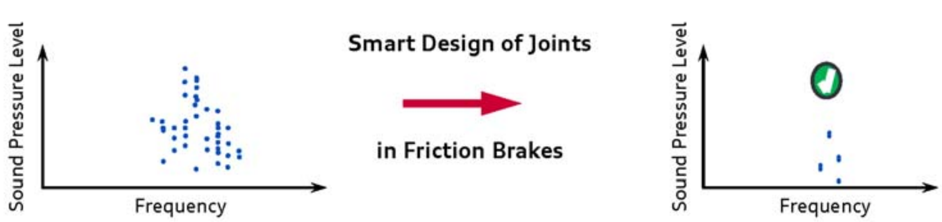
- Many different parts with different properties and individual production variations involved
- Today, no overall predictive description of brake noise possible
- Modern simulations tools, standardized tests and systematic procedures required for successful brake NVH development

Summary

- **Joint design / manipulation** = NVH countermeasure
  - Energy sinks

Challenges

- Design guidelines (for maximum dissipation) missing
- Proper modelling in FE environment difficult
- Predictive calculations not possible



5.3. Submitted Short Talks from the First Evening

Tore Butlin: Dynamic Friction Work at Cambridge (and Bristol)

Dynamic friction work at Cambridge (and Bristol)

Prof Jim Woodhouse
Alessandro Cabboi
Andrew McKay
Tore Butlin
(Kevin Wang)
(Philippe Duffour)

Thibaut Putelat

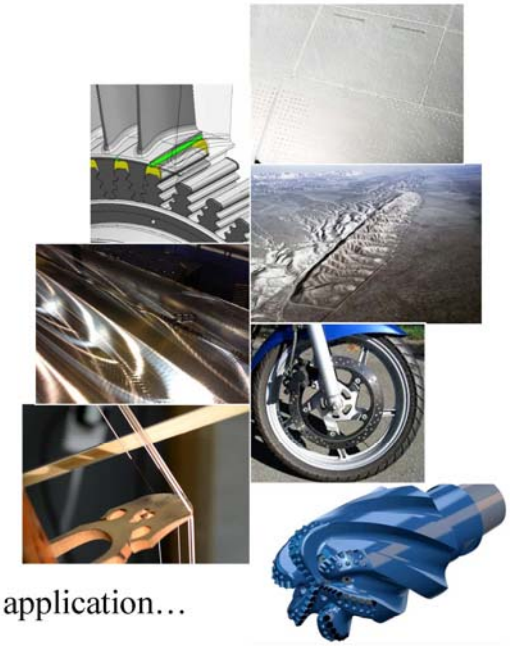
UNIVERSITY OF CAMBRIDGE
Department of Engineering

University of BRISTOL

Friction

Categories

- Frictional damping
  - unintended
  - deliberate
- Stick-slip and position control
- Self-excited vibration:
  - stability thresholds
  - limit cycle prediction
  - transient prediction



Friction model fidelity depends on application…

Starting point

Bowed string instruments (Prof Woodhouse)

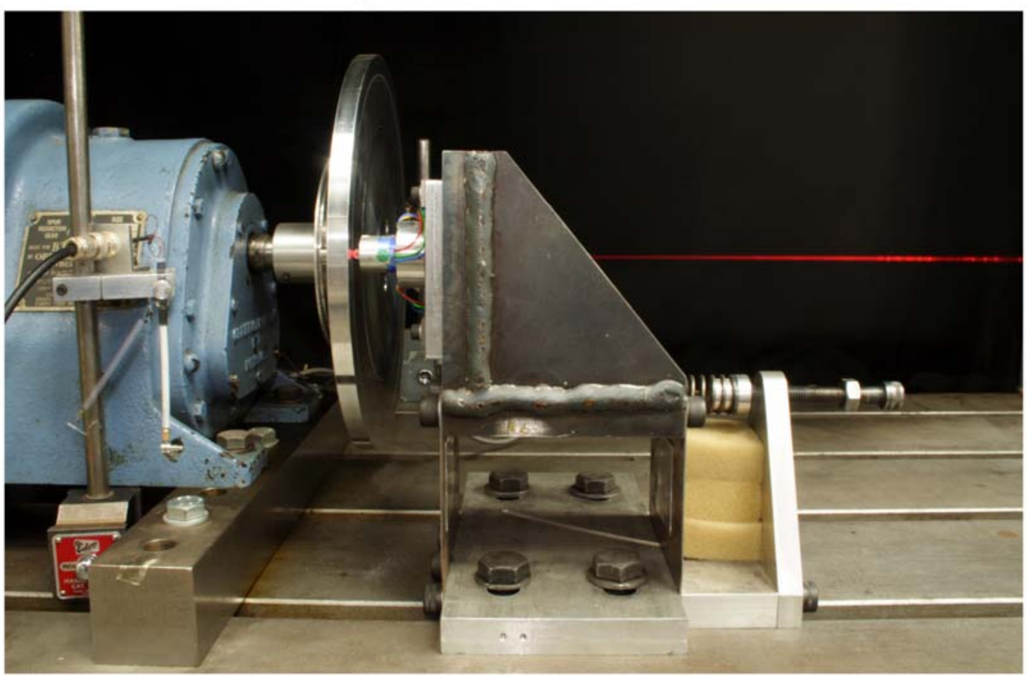
- Transients and limit cycles are important
- Led to a good understanding of friction in that context

Another example of friction-induced sound…

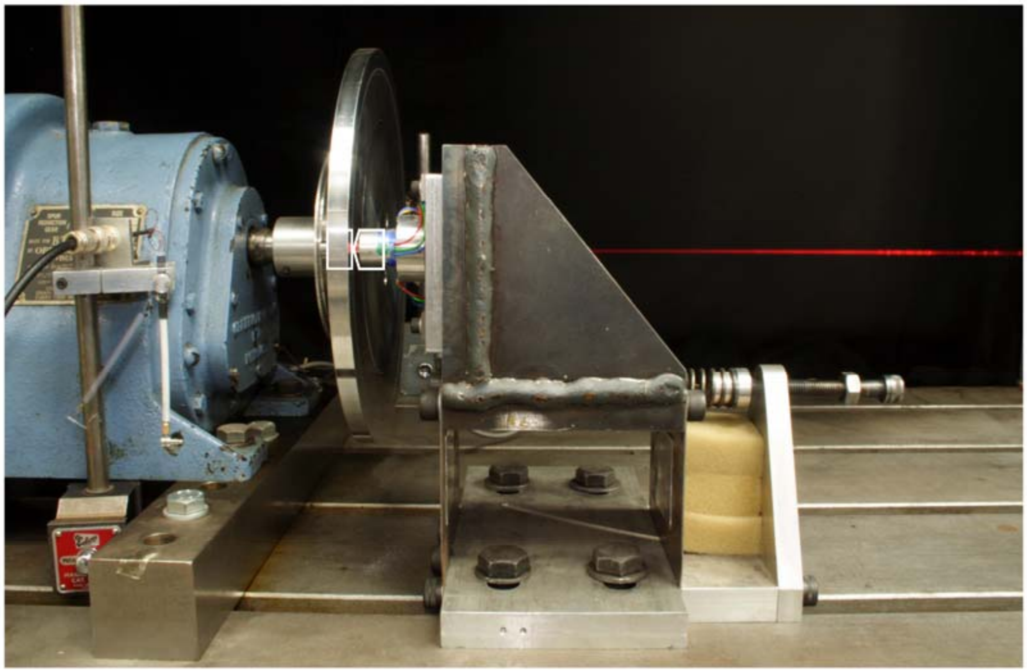
Vehicle brake squeal

- Only care about whether or not it happens
- Linearised stability threshold

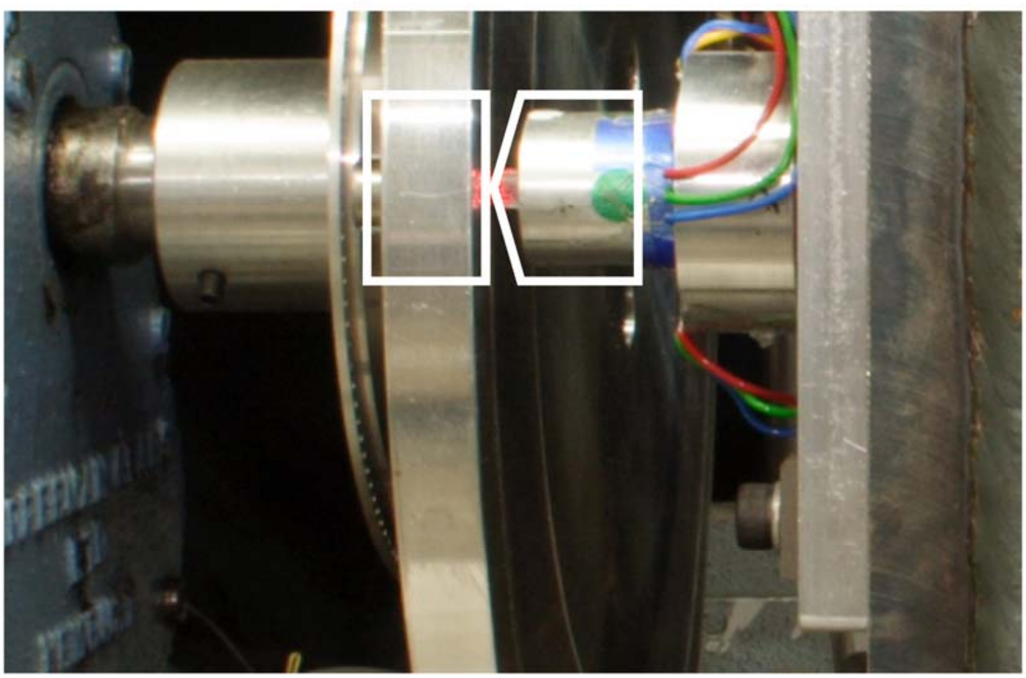
Vehicle brake squeal



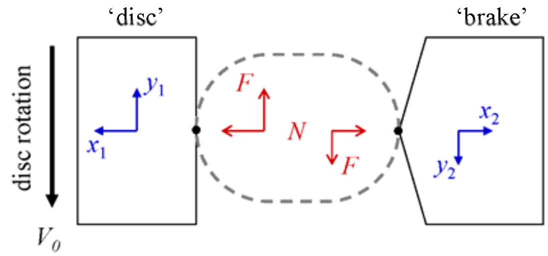
Vehicle brake squeal



Vehicle brake squeal



Vehicle brake squeal



Lots of models... $F' = \alpha N'$ (Coulomb's law)

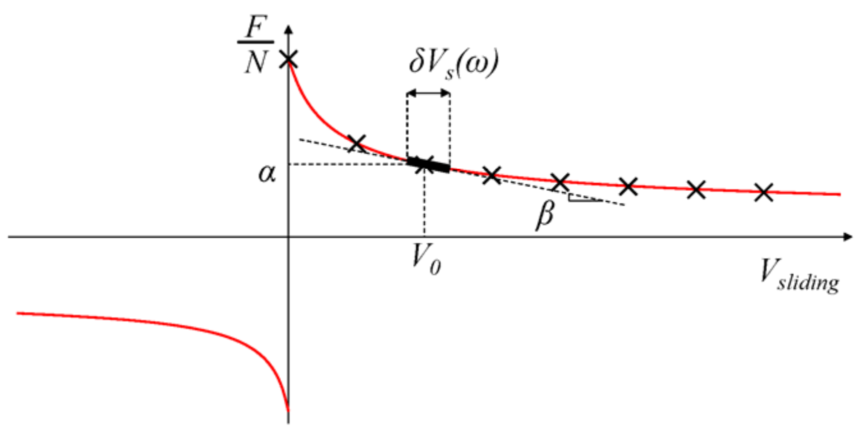
$F' = \alpha N' + \beta V'$ (Velocity dependent law)

⋮

All models reduce to this form in context of linear theory
α and β are Transfer Functions of dynamic friction

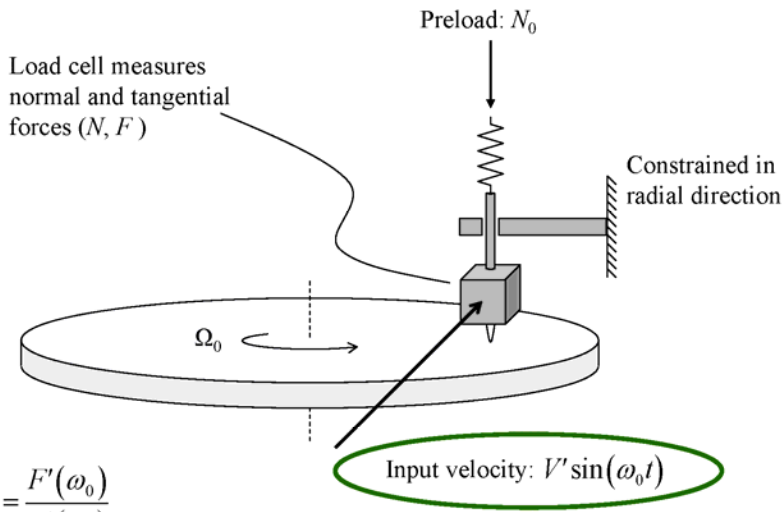
Measuring β

Classic pin-on-disc

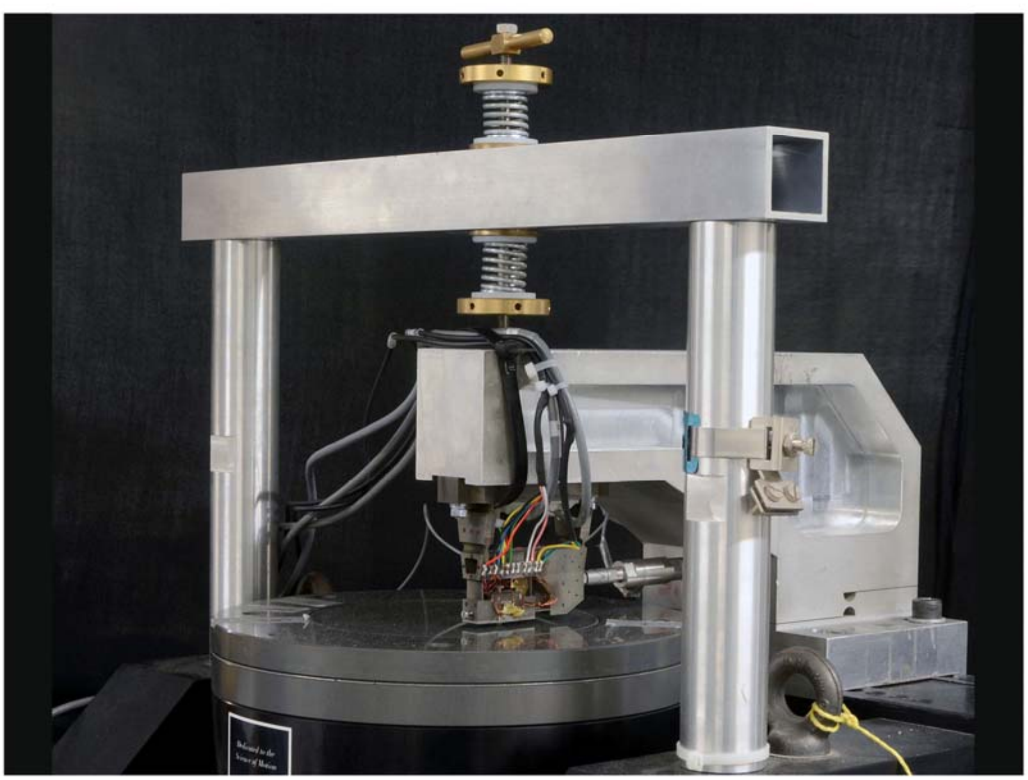


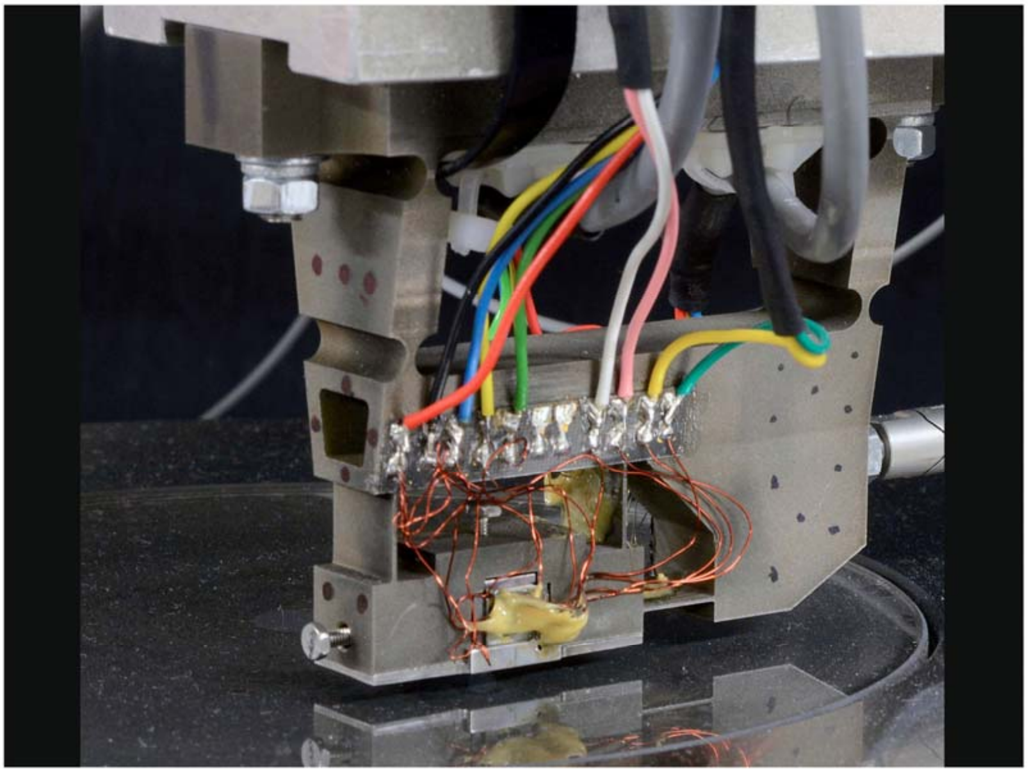
Measuring β

Cambridge friction rig



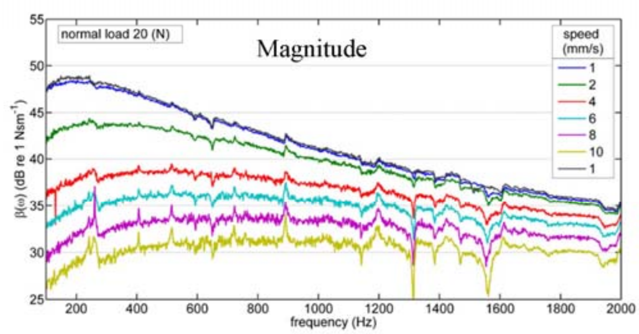
$$\beta(\omega_0) = \frac{F'(\omega_0)}{V'(\omega_0)}$$

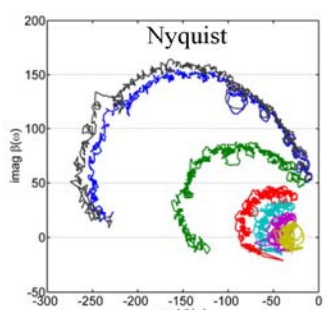


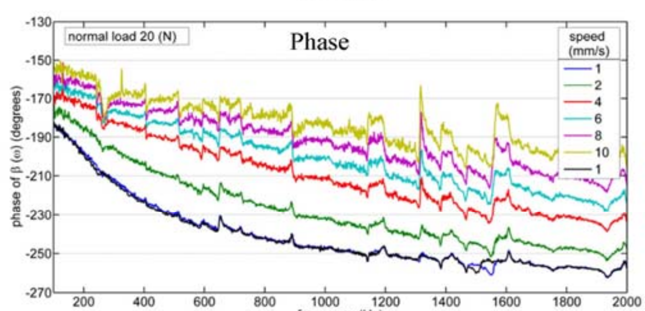


Example results for β (nylon on glass)

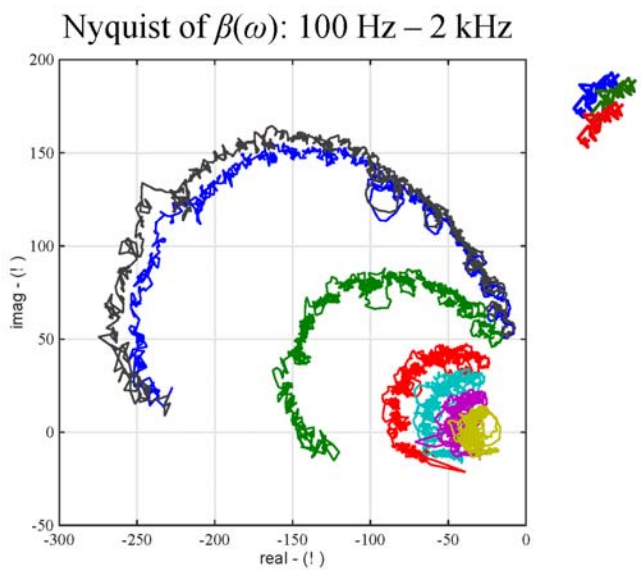
mm/s	10 N	20 N	30 N	40 N	50 N
1					
2					
4					
...					
10					
20		data matrix			
...					
40					



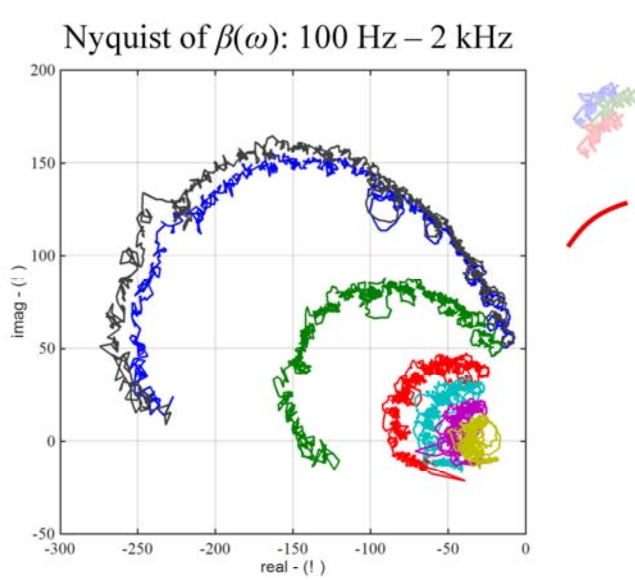




Friction model fit?



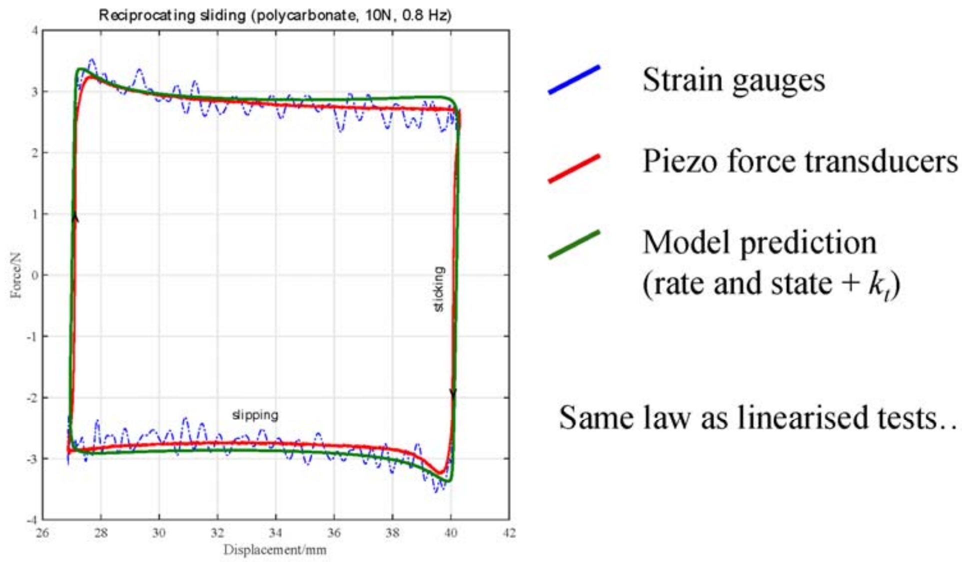
Friction model fit?



mm/s	10 N	20 N	30 N	40 N	50 N
1					
2					
4					
...					
10					
20					
...					
40					

data matrix

Nonlinear tests: reciprocating sliding



Same law as linearised tests…

Conclusions

- The ‘right’ friction model depends on the question
- For predicting stability thresholds: need dynamic friction transfer functions
- Need to carry out new measurement
- Results are surprisingly repeatable
- For our materials: a modified rate and state law gives excellent agreement and prediction of trends

Friction Workshop:

"Friction as it is relevant to structural vibration"

at Cambridge University Engineering Department

Friday 11th December: 9:30am to 4:30pm
Refreshments and lunch included
Please register: Prof Jim Woodhouse (jw12@cam.ac.uk)

This workshop will be dedicated to the memory of Professor Ken Johnson (1925–2015)

Engineering Nonlinearity
Dynamic design tools for engineering structures

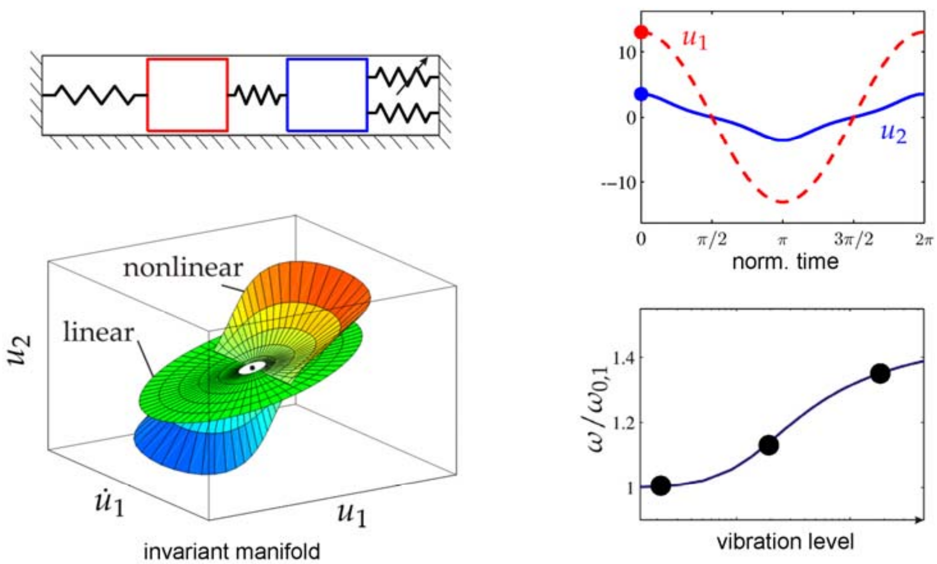
Leibniz Universität Hannover

Institut für Dynamik und Schwingungen IDS
Leibniz Universität Hannover

Nonlinear Modal Analysis and Modal Reduction of Jointed Structures

Malte Krack
Leibniz Universität Hannover
Institute of Dynamics and Vibration Research
October 2015

Nonlinear modes – central ideas



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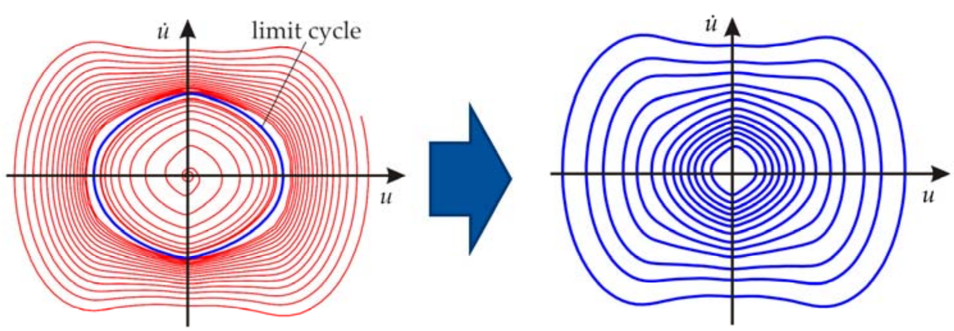
Nonlinear modal analysis of jointed structures

Regime of interest: periodic vibrations, sustained excitation

'Make the motions periodic'

$$M\ddot{u}(t) + f(u(t), \dot{u}(t)) = 0 \qquad M\ddot{u}(t) - \delta M\dot{u}(t) + f(u(t), \dot{u}(t)) = 0$$



Reduction to isolated nonlinear mode

- coordinate transform

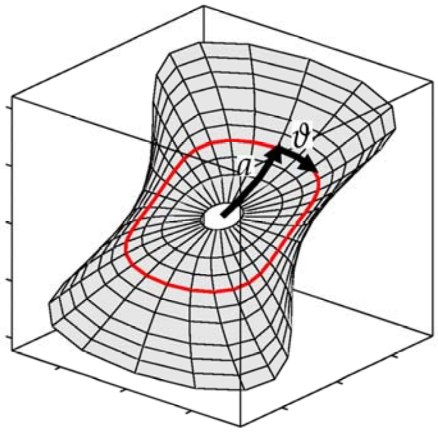
$$\boldsymbol{u}(t) = \boldsymbol{U}\left(a(t), \vartheta(t)\right), \dot{\boldsymbol{u}}(t) = \boldsymbol{V}\left(a(t), \vartheta(t)\right)$$

- averaging around periodic motion

unsteady $\begin{bmatrix} \dot{a} \\ \Delta\dot{\vartheta} \end{bmatrix} = \boldsymbol{h}(a, \Delta\vartheta)$

steady-state $\boldsymbol{0} = \boldsymbol{h}(a, \Delta\vartheta)$

} modal single-DOF-oscillator

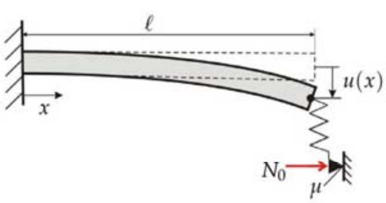


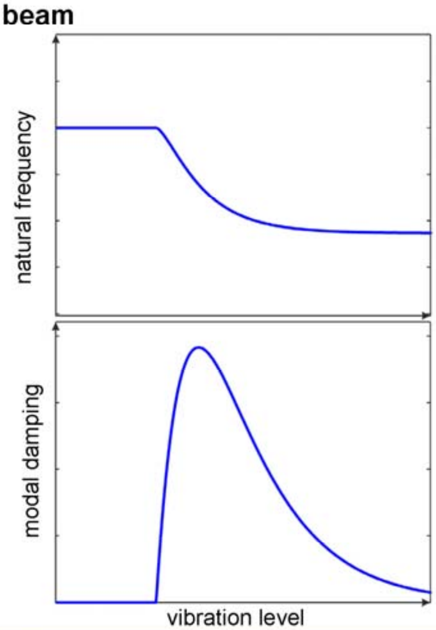
Inherent parameter space

a. external forcing
b. linear modal damping
c. preload*

*in the case of the elastic Coulomb and unilateral-elastic contact laws (owing to their scale invariance)

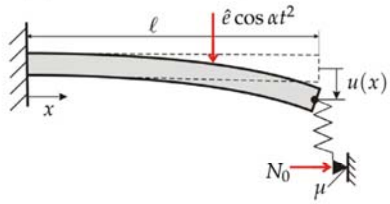
Application example: Friction-damped beam

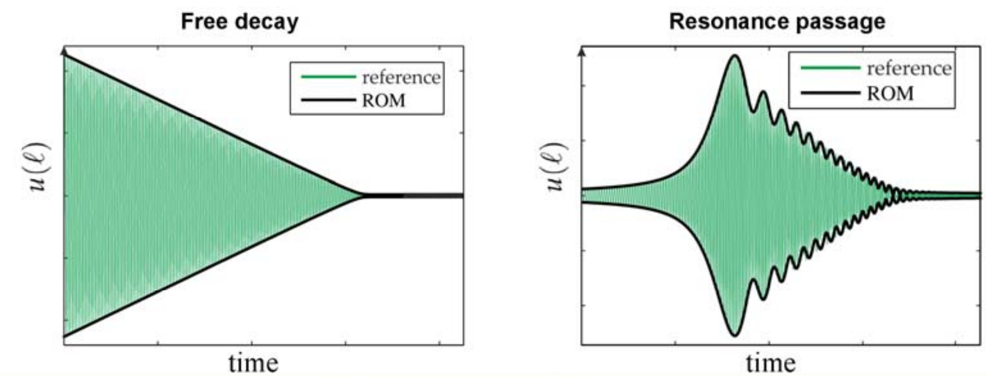




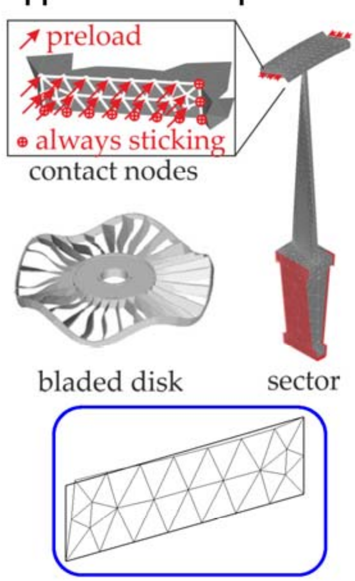
Modal characteristics
first bending mode

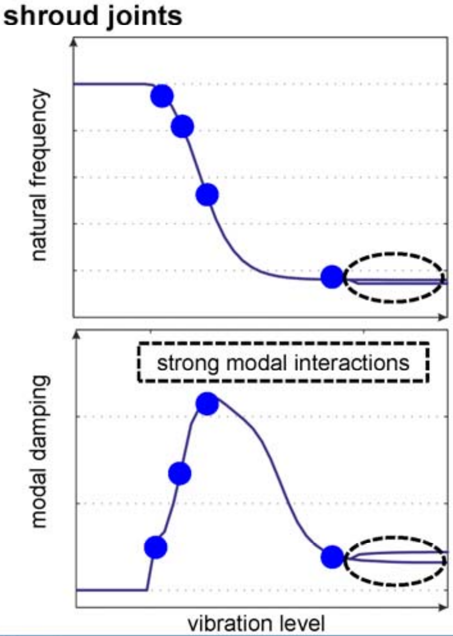
Application example: Friction-damped beam



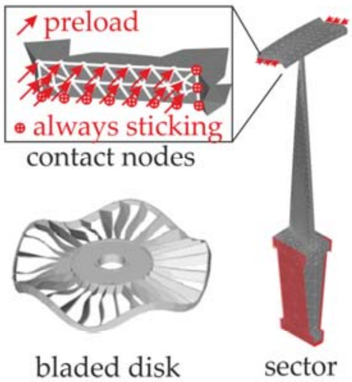


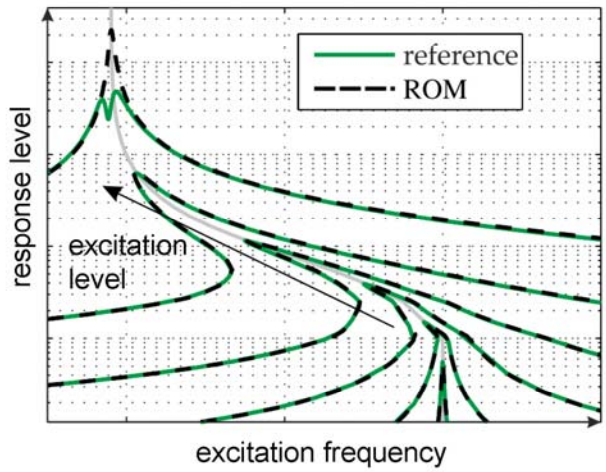
Application example: Bladed disk with shroud joints





Application example: Bladed disk with shroud joints





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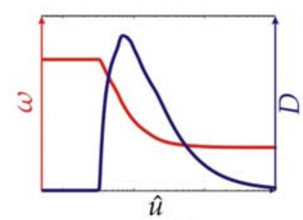
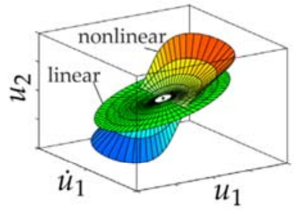
Conclusions

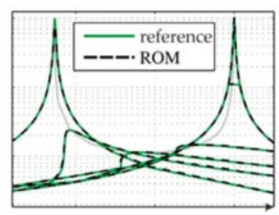
- direct analysis of natural frequencies and effective damping as function of vibration level



- detection of modal interactions and localization phenomena



- reduction to single modal oscillator



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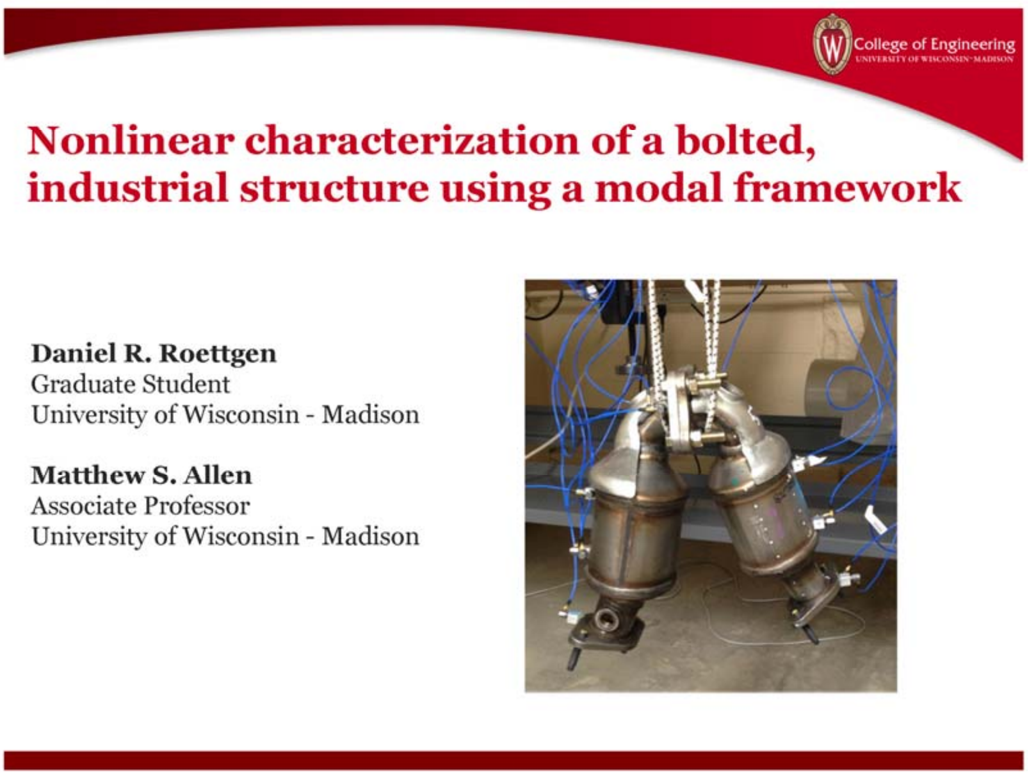
Krack, M.; Panning-von Scheidt, L.; Wallaschek, J.: **A Method for Nonlinear Modal Analysis and Synthesis: Application to Harmonically Forced and Self-Excited Mechanical Systems**. Journal of Sound and Vibration 332 (25): 6798–6814, 2013.

Krack, M.; Panning-von Scheidt, L.; Wallaschek, J.; Siewert, C.: Hartung, A.: **Reduced Order Modeling Based on Complex Nonlinear Modal Analysis and its Application to Bladed Disks With Shroud Contact**. J. Eng. Gas Turbines Power 135 (10), 102502–102509, 2013.

Dan Roettgen: Nonlinear Characterization of a Bolted Industrial Structure Using a Modal Framework



Motivation

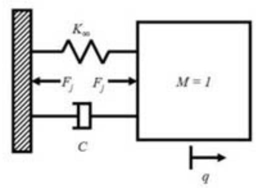
If we have many joints, it becomes cumbersome to identify the parameters of each joint separately!

$$\left\{\begin{matrix} F_S^1, K_T^1, \chi^1, \beta^1 \\ F_S^2, K_T^2, \chi^2, \beta^2 \\ F_S^3, K_T^3, \chi^3, \beta^3 \\ \vdots \end{matrix}\right\}$$

Experimental evidence has shown that many jointed structures can be tested and represented with uncoupled as weakly nonlinear modes.[1][2]

[1] D. R. Roettgen and M. S. Allen, "Nonlinear characterization of a bolted, industrial structure using a modal framework", In submission at Mechanical Systems and Signal Processing.
[2] R. L. Mayes, B. P. Pacini and D. R. Roettgen, "A Modal Model to Simulate Typical Structural Dynamic Nonlinearity", Submitted to the International Modal Analysis Conference XXXIV, Orlando, FL USA, 2016.

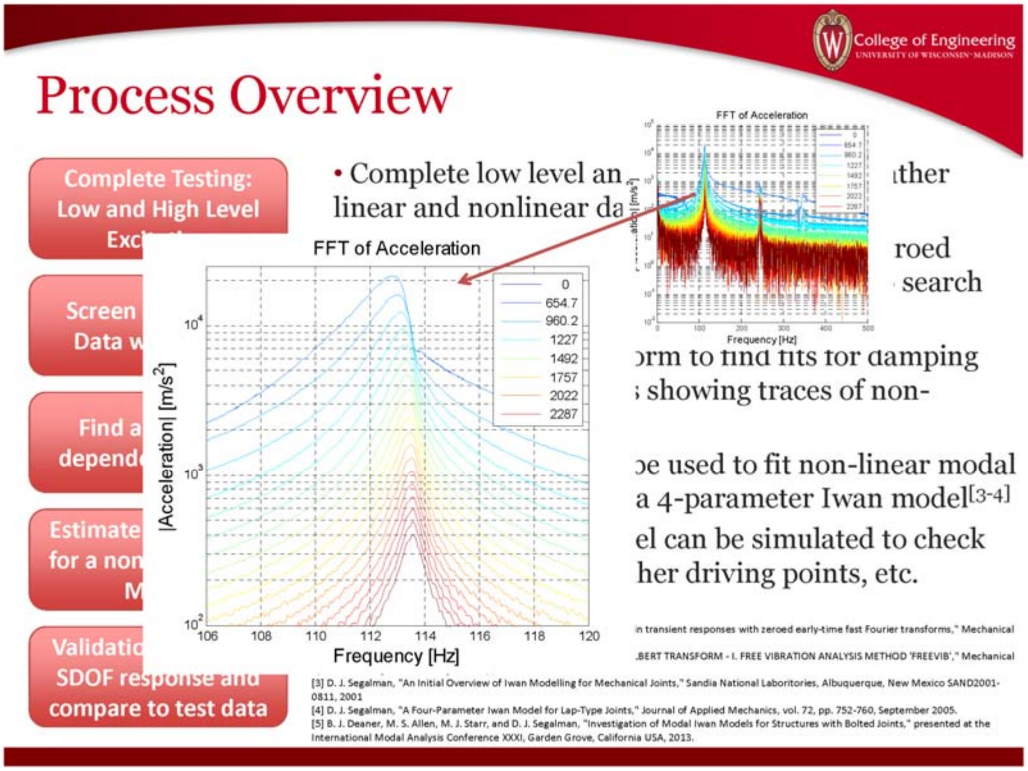
We now have quite a bit of experimental evidence that the modal model is appropriate in many cases.



Assumes that the linear modes are preserved, no coupling between modes!

Background

- Tested using an impact hammer at various driving points and force levels.
- Industrial torques and seals used in the assembly of the system.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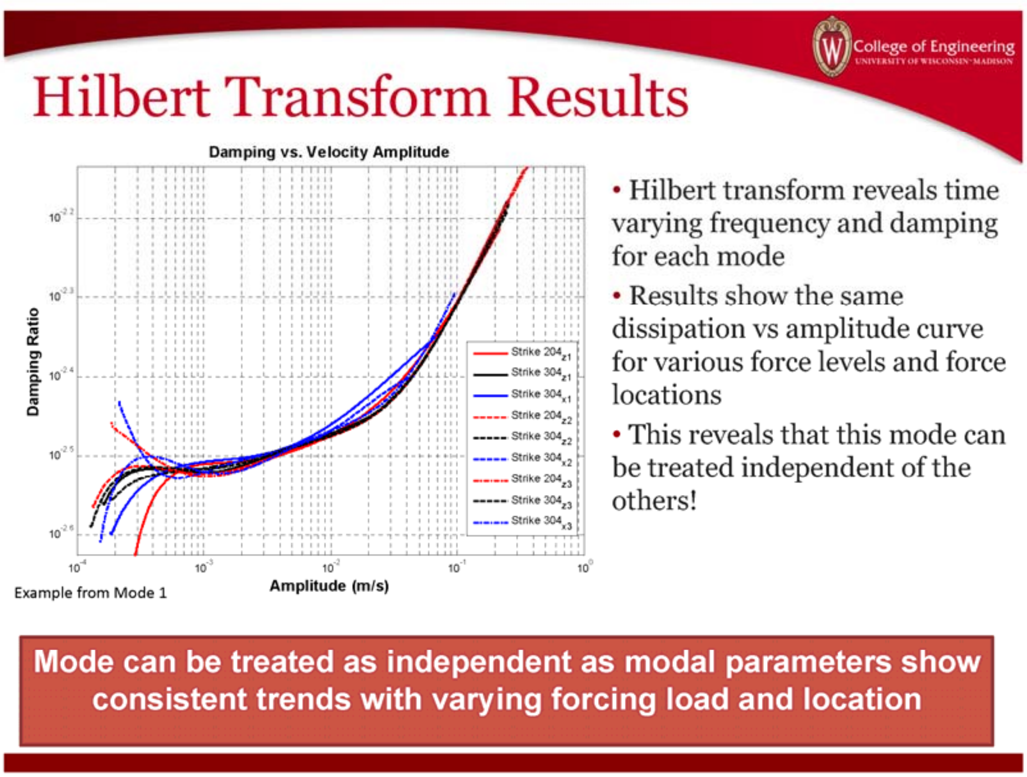
Process Overview
Complete Testing: Low and High Level
• Complete low level an
ther
linear and nonlinear da
roed
search
Screen
Data w
Find a
depend
Estimate
for a non
Validatio
SDOF response and
compare to test data
FFT of Acceleration
|Acceleration| [m/s^2]
Frequency [Hz]
0
654.7
960.2
1227
1492
1757
2022
2287
orm to find fits for damping
showing traces of non-
be used to fit non-linear modal
a 4-parameter Iwan model[3-4]
el can be simulated to check
her driving points, etc.
in transient responses with zeroed early-time fast Fourier transforms," Mechanical
BERT TRANSFORM - I. FREE VIBRATION ANALYSIS METHOD 'FREEVIB'," Mechanical
[3] D. J. Segalman, "An Initial Overview of Iwan Modelling for Mechanical Joints," Sandia National Laboritories, Albuquerque, New Mexico SAND2001-0811, 2001
[4] D. J. Segalman, "A Four-Parameter Iwan Model for Lap-Type Joints," Journal of Applied Mechanics, vol. 72, pp. 752-760, September 2005.
[5] B. J. Deaner, M. S. Allen, M. J. Starr, and D. J. Segalman, "Investigation of Modal Iwan Models for Structures with Bolted Joints," presented at the International Modal Analysis Conference XXXI, Garden Grove, California USA, 2013.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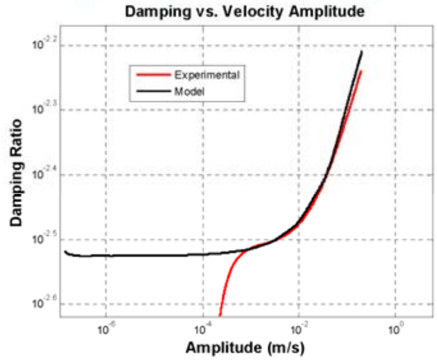
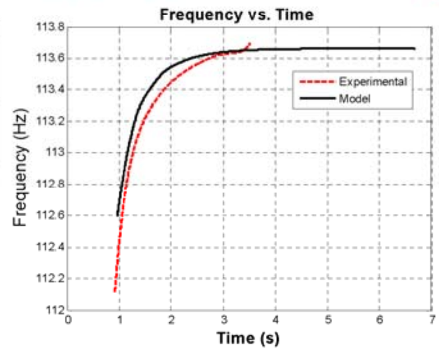
UNIVERSITY OF WISCONSIN-MADISON
Hilbert Transform Results
Damping vs. Velocity Amplitude
Damping Ratio
Amplitude (m/s)
Strike 204z1
Strike 304z1
Strike 304x1
Strike 204z2
Strike 304z2
Strike 304x2
Strike 204z3
Strike 304z3
Strike 304x3
Example from Mode 1
• Hilbert transform reveals time varying frequency and damping for each mode
• Results show the same dissipation vs amplitude curve for various force levels and force locations
• This reveals that this mode can be treated independent of the others!
Mode can be treated as independent as modal parameters show consistent trends with varying forcing load and location

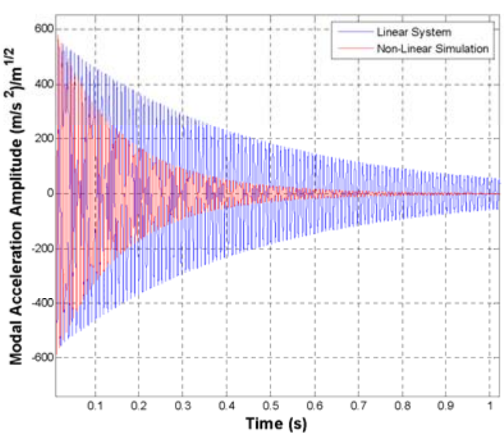
Iwan Simulation Results





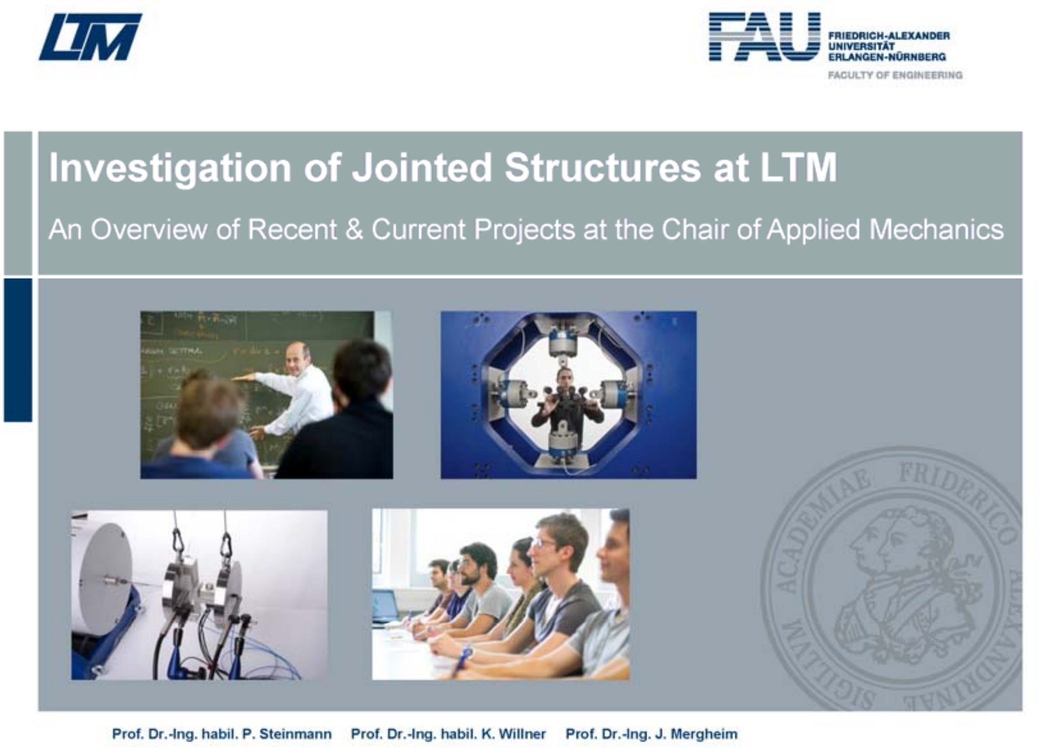
- Using the a power-law relationship from the damping versus amplitude we can begin to define parameters for the 4-Parameter Iwan model.
- χ and the intercept are easily obtained from the Hilbert results
- This model is only presumed to be accurate for micro-slip conditions
- These parameters can be used in a Newmark Integration algorithm to simulate the response and to compare to experimental data

Conclusions

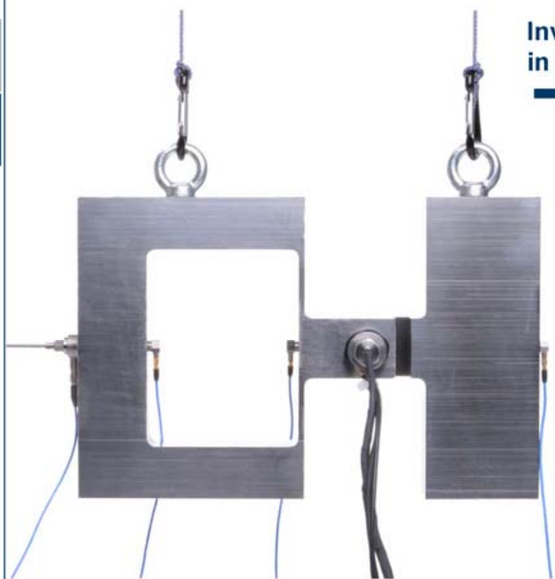


- Using a modal model approach is a large improvement over standard industry practice (using low level linear test data and extrapolating up to high load levels)
- Many practical jointed structures are weakly nonlinear in nature which allows us to treat the modes as uncoupled
- Damping and frequency parameters fit onto the same curve independent of driving point location and force level

Dominik Süß: Investigation of Jointed Structures at LTM: An Overview of Recent and Current Projects at the Chair of Applied Mechanics



Motivation



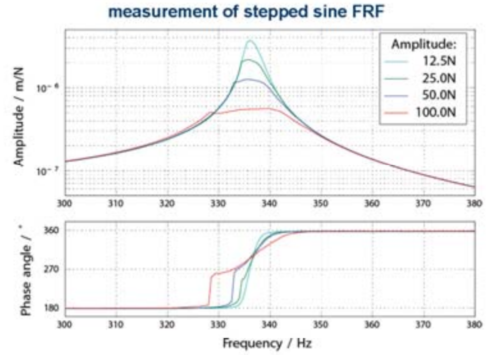
Investigation of Jointed Structures in the Frequency Domain
➡ Harmonic Balance Method

Prof. Dr.-Ing. habil. P. Steinmann Prof. Dr.-Ing. habil. K. Willner Prof. Dr.-Ing. J. Mergheim 2

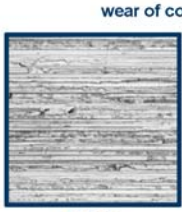
Experimental Investigation of Friction Resonator

Dominik Süß and Kai Willner

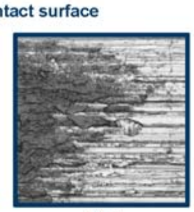
measurement of stepped sine FRF



wear of contact surface

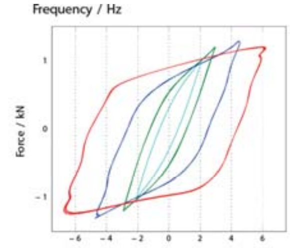


before



after

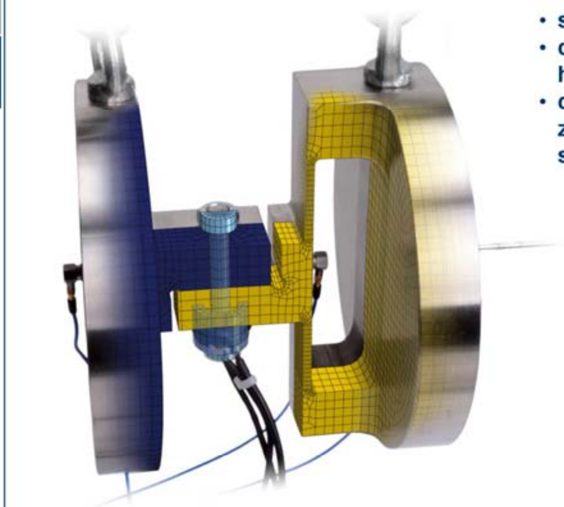
corresponding friction hystereses:



Prof. Dr.-Ing. habil. P. Steinmann Prof. Dr.-Ing. habil. K. Willner Prof. Dr.-Ing. J. Mergheim 3

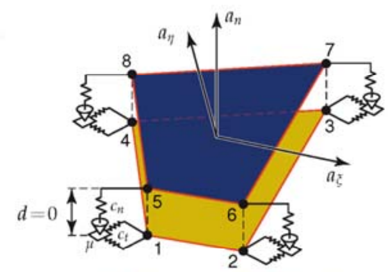
Numerical Investigation of Friction Resonator using Zero-Thickness-Elements

Dominik Süß and Kai Willner



Academic friction oscillator with overlaid finite element mesh

- symmetry boundary conditions
- discretization of linear parts using hexahedral elements
- discretization of contact plane using zero-thickness elements, see [SUESS and WILLNER, 2015]



ZT8-element, see [MAYER and GAUL, 2007], [GEISLER, 2010]:

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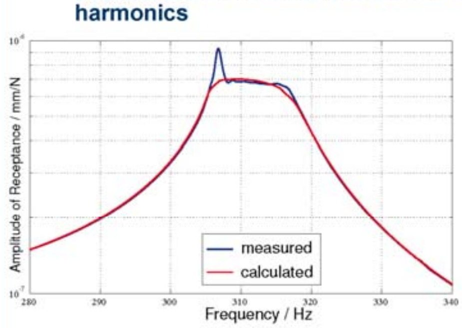
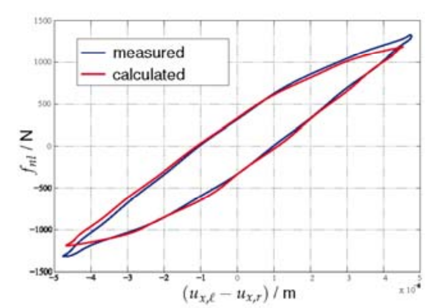
Numerical Investigation of Friction Resonator using the Multi Harmonic Balance Method

Dominik Süß and Kai Willner

- comparison of measured and calculated FRF and friction hysteresis
- corresponding stationary behavior in time domain for an excitation frequency of 306Hz
- physically reasonable hysteresis loops only possible because of higher harmonics





FE mesh:	calculational parameters:	contact parameters:
19526 nodes	$\lvert\hat{F}_{e,(1)}\rvert = 100$ N	$c_n = 1.00 \cdot 10^6$ N/mm^3
15454 Hex8 elements	$n_h = 11$	$c_t = 5.51 \cdot 10^2$ N/mm^3
58 ZT elements		$\mu = 0.66$

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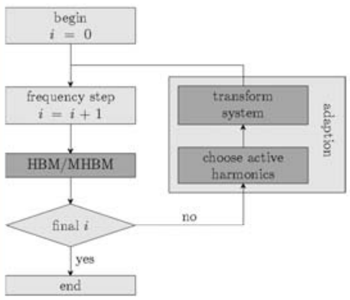
Numerical Investigation of Systems with Friction using an Adaptive Harmonic Balance Method

Dominik Süß and Kai Willner

Adaptive selection of harmonics during calculation



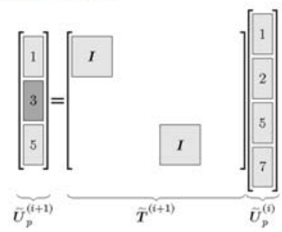
No additional calculation needed / computation with low effort needed, depending on algorithm used

➡ **Procedure using as less harmonics as possible and as many harmonics as needed**

Choosing harmonics via one of the following criteria:

- **Distortion factor directly applied to estimation of response displacement harmonics**
- **Criterion based on partial derivatives of nonlinear forces → sensitivity of harmonics**
- **Tangent-predictor for estimating response harmonics (similar to [Grolet and Thouverez, 2012])**

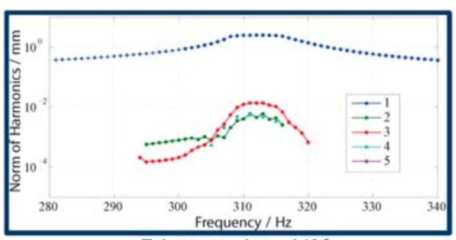
Transformation of system equations via simple transformation matrix



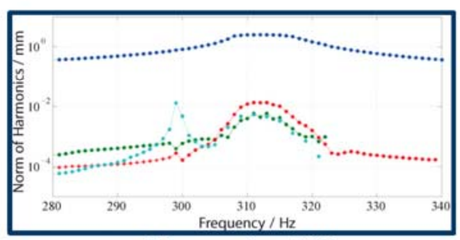
Numerical Investigation of Systems with Friction using an Adaptive Harmonic Balance Method

Dominik Süß and Kai Willner

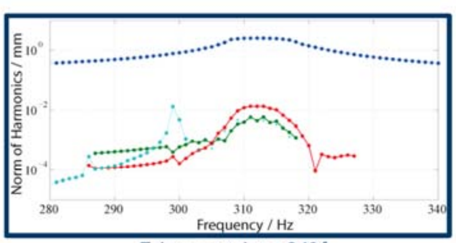
Adaptive selection of harmonics during calculation, e.g. using distortion factor of displacement harmonics



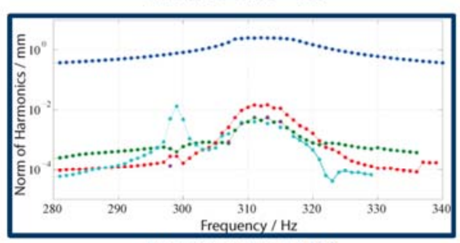
Tolerance value = $4 \cdot 10^{-5}$



Tolerance value = $3 \cdot 10^{-5}$



Tolerance value = $2 \cdot 10^{-5}$



Tolerance value = $1 \cdot 10^{-5}$

Measurement (and calculation) of beam structure

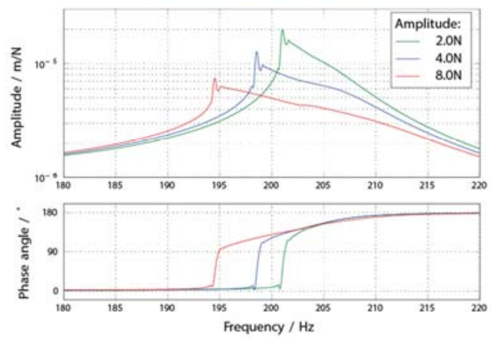
Dominik Süß and Kai Willner

beam equal to [Brake et. al., 2014]

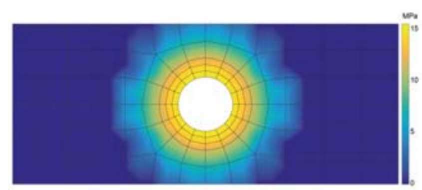
modified beam with only one bolt

tangential and normal investigation

variation of excitation amplitude:

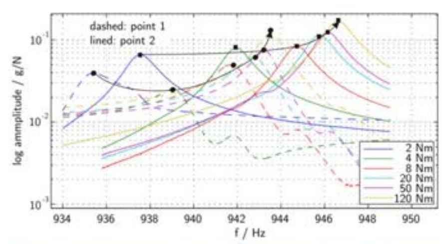


FE-simulation of contact normal stress



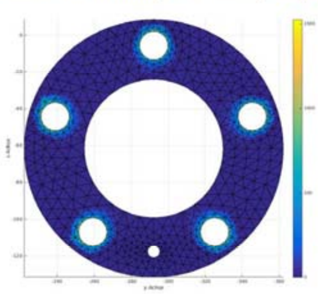
Experimental and Numerical Investigation of Braking-Disk-Joints

Dominik Süß, Martin Jerschl and Kai Willner



Moving resonances due to changes of bolting torque

- Bolted joint connection of braking disk and hub
- Experiments using excitation normal and tangential to contact plane



FE-simulation of contact normal stress

Dynamic systems with rotating geometry under non-linear oscillation impact

Tim Weidauer and Kai Willner

- Main topic:
  - Systems under self-/external excitation (e.g. frictional contact)
  - Instability of vibrational behavior
  - Application of ALE-FEM-Algorithm
- Application example:
  - Brake disc ➡ brake squeal
  - Gantry in CT apparatus ➡ noise

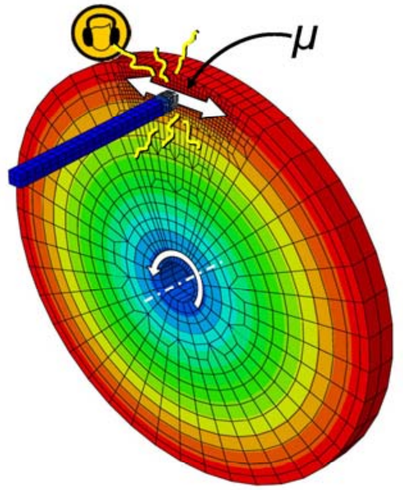


Fig. 1: Pin-on-disc model

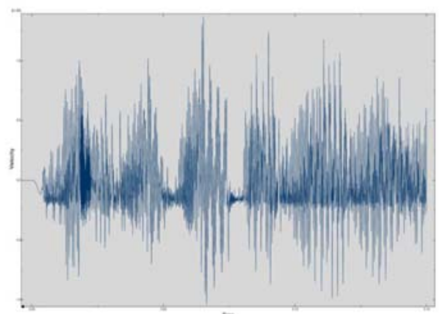
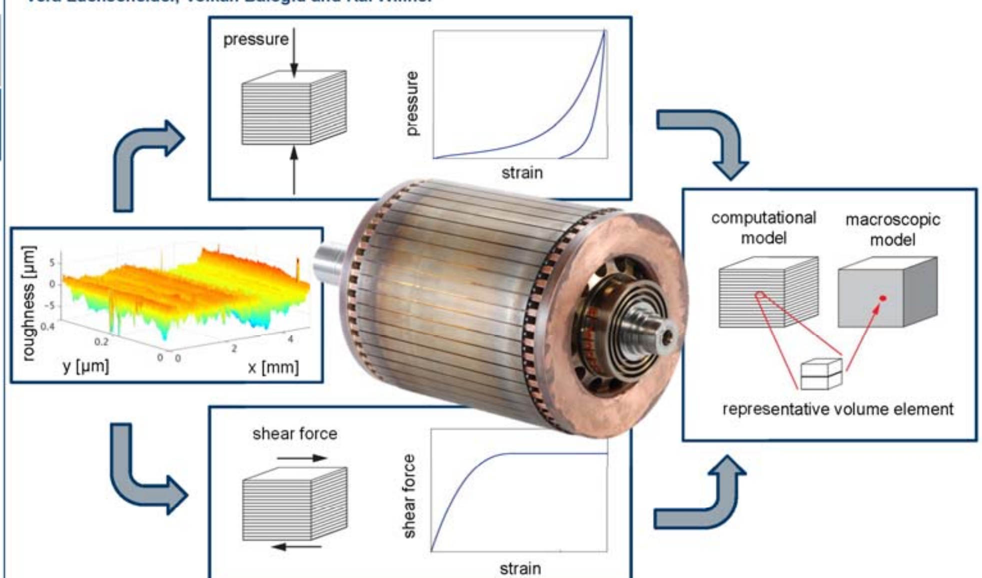
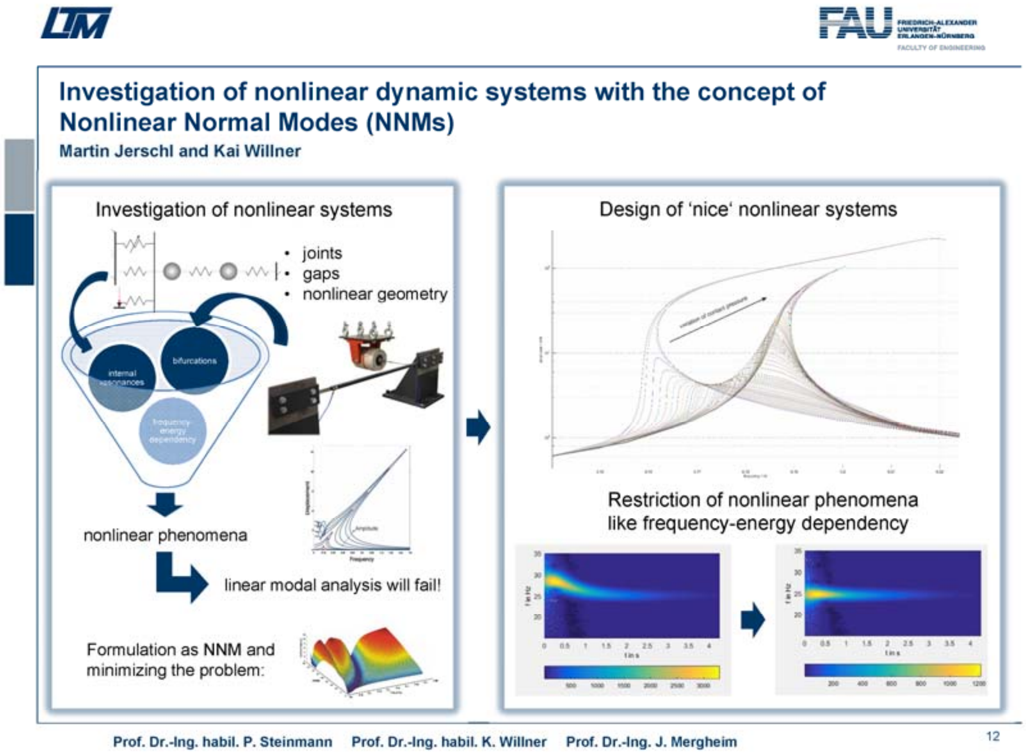


Fig. 2: Pin tip velocity (tangential direction)

Identification of a representative model for a layered structure with contacts

Vera Luchscheider, Volkan Baloglu and Kai Willner




Investigation of nonlinear dynamic systems with the concept of Nonlinear Normal Modes (NNMs)
Martin Jerschl and Kai Willner
Investigation of nonlinear systems
joints
gaps
nonlinear geometry
internal resonances
bifurcations
frequency-energy dependency
nonlinear phenomena
linear modal analysis will fail!
Formulation as NNM and minimizing the problem:
Design of 'nice' nonlinear systems
Restriction of nonlinear phenomena like frequency-energy dependency
Prof. Dr.-Ing. habil. P. Steinmann
Prof. Dr.-Ing. habil. K. Willner
Prof. Dr.-Ing. J. Mergheim
12

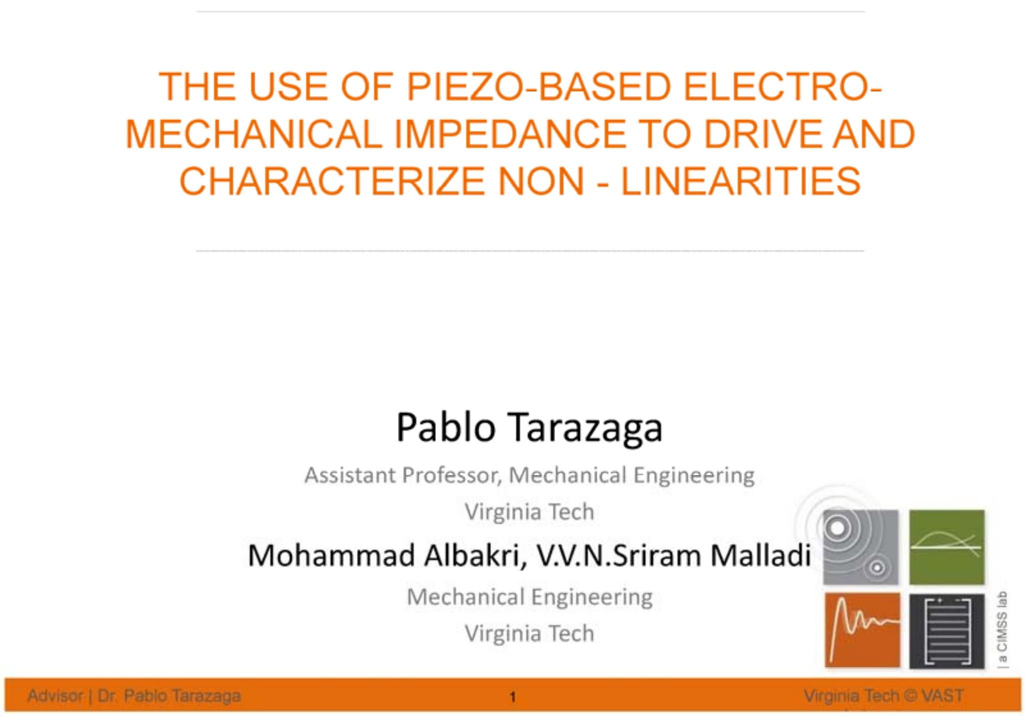
4

Summary and Outlook

Research topics:		Addressed Joint Challenges (WS3):
• Investigation of friction resonator	➡	2 & 1
• Investigation of beam structure	➡	2
• Investigation of braking disk joints		
• Multiscale modeling of layered structure	➡	6
• Half space modelling of contact interfaces	➡	10
• Experimental application of NNMs on jointed systems		
• Two proposals for DFG projects in the framework of uncertainty modeling and damping	➡	8

Thank you for your attention!

Pablo Tarazaga: The Use of Piezo-Based Electro-Mechanical Impedance to Drive and Characterize Non-Linearities



Outline

- Introduction and motivation
- High-voltage impedance measurements
- Nonlinear phenomena classification based on impedance measurements
- High frequency damage-induced nonlinearities
- Conclusions

Impedance measurements have been very attractive to detect the health of a structure.

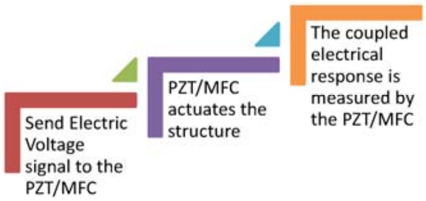
SHM

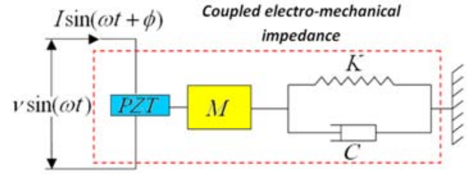
Requirements

- Nondestructive
- Nonintrusive
- Sensitive to incipient damages
- Applicable to complex structures
- Low cost

Impedance-based SHM

Impedance measurements study the behavior of the structure through electrical parameters.





$$Y(\omega) = i\omega \frac{wl}{h}\left[\bar{\varepsilon}_{33}^{T} - d_{31}^{2} + \frac{Z_{pzt}}{Z_{pzt}+Z_{St}} d_{31}^{2}\bar{Y}_{11}^{E}\left(\frac{\tan(kl)}{kl}\right)\right]$$

Electrical impedance of the piezoelectric transducer depends on the mechanical impedance of the host structure

Electrical Admittance (We measure this)

Coupled Mechanical Impedance

Structural Impedance (We are interested in this)

| a CIMSS lab

Liang, C., F. P. Sun, and C. A. Rogers. "Coupled electro-mechanical analysis of adaptive material systems—determination of the actuator power consumption and system energy transfer." *Journal of Intelligent Material Systems and Structures* 5.1 (1994): 12-20.

CAN THIS BE USED TO DETECT NONLINEARITY IN STRUCTURES?

- ***Objectives:***

1- Determine the possibility of using PZT excitation and EMI measurements to detect structural nonlinearities.

2- Identify nonlinear phenomena based on impedance measurements

3- Study the effects of damage-induced nonlinearities on high-frequency dynamic response

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Determine the possibility of using PZT excitation and EMI measurements to detect structural nonlinearities

- ***Approach:***

  1- Develop a high-voltage EMI measurement setup

  2- Compare the impedance signature of linear and nonlinear structures at different levels of excitations

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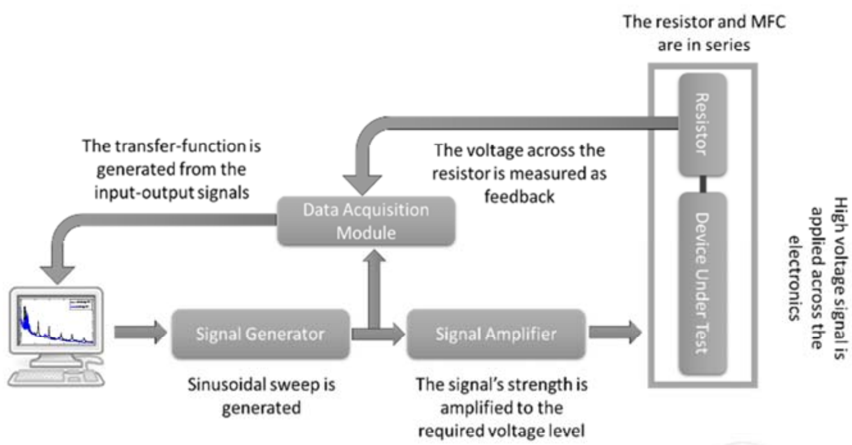
High-voltage EMI setup



This setup works from a frequency range of 10 Hz to 400kHz and amplitude up to 1000V.

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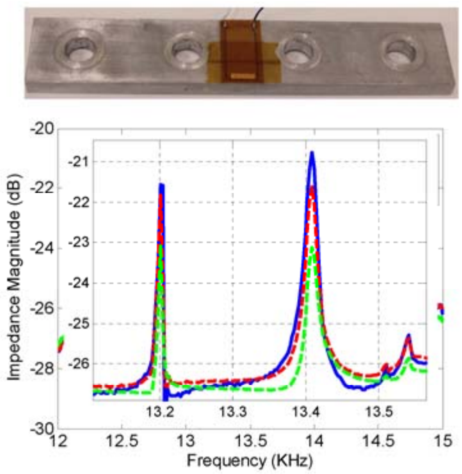
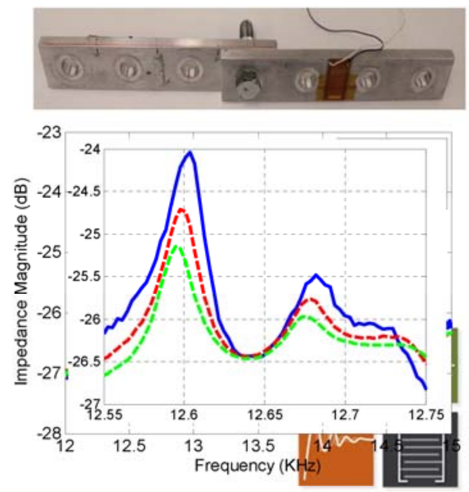
Initial experiments show that piezoceramic excitation can potentially be used to excite non-linearities in joints.

Linear Structure | **Lap Joint**





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Identify nonlinear phenomena based on impedance measurements

- ***Approach:***

  1- Develop a <u>simple</u> EMI model for nonlinear structures

  2- Simulate impedance signature for different types of structural nonlinearities

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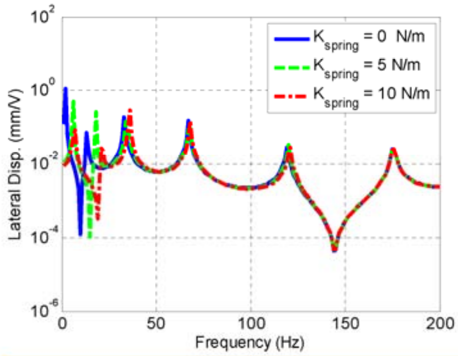
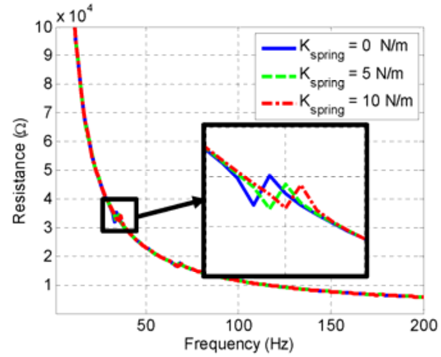
Start by looking at the linear response of a cantilever beam

- **A cantilever-beam with a linear spring is modeled using FEM**
- **Spring stiffness is varied and impedance signature and FRF are calculated**





- **An impedance peak sensitive to changes in spring stiffness is identified**

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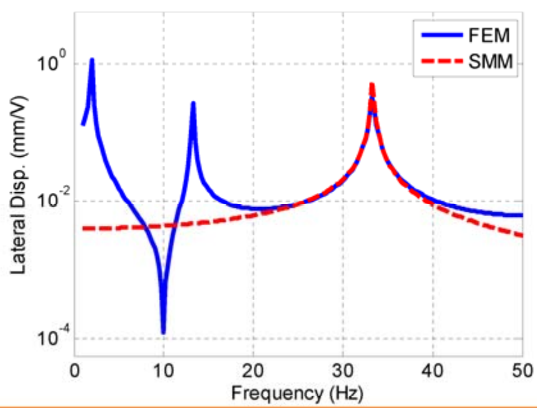
Develop a single mode model that could represent that response

- **The corresponding region in the reciprocity FRF is used to fit a single-mode model**

$$m_r\ddot{v}(t) + 2\zeta_r\omega_n\dot{v}(t) + k_r v(t) = V(t)$$



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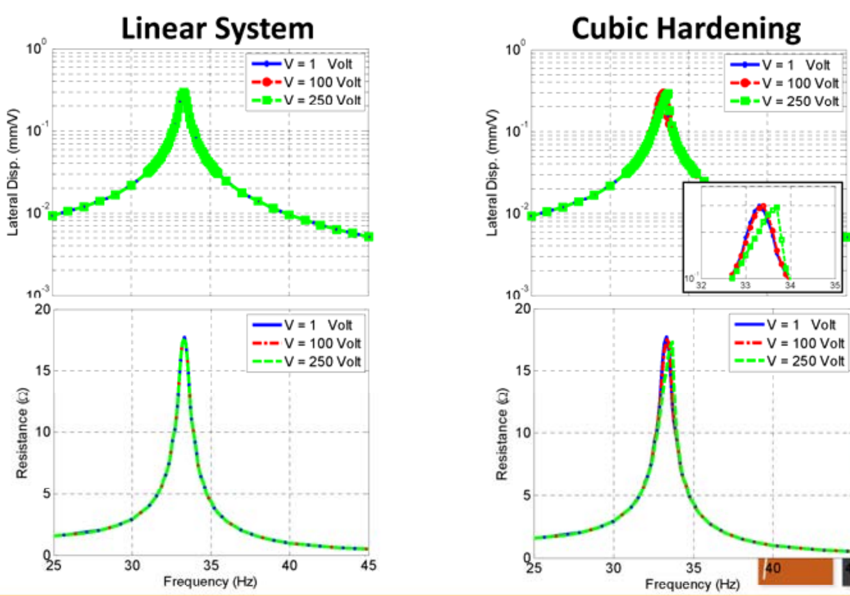
Develop a model to see the effects of non linearities on the Electromechanical Impedance

- Nonlinearities are introduced to the single mode model in the form of:
  - Nonlinear (cubic) stiffness
  - Nonlinear (cubic) damping

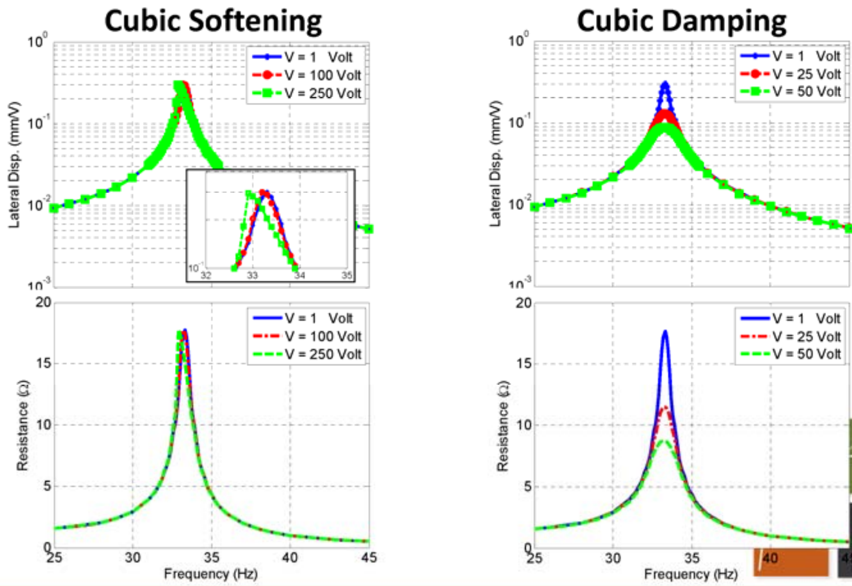
$$m_r \ddot{v} + 2\zeta_r \omega_n \dot{v} + c_c \dot{v}^3 + k_r v + k_c v^3 = V(t)$$

- Mechanical impedance FRFs are calculated for the structures with nonlinearities
- Single-DoF impedance model is used to calculate the corresponding electrical impedance

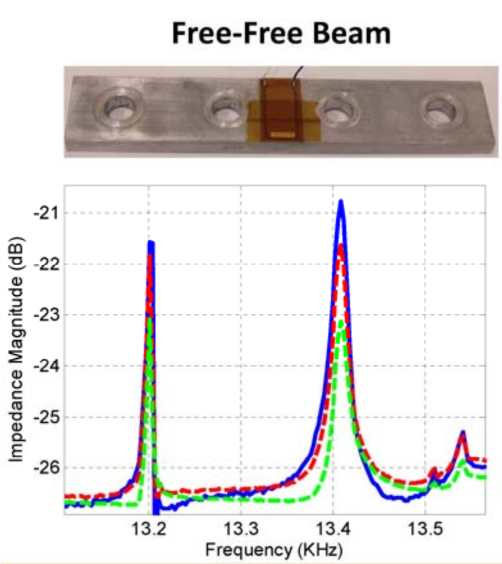
Effects of Nonlinearities on EMI are observed in this simplified model

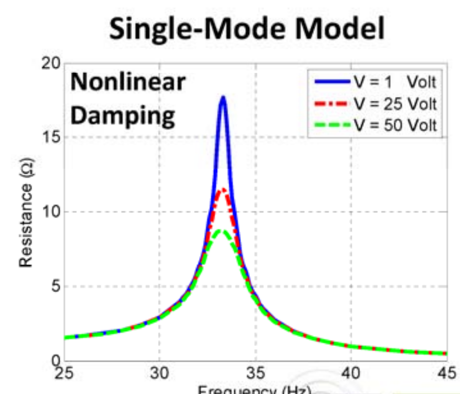


Effects of Nonlinearities on EMI are observed in this simplified model



Experimental results are then revisited in light of the simplified model



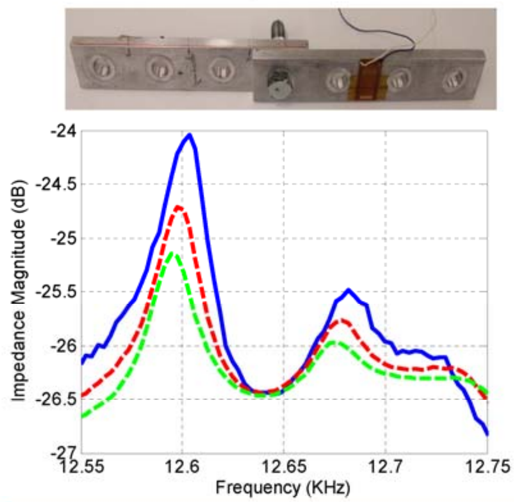


Can be induced by:

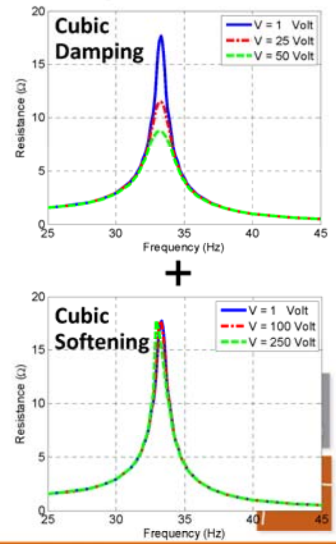
- **The PZT actuator**
- **The adhesive bonding layer**

Experimental results are then revisited in light of the simplified model

Lap Joint



Single-Mode Model



Study the effects of damage-induced nonlinearities on the high frequency dynamic response

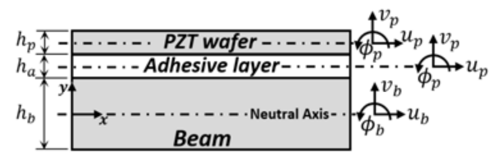
- ***Approach:***

  1- Develop coupled-field elements for PZT-structure interaction

  2- Study the effects of the adhesive bonding layer on the high frequency dynamic response

Coupled-field Elements for PZT-structure Interaction

- Three-layer element is formulated



Assumptions

- Linear elasticity and piezoelectricity
- Timoshenko beam theory for lateral displacements
- Elementary rod theory for longitudinal displacements

Equations of Motion

$$\rho A_1 \ddot{u}_b - \rho A_2 \ddot{\phi}_b - \rho A_3 \ddot{\phi}_a - \rho A_4 \ddot{\phi}_p - EA_1 \grave{\dot{u}}_b + EA_2 \grave{\dot{\phi}}_b + EA_3 \grave{\dot{\phi}}_a + EA_4 \grave{\dot{\phi}}_p = 0$$

$$\rho A_1 \ddot{v}_b - GA_1 \grave{\dot{v}}_b + GA_2 \grave{\phi}_b + GA_3 \grave{\phi}_a + GA_4 \grave{\phi}_p = 0$$

$$-\rho A_2 \ddot{u}_b + \rho I_1 \ddot{\phi}_b + \rho I_2 \ddot{\phi}_a + \rho I_3 \ddot{\phi}_p + EA_2 \grave{\dot{u}}_b - EI_1 \grave{\dot{\phi}}_b - EI_2 \grave{\dot{\phi}}_a + EI_3 \grave{\dot{\phi}}_p - GA_2 (\grave{v}_b - \phi_b) = 0$$

$$-\rho A_3 \ddot{u}_b + \rho I_2 \ddot{\phi}_b + \rho I_4 \ddot{\phi}_a + \rho I_5 \ddot{\phi}_p + EA_3 \grave{\dot{u}}_b - EI_2 \grave{\dot{\phi}}_b - EI_4 \grave{\dot{\phi}}_a - EI_5 \grave{\dot{\phi}}_p - GA_3 (\grave{v}_b - \phi_a) = 0$$

$$-\rho A_4 \ddot{u}_b + \rho I_3 \ddot{\phi}_b + \rho I_5 \ddot{\phi}_a + \rho I_6 \ddot{\phi}_p + EA_4 \grave{\dot{u}}_b - EI_3 \grave{\dot{\phi}}_b - EI_5 \grave{\dot{\phi}}_a - EI_6 \grave{\dot{\phi}}_p - GA_4 (\grave{v}_b - \phi_p) = 0$$

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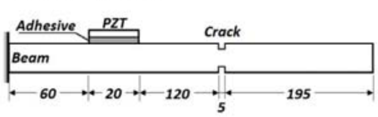
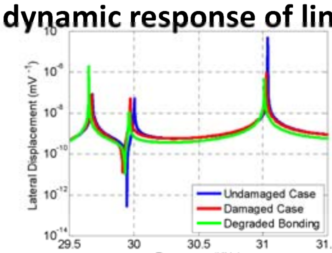
High Frequency Dynamic Response

- Spectral and finite element methods are employed to simulate the high frequency dynamic response of linear structures.





Future Work:

- Extend the model to account for structural nonlinearities
- Study the effects of nonlinearities (friction, nonlinear boundaries, breathing cracks) on the high frequency dynamic response

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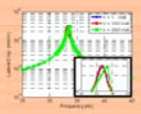
IN SUMMARY

SDOF

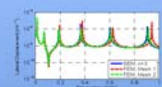
Coupled model providing the EMI with nonlinearities



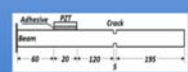
High Frequency ??

Representative Model

High Frequency Models



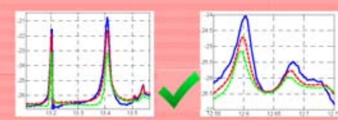
Electro mechanical Coupling with bonding layer



Non-linear high frequency ??

Experiments

High frequency "Linear" and non-linear response



Being able to relate the mode to the non-linear behavior ??

Full high frequency models with nonlinear behavior in order to verify experiential results

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Conclusions

- Experimental findings suggest the possibility of detecting structural nonlinearities using impedance measurements
- For a structure with no apparent nonlinearities, no frequency "shifts are noticed" in impedance response
- For a lap joint, impedance peaks are found to shift to lower frequencies as excitation voltage increases
- Numerical simulations can guide the classification and identification of structural nonlinearities
- With amplitude sweeps, impedance measurement can provide an easy, quick, and affordable way to track structural nonlinearities

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PATH FORWARD

- Compare standard/common testing methods for non-linearities to EMI monitoring with known non-linear behavior (to be presented at IMAC)
- Capabilities and bounds of of EMI technique to detect, excite and characterize non-linear behavior.
- How does nonlinearity of PZT interact with the nonlinearity of host structure
  - Will it affect (amplify) the measurement
  - How does the range of sensitivity affect the range of detection (global vs local effects)
- Expand the full model to incorporate non-linearities

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Advisor | Dr. Pablo Tarazaga 22/16 Virginia Tech © VAST

Thank You

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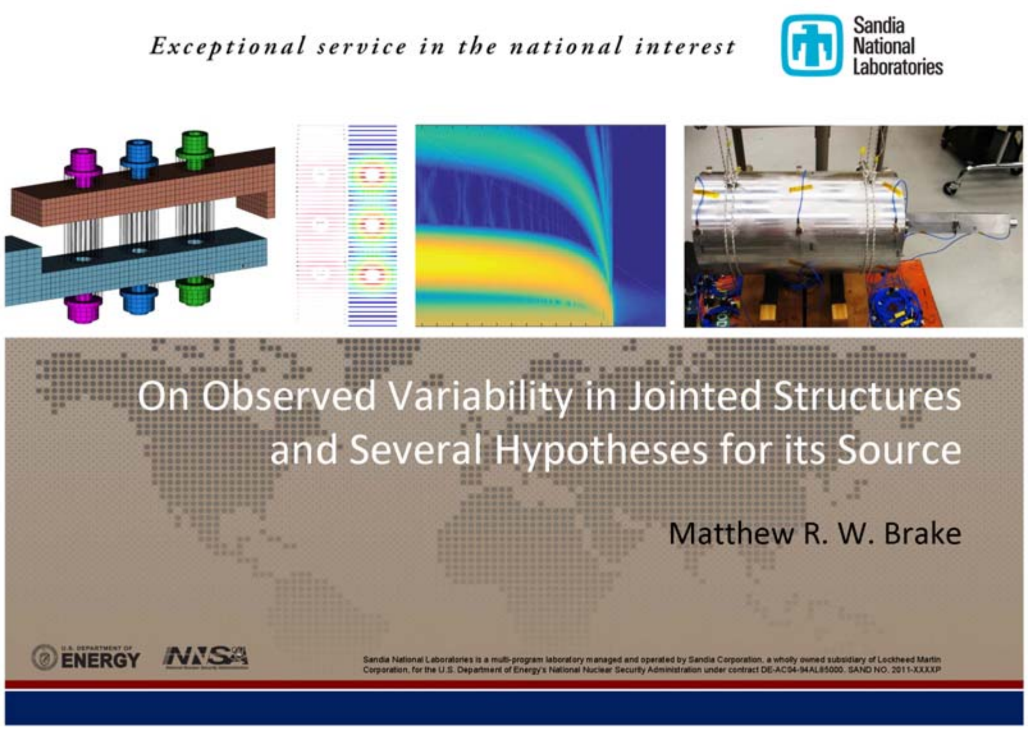
Advisor | Dr. Pablo Tarazaga 23 Virginia Tech © VAST

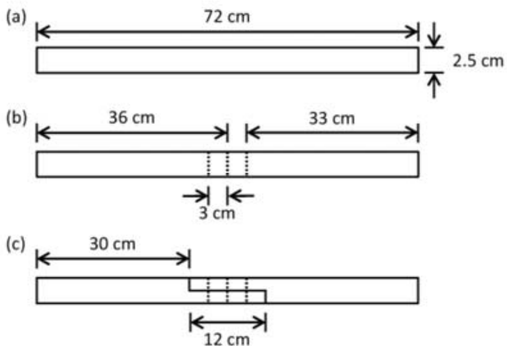
5.4. Session 2: Repeatability

Matthew Brake: On Observed Variability in Jointed Structures and Several Hypotheses for its Source



Benchmark System

- Brake-Reuß beam: 72 cm long beam with a lap joint
- Multiple versions to assess contribution of joint to dynamics
- Results from three studies summarized here

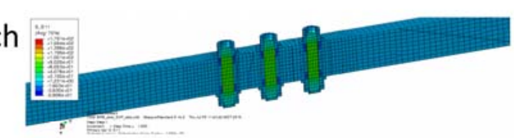


v1: metric units, off-center excitation
v2: English units, on-center excitation
v3: English units, on-center and off-center excitation

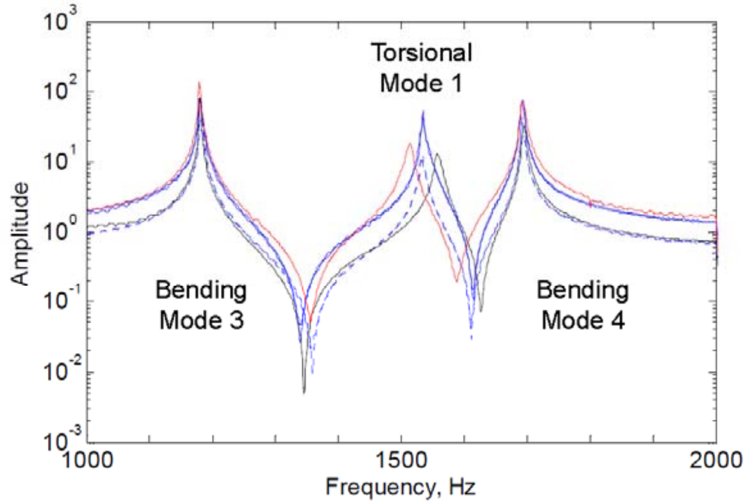
Monolithic Beam Results

	Monolithic		Monolithic with Bolts Tightened to 10 Nm	
Mode	Natural Frequency (Hz)	Damping Ratio (%)	Natural Frequency (Hz)	Damping Ratio (%)
Bending 1	251.4	0.0594	242.7	0.0991
Bending 2	687.4	0.0358	685.4	0.0505
Bending 3	1333.1	0.1199	1290.9	0.1840
Bending 4	2172.1	0.1216	2146.3	0.3198
Bending 5	3196.4	0.2848	3118.2	0.3560
Bending 6	4385.4	0.2374	4307.3	0.2958
Torsional 1	2053.1	0.1672	2027.9	0.2313
Torsional 2	4093.0	1.1047	3778.5	0.6863

- Results very repeatable.
- Monolithic beam results match FEA frequencies exactly.
- Some variability observed in torsional modes due to supports

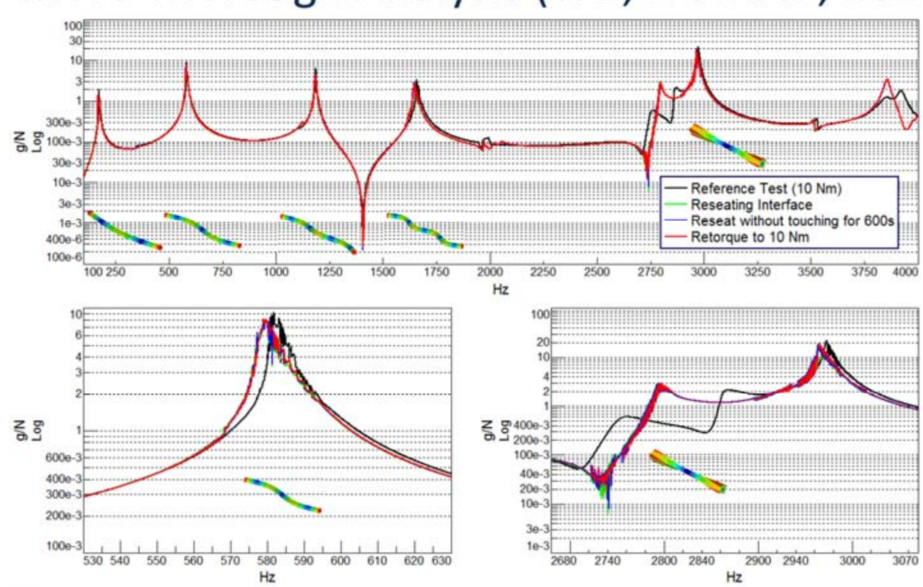


Preliminary Analysis (v. 1, Stuttgart, 2013)



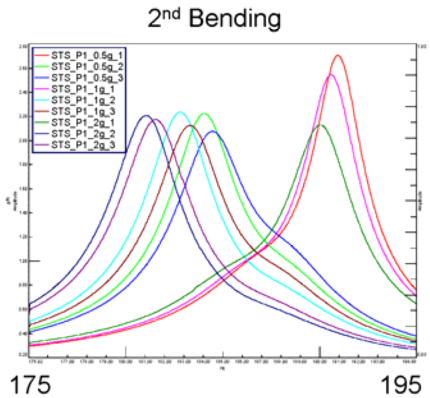
- 5 Nm torque, free-free BC, shaker excitation off-center line
- Torsional modes very sensitive to preloads and tightening order.
- Colors: different tightening orders; Lines: different assemblies of same system

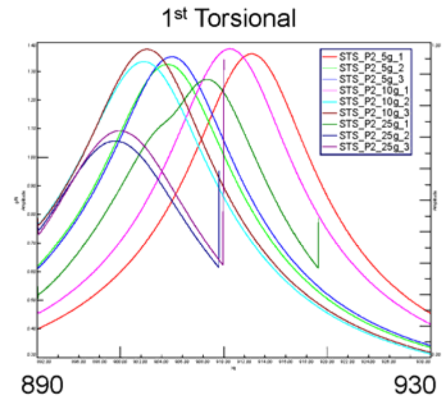
More Thorough Analysis (v. 2, NOMAD, 2014)



- Free-free boundary condition, 10 Nm torque, shaker excitation on-center line

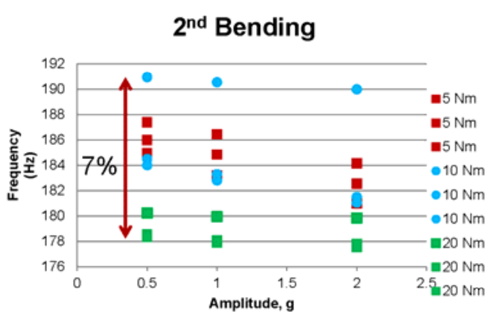
In Depth Analysis (v. 3, NOMAD 2015)

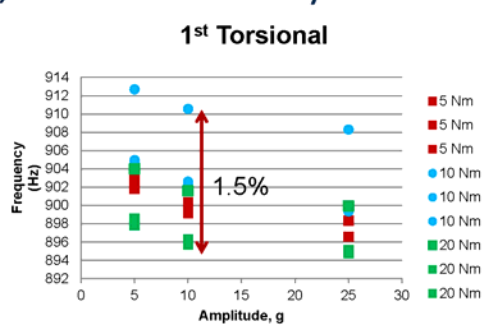


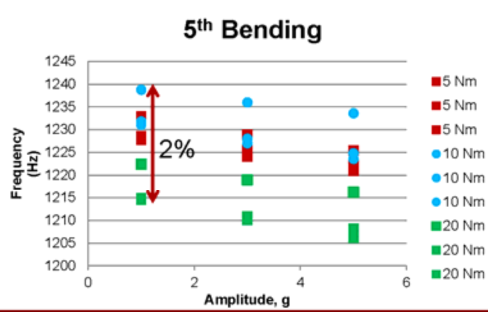


- 10 Nm torque, Clamped-Free boundary condition, shaker excitation off-center line
- 3 different excitation levels tested
- Focus of NOMAD 2015 project on control scheme for reliable FRF measurements

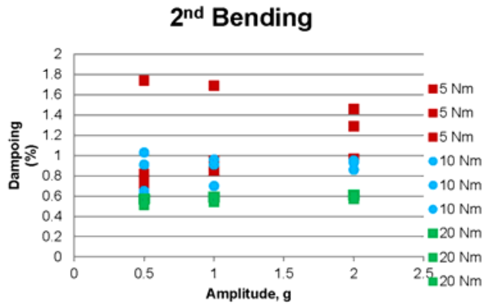
In Depth Analysis (v. 3, NOMAD 2015)

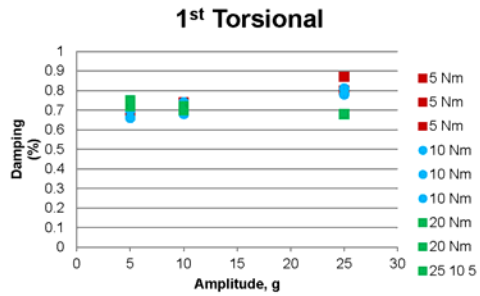


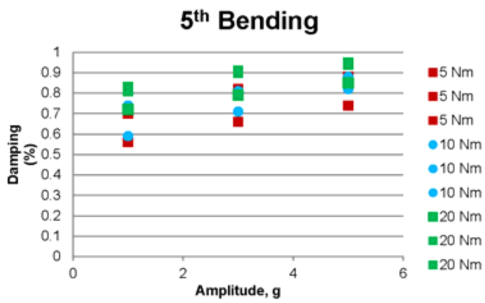




In Depth Analysis (v. 3, NOMAD 2015)



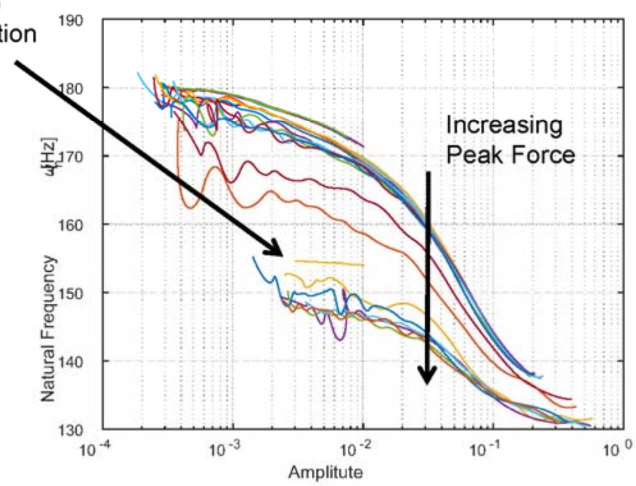




- Opposite trend observed for clamped-free case: torsional modes exhibit lowest variability
- In contrast with previous free-free experiments

In Depth Analysis (v. 3, NOMAD 2015)

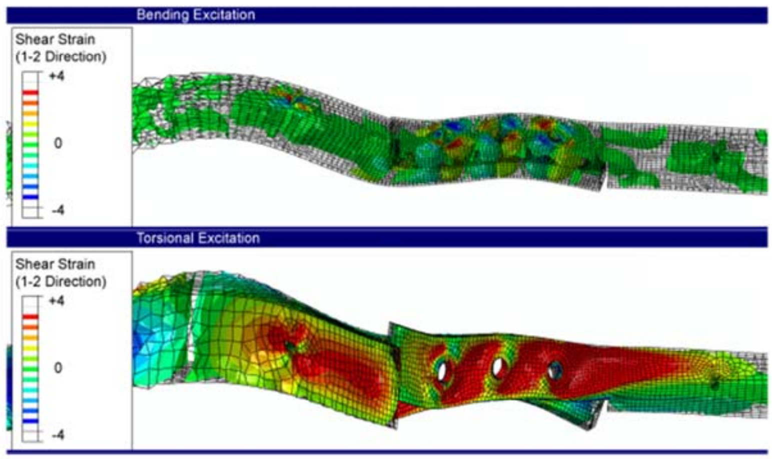
Retested after macroslip saturation; lowest excitation amplitude used.



- For samples that underwent macroslip testing, numerical modeling confirms that plasticity permanently altered frequency response

In Depth Analysis (v. 3, NOMAD 2015)

- Analysis of wave propagation through interface shows shear waves affected significantly by lower lateral stiffness…



A Few Hypotheses…

- Al Ferri: The joint characteristics are dependent on the surrounding structure (might explain cantilevered versus free differences)
- Randy Mayes: If you account for effective mass in the surrounding structure, joint properties should be intransient (at odds with Al's hypothesis)
- Matthew Brake: The differences in variability is attributable to the differences in stiffness in the characteristic directions (lap joints are designed to be stiff one way, but not the other…)

Acknowledgements

- University of Stuttgart: Simone Catalfamo, Florian Morlock, Fabian Schempp, Pascal Reuß
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- University of Wisconsin, Madison: Matt Bonney
- University of Illinois, Urbana-Champaign: Keegan Moore
- Oxford University: Rob Flicek
- Support provided by Simulia, Altair, and Siemens
- The 2014 NOMAD Research Institute was hosted by Sandia National Laboratories
- The 2015 NOMAD Research Institute was hosted by Sandia National Laboratories at the University of New Mexico

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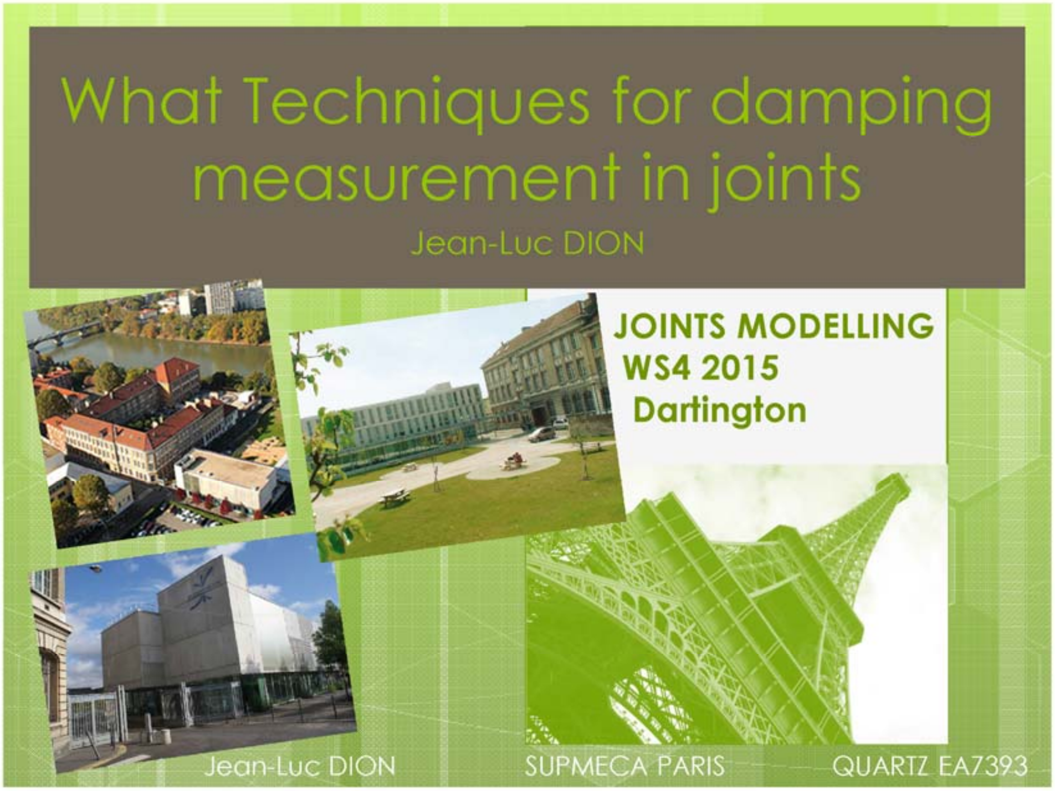
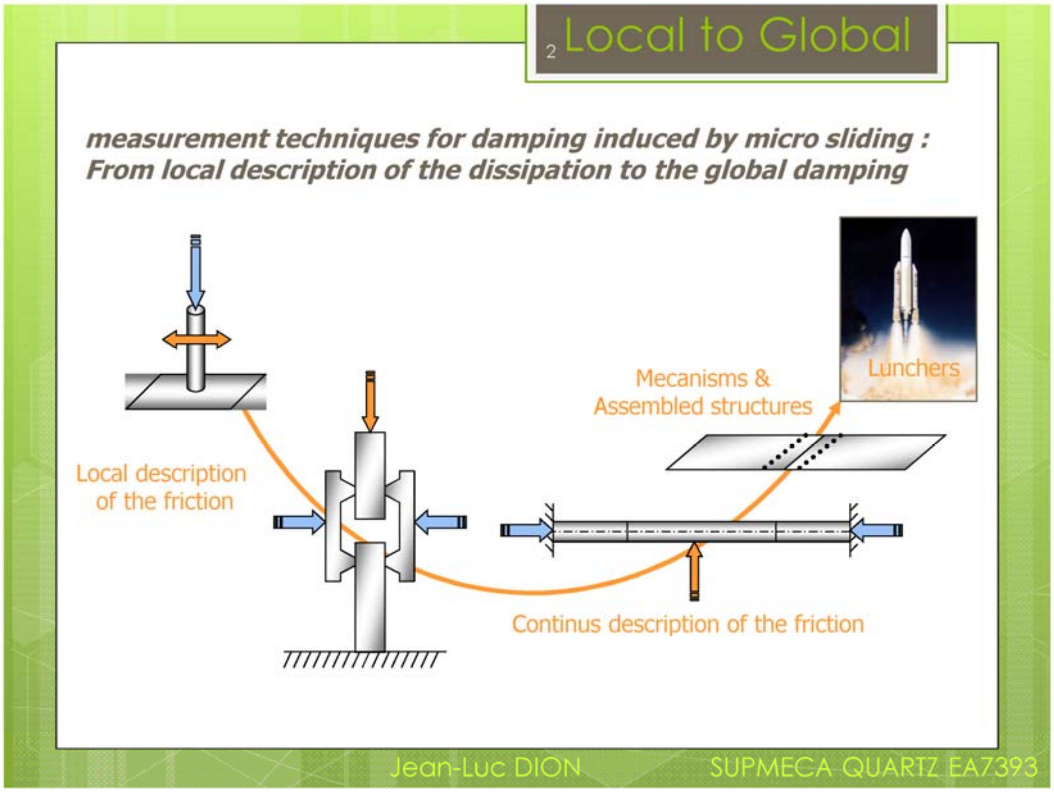
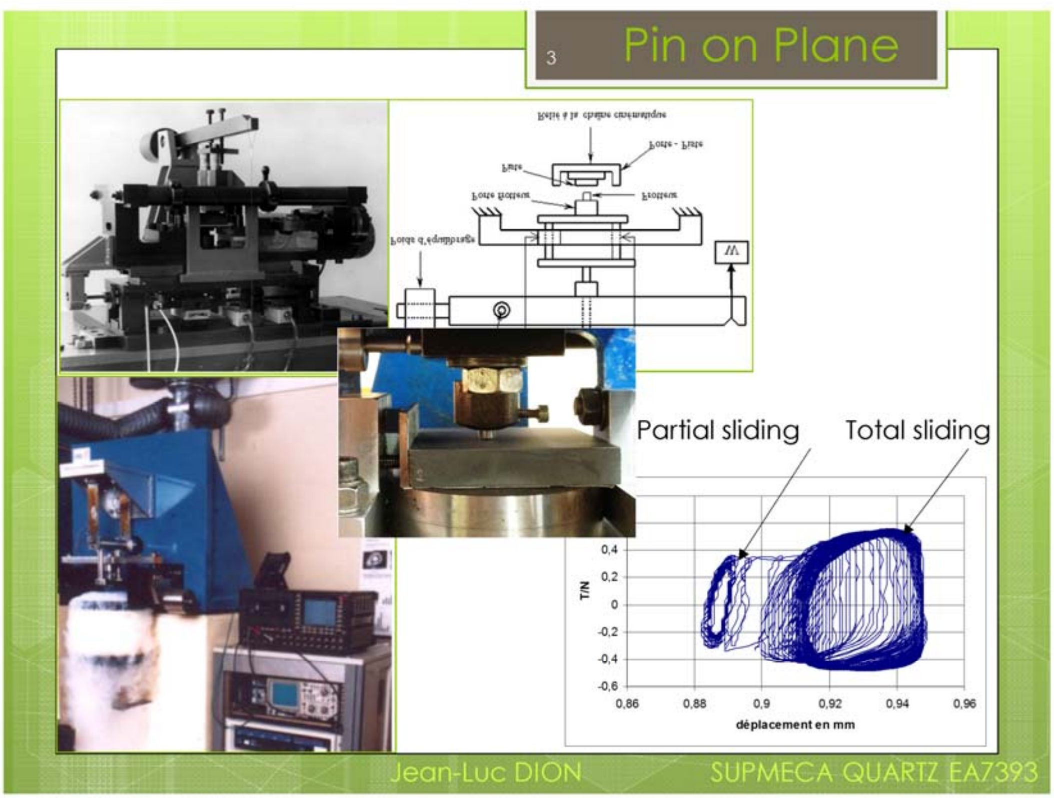
3. M. R. Brake, P. Reuss, D. J. Segalman, and L. Gaul, "Variability and Repeatability of Jointed Structures with Frictional Interfaces," *IMAC XXXII A Conference and Exposition on Structural Dynamics*, Orlando, FL, February 2014.
4. S. Smith, J. C. Bilbao-Ludena, S. Catalfamo, M. R. W. Brake, P. Reuss, and C. W. Schwingshackl, "The Effects of Boundary Conditions, Measurement Techniques, and Excitation Type on Measurements of the Properties of Mechanical Joints," *IMAC XXXIII A Conference and Exposition on Structural Dynamics*, Orlando, FL, February 2015.
5. S. Catalfamo, S. A. Smith, F. Morlock, C. Schwingshackl, P. Reuss, and M. R. W. Brake, "Effects of Experimental Methods on the Measurement of a Nonlinear System," *IMAC XXXIV A Conference and Exposition on Structural Dynamics*, Orlando, FL, January, 2016.
6. M. S. Bonney, B. A. Robertson, F. Schempp, M. R. W. Brake, M. Mignolet, "Experimental Determination of Frictional Interface Models," *IMAC XXXIV A Conference and Exposition on Structural Dynamics*, Orlando, FL, January, 2016.
7. R. C. Flicek, K. J. Moore, G. M. Castelluccio, C. Hammetter, and M. R. W. Brake, "Stress Waves Propagating Through Jointed Connections," *IMAC XXXIV A Conference and Exposition on Structural Dynamics*, Orlando, FL, January, 2016.

Jean-Luc Dion: What Techniques for Damping Measurement in Joints




2
Local to Global
measurement techniques for damping induced by micro sliding :
From local description of the dissipation to the global damping
Lunchers
Mecanisms &
Assembled structures
Local description
of the friction
Continus description of the friction
Jean-Luc DION
SUPMECA QUARTZ EA7393


3
Pin on Plane
Partial sliding
Total sliding
T/N
0,4
0,2
0
-0,2
-0,4
-0,6
0,86
0,88
0,9
0,92
0,94
0,96
déplacement en mm
Jean-Luc DION
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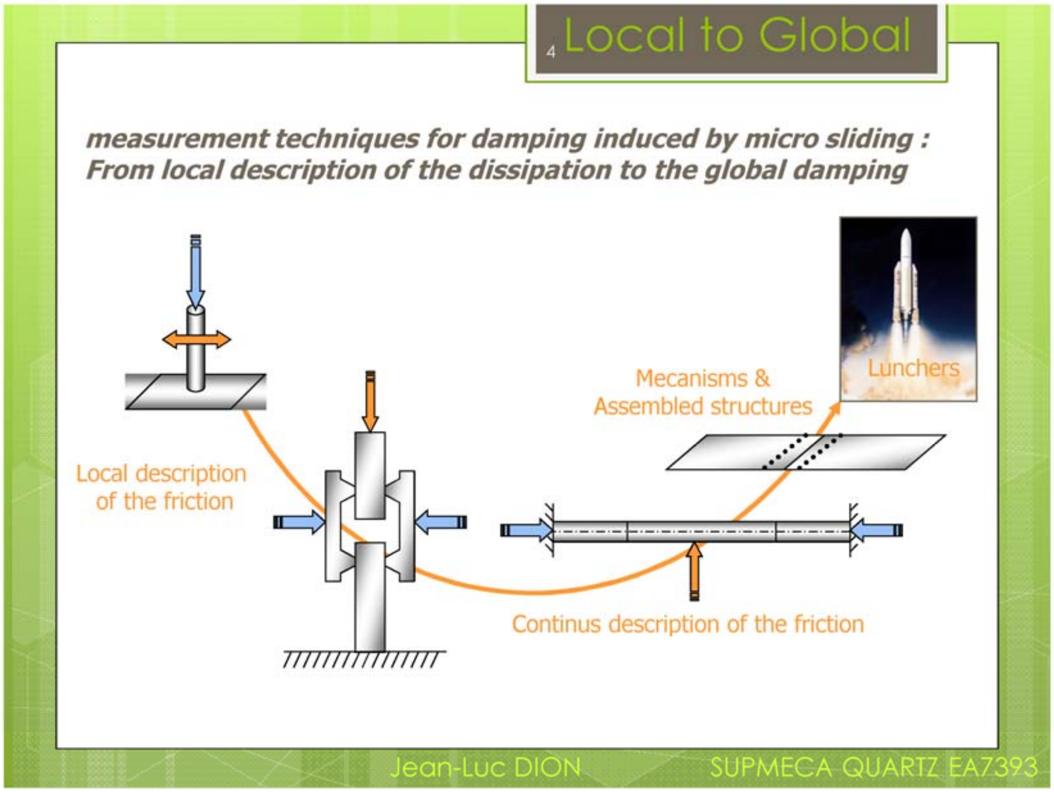
4
Local to Global
measurement techniques for damping induced by micro sliding :
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Lunchers
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Local description
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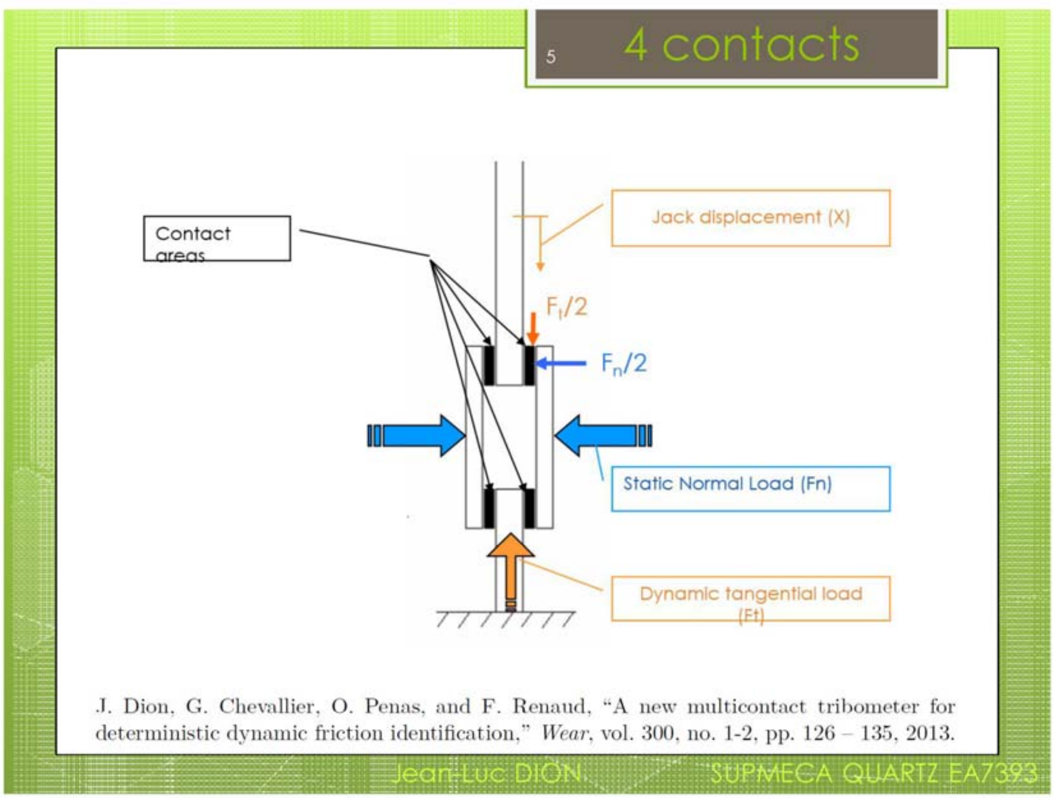
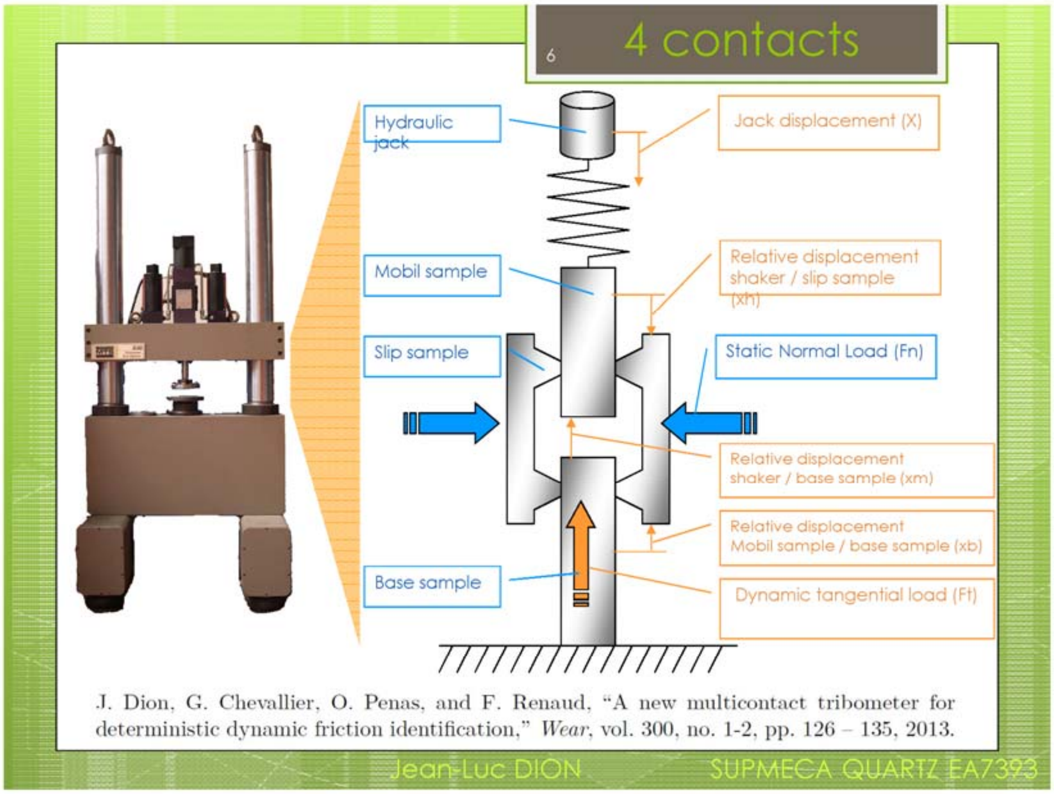

5
4 contacts
Contact
areas
Jack displacement (X)
$F_t/2$
$F_n/2$
Static Normal Load (Fn)
Dynamic tangential load
(Ft)
J. Dion, G. Chevallier, O. Penas, and F. Renaud, "A new multicontact tribometer for deterministic dynamic friction identification," *Wear*, vol. 300, no. 1-2, pp. 126 – 135, 2013.
Jean-Luc DION
SUPMECA QUARTZ EA7393


6
4 contacts
Hydraulic jack
Jack displacement (X)
Mobil sample
Relative displacement shaker / slip sample (xh)
Slip sample
Static Normal Load (Fn)
Relative displacement shaker / base sample (xm)
Relative displacement Mobil sample / base sample (xb)
Base sample
Dynamic tangential load (Ft)
J. Dion, G. Chevallier, O. Penas, and F. Renaud, "A new multicontact tribometer for deterministic dynamic friction identification," Wear, vol. 300, no. 1-2, pp. 126 – 135, 2013.
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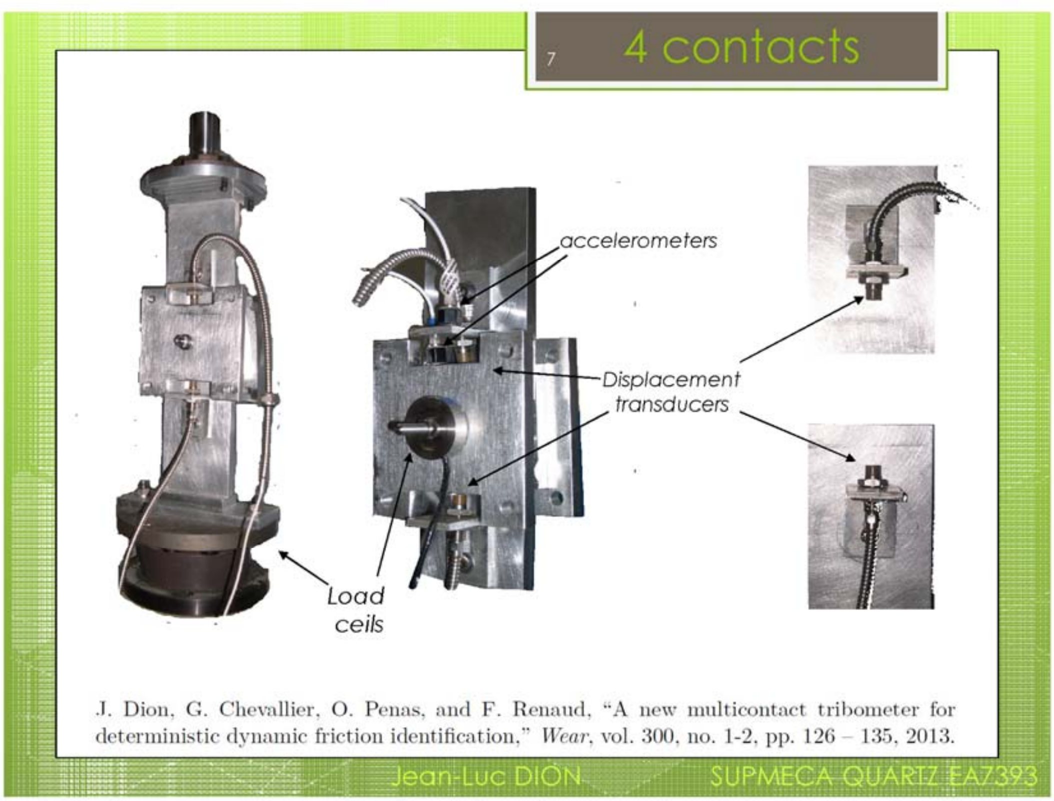
7
4 contacts
accelerometers
Displacement transducers
Load ceils
J. Dion, G. Chevallier, O. Penas, and F. Renaud, "A new multicontact tribometer for deterministic dynamic friction identification," Wear, vol. 300, no. 1-2, pp. 126 – 135, 2013.
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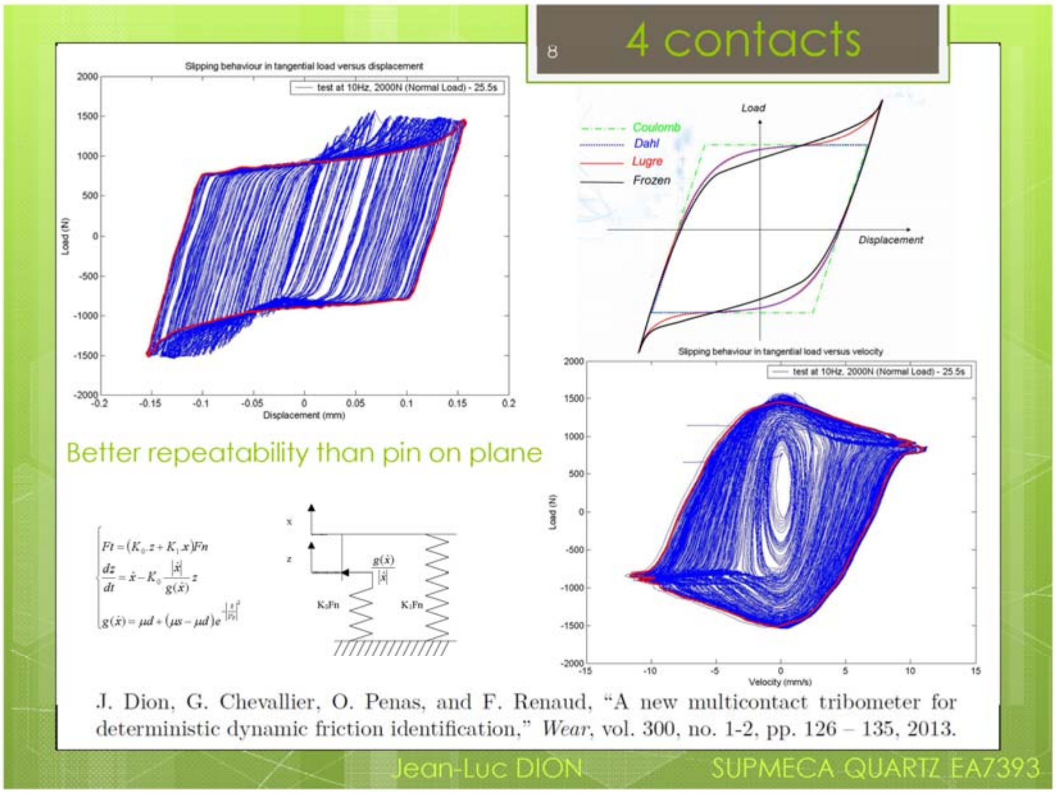
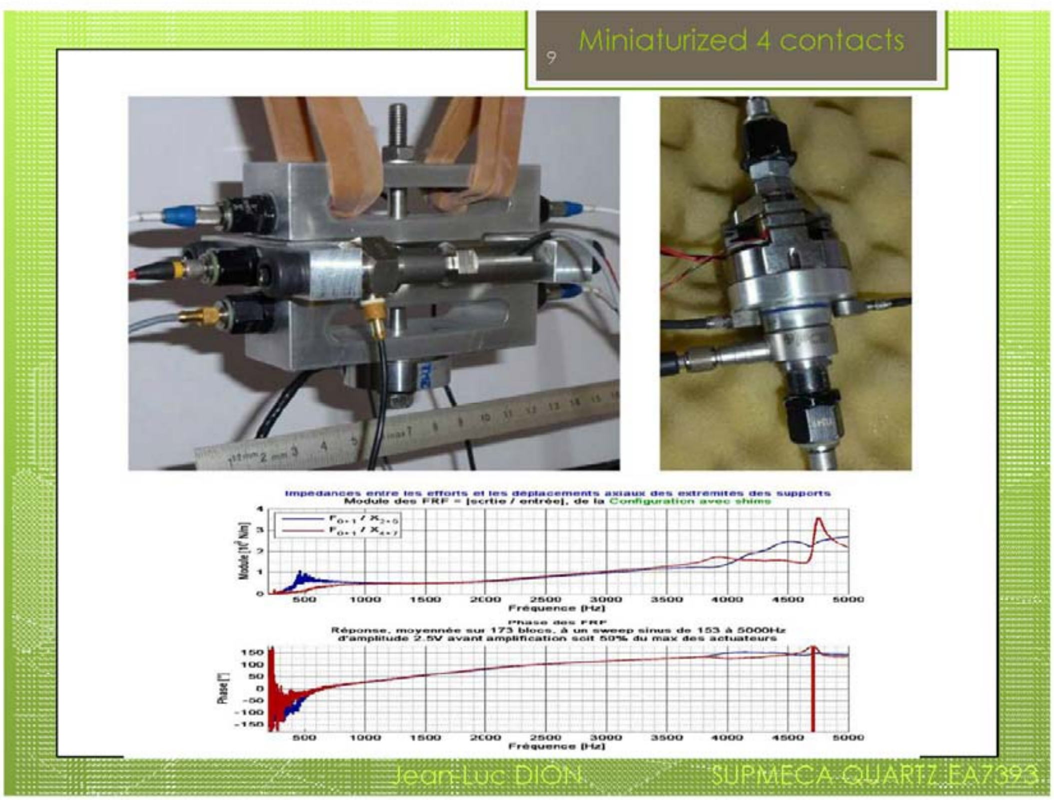
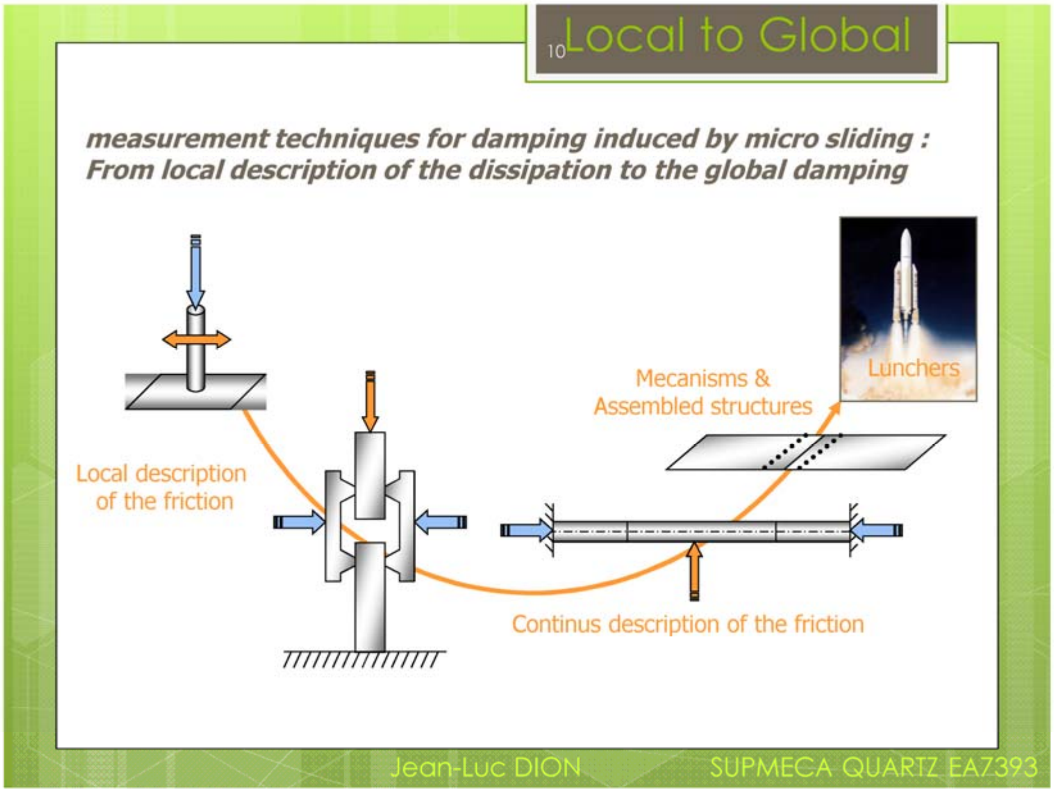

8
4 contacts
Slipping behaviour in tangential load versus displacement
test at 10Hz, 2000N (Normal Load) - 25.5s
Load (N)
Displacement (mm)
Load
Coulomb
Dahl
Lugre
Frozen
Displacement
Slipping behaviour in tangential load versus velocity
test at 10Hz, 2000N (Normal Load) - 25.5s
Load (N)
Velocity (mm/s)
Better repeatability than pin on plane
K0Fn
K1Fn
J. Dion, G. Chevallier, O. Penas, and F. Renaud, "A new multicontact tribometer for deterministic dynamic friction identification," Wear, vol. 300, no. 1-2, pp. 126 – 135, 2013.
Jean-Luc DION
SUPMECA QUARTZ EA7393


9
Miniaturized 4 contacts
Fréquence [Hz]
Phase [°]
Jean-Luc DION
SUPMECA QUARTZ EA7393


10 Local to Global
measurement techniques for damping induced by micro sliding :
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Local description
of the friction
Continus description of the friction
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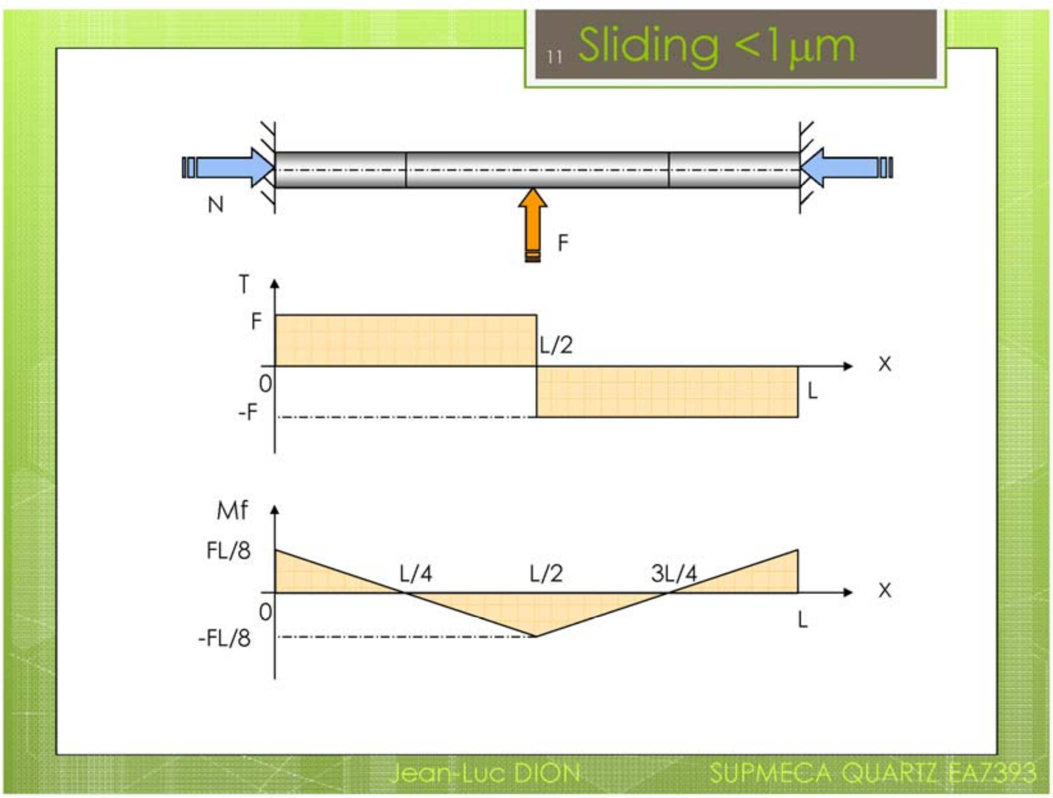
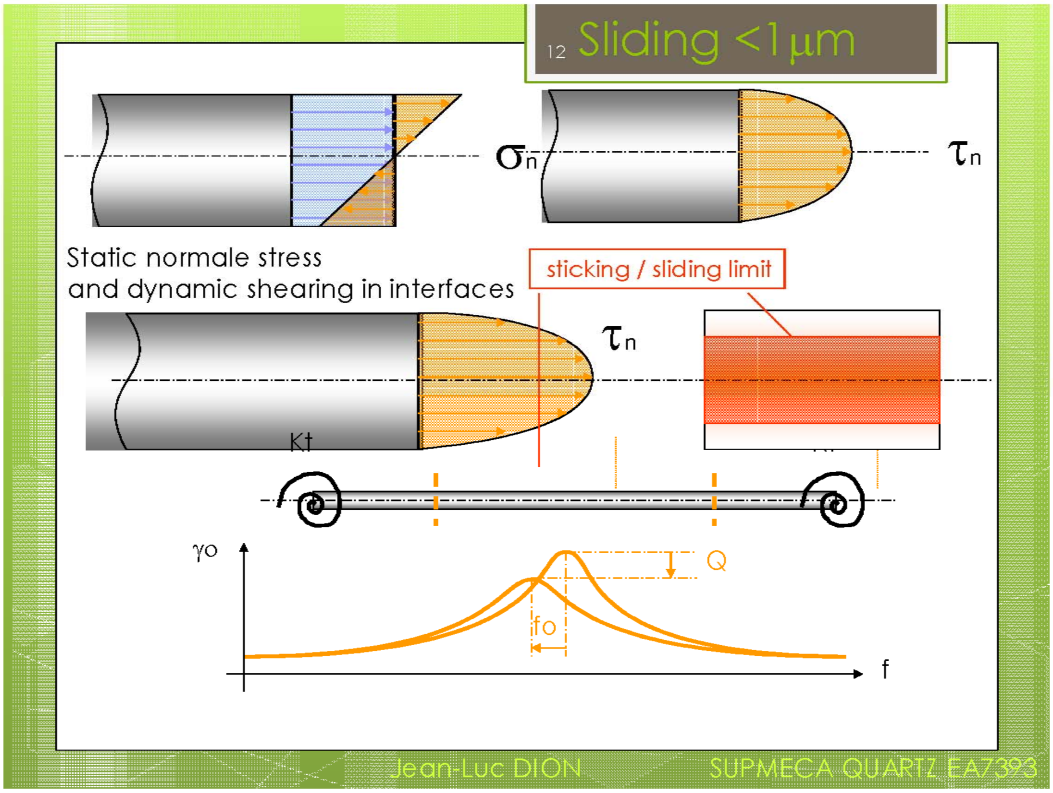
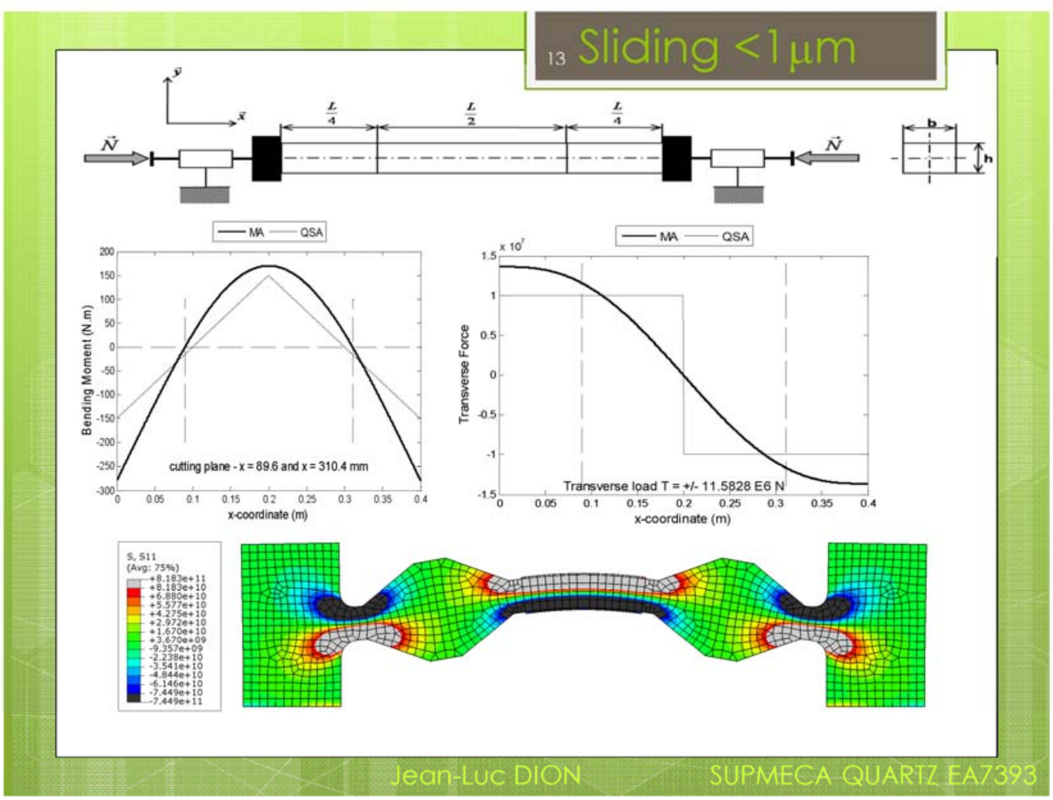
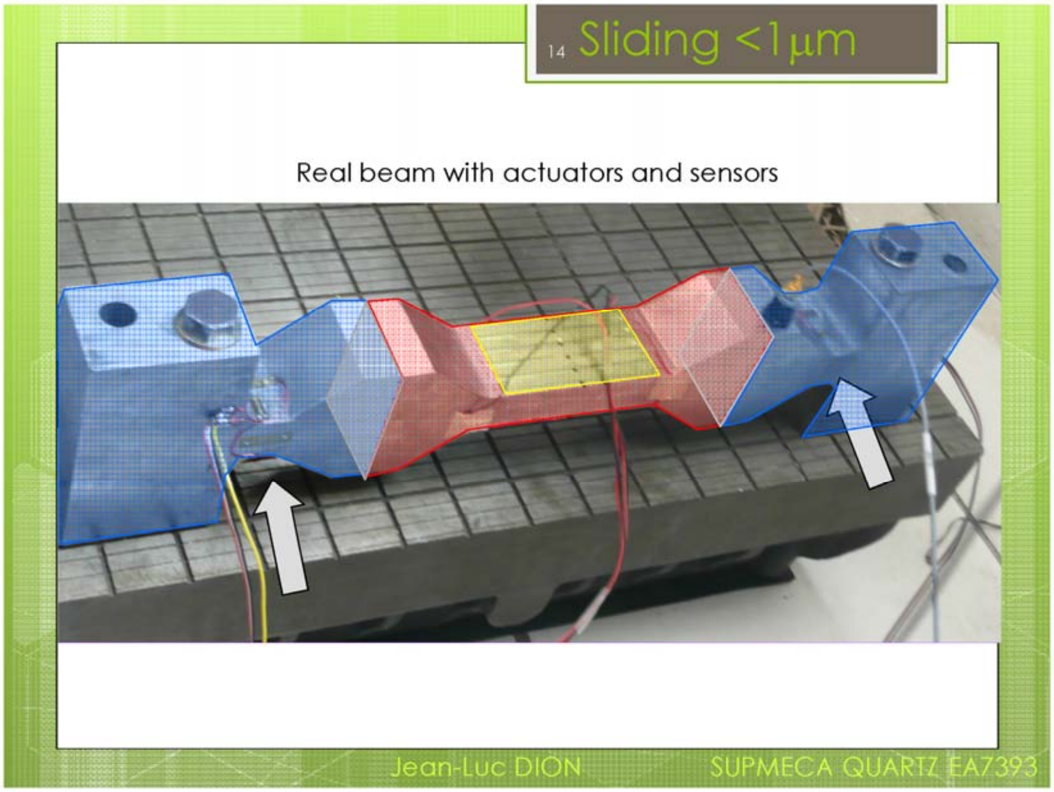
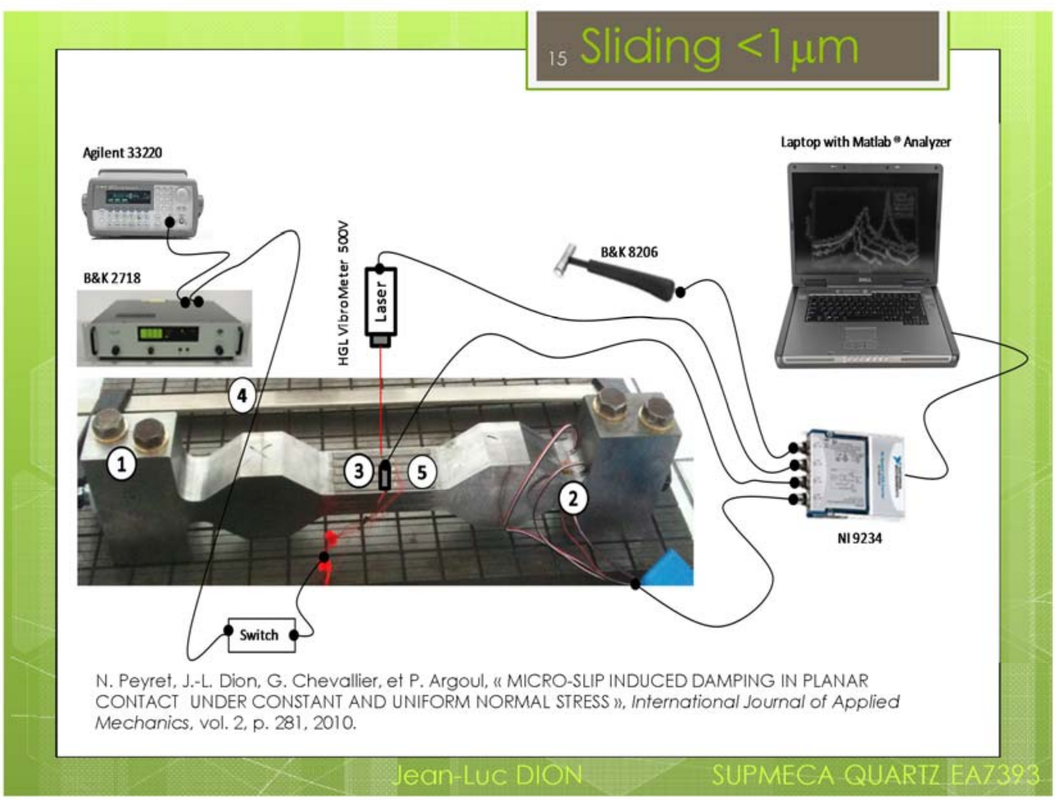
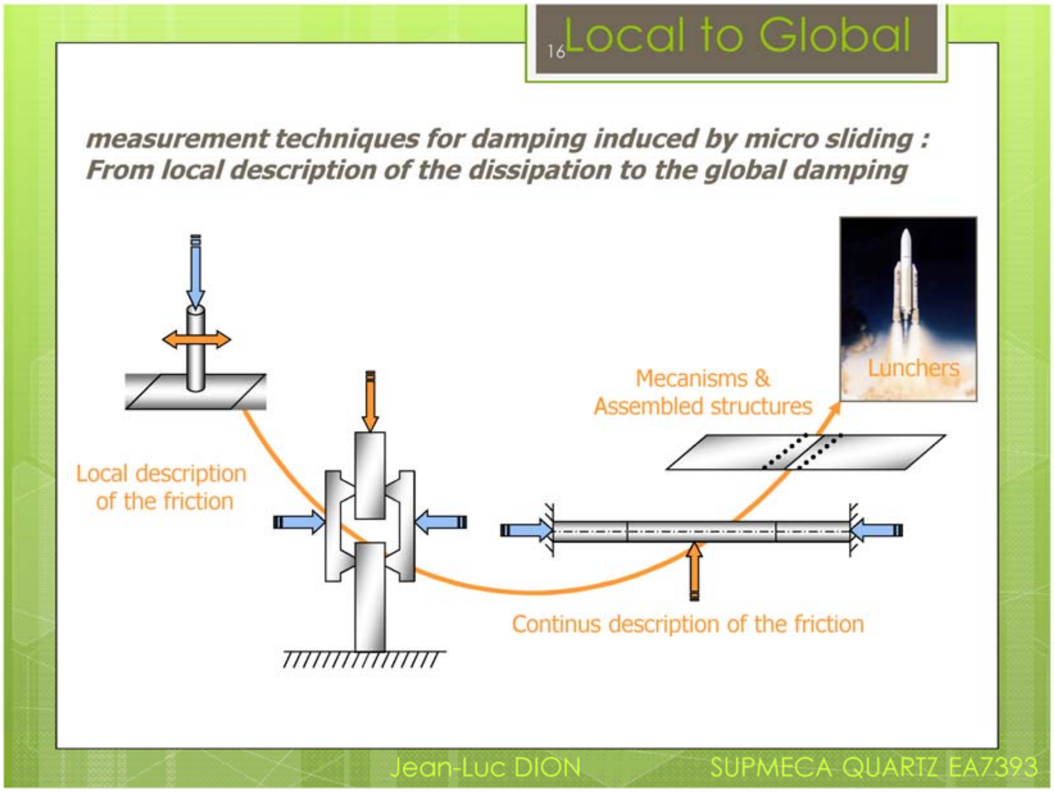

11 Sliding <1 μm
N
F
T
F
L/2
X
0
L
-F
Mf
FL/8
L/4
L/2
3L/4
X
0
L
-FL/8
Jean-Luc DION
SUPMECA QUARTZ EA7393

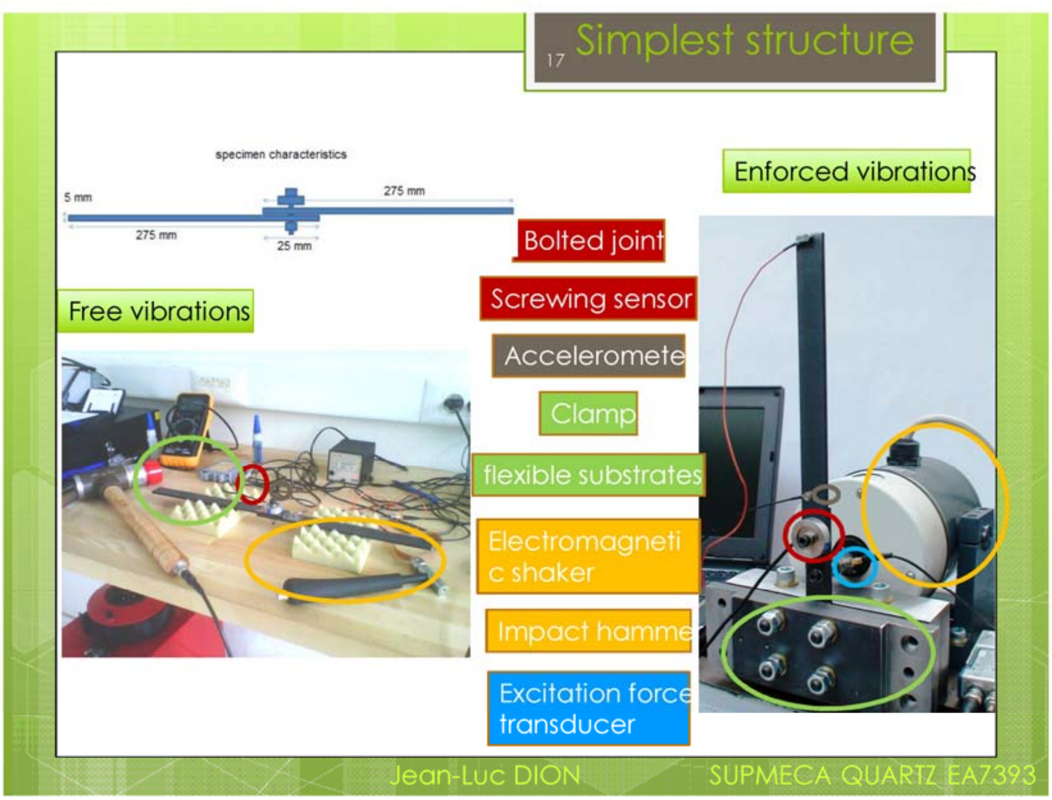

12 Sliding <1 µm
σn
τn
Static normale stress
and dynamic shearing in interfaces
sticking / sliding limit
τn
Kt
γo
Q
fo
f
Jean-Luc DION
SUPMECA QUARTZ EA7393


13 Sliding <1 µm
MA
QSA
Bending Moment (N.m)
cutting plane - x = 89.6 and x = 310.4 mm
x-coordinate (m)
Transverse Force
Transverse load T = +/- 11.5828 E6 N
x-coordinate (m)
S, S11
(Avg: 75%)
Jean-Luc DION
SUPMECA QUARTZ EA7393

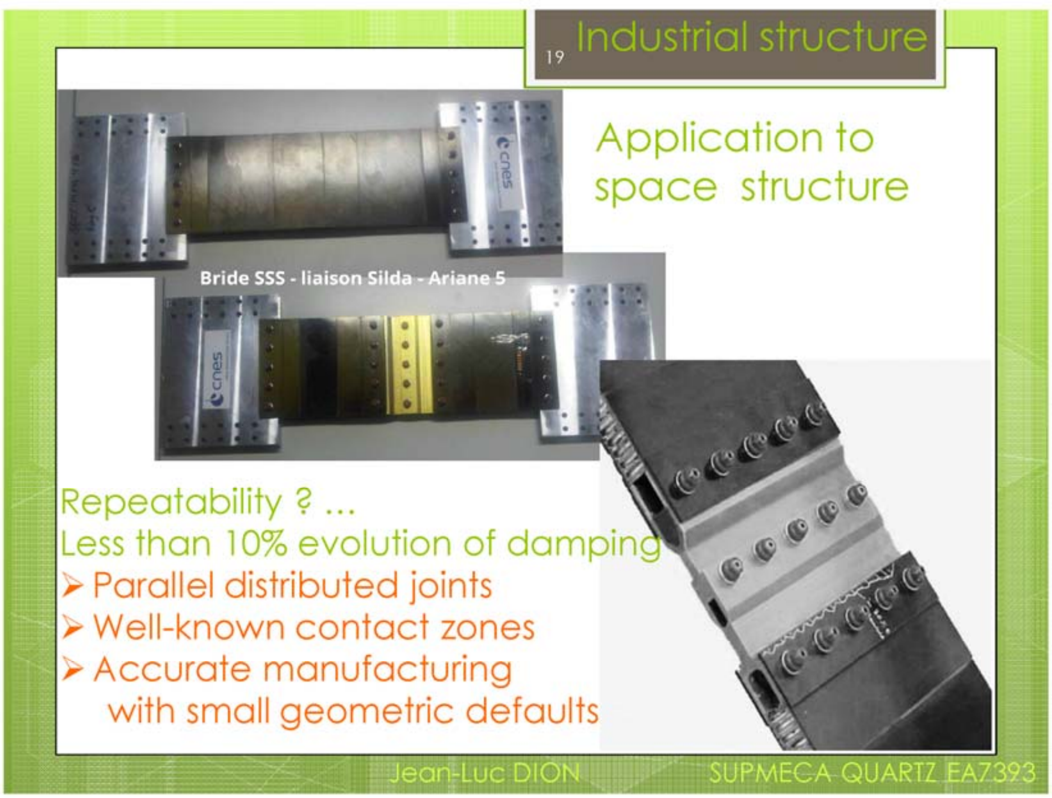

14 Sliding <1µm
Real beam with actuators and sensors
Jean-Luc DION
SUPMECA QUARTZ EA7393

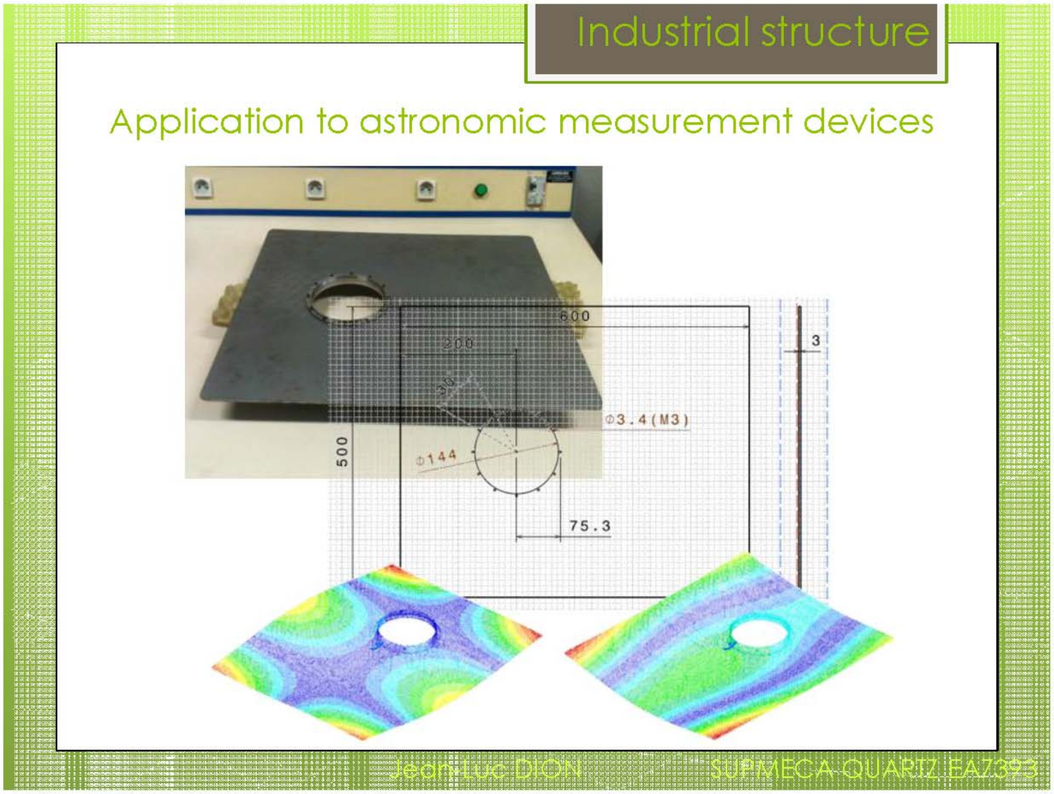

15 Sliding <1µm
Agilent 33220
B&K 2718
HGL VibroMeter 500V
Laser
B&K 8206
Laptop with Matlab ® Analyzer
NI 9234
Switch
1
2
3
4
5
N. Peyret, J.-L. Dion, G. Chevallier, et P. Argoul, « MICRO-SLIP INDUCED DAMPING IN PLANAR CONTACT UNDER CONSTANT AND UNIFORM NORMAL STRESS », International Journal of Applied Mechanics, vol. 2, p. 281, 2010.
Jean-Luc DION
SUPMECA QUARTZ EA7393

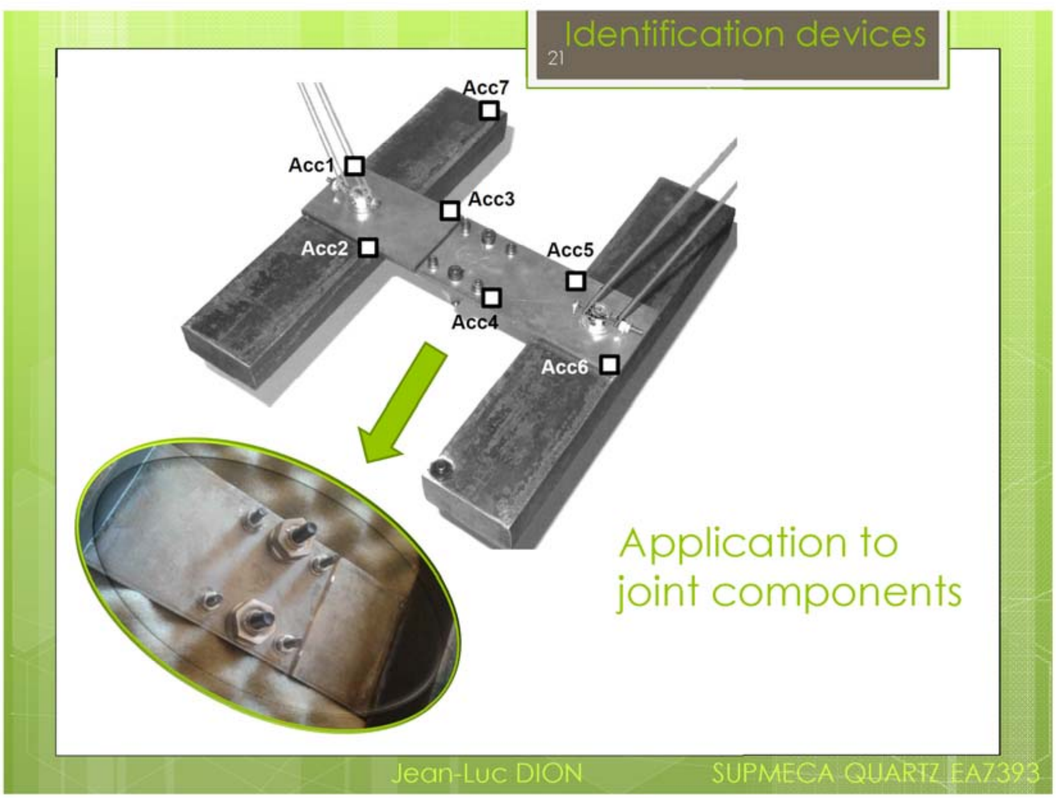

16 Local to Global
measurement techniques for damping induced by micro sliding :
From local description of the dissipation to the global damping
Local description
of the friction
Mecanisms &
Assembled structures
Lunchers
Continus description of the friction
Jean-Luc DION
SUPMECA QUARTZ EA7393

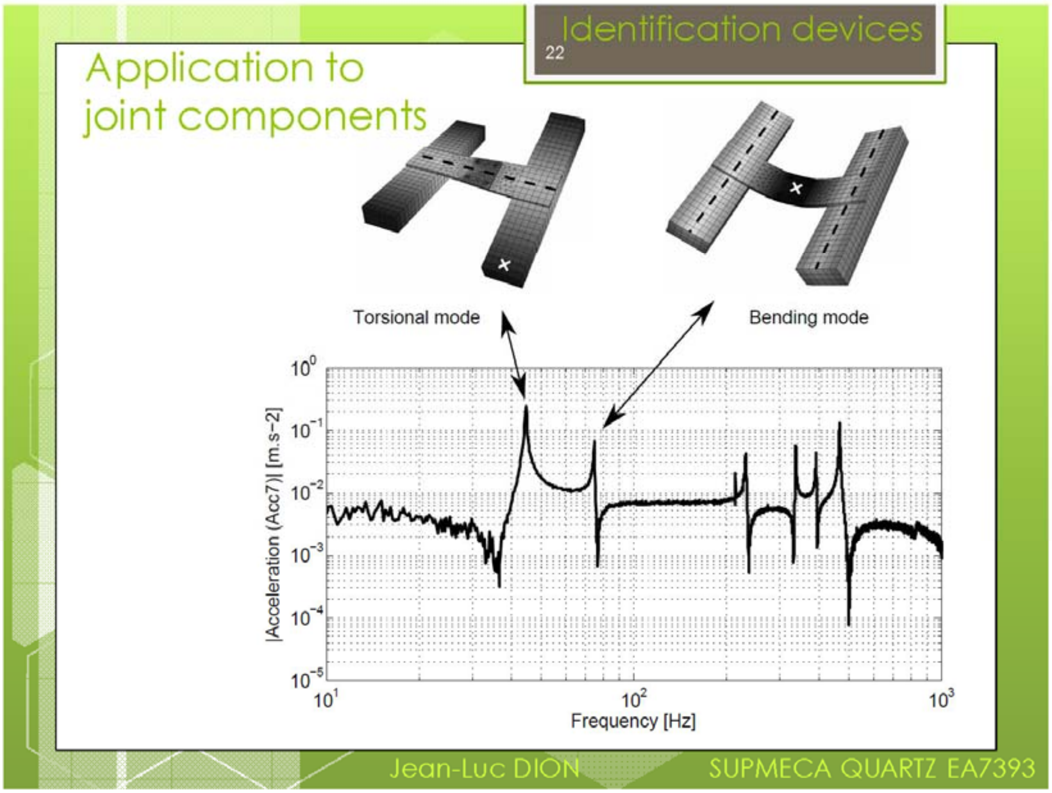

17 Simplest structure
specimen characteristics
5 mm
275 mm
275 mm
25 mm
Free vibrations
Enforced vibrations
Bolted joint
Screwing sensor
Acceleromete
Clamp
flexible substrates
Electromagneti
c shaker
Impact hamme
Excitation force
transducer
Jean-Luc DION
SUPMECA QUARTZ EA7393

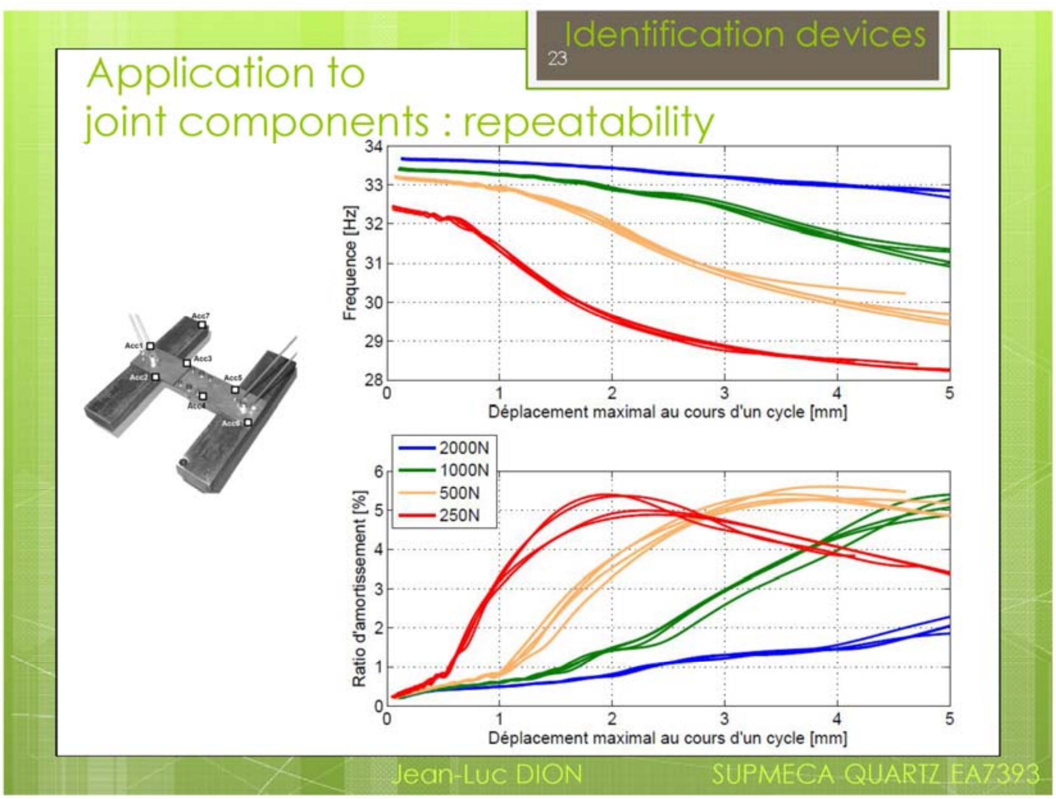

18
Simplest structure
Repeatability ? …
more than 100% evolution of damping
B&K 4517
B&K 8206
Mesure et post-traitements
NI 9234
Jean-Luc DION
SUPMECA QUARTZ EA7393


19
Industrial structure
Application to
space structure
Bride SSS - liaison Silda - Ariane 5
Repeatability ? …
Less than 10% evolution of damping
➢ Parallel distributed joints
➢ Well-known contact zones
➢ Accurate manufacturing
with small geometric defaults
Jean-Luc DION
SUPMECA QUARTZ EA7393

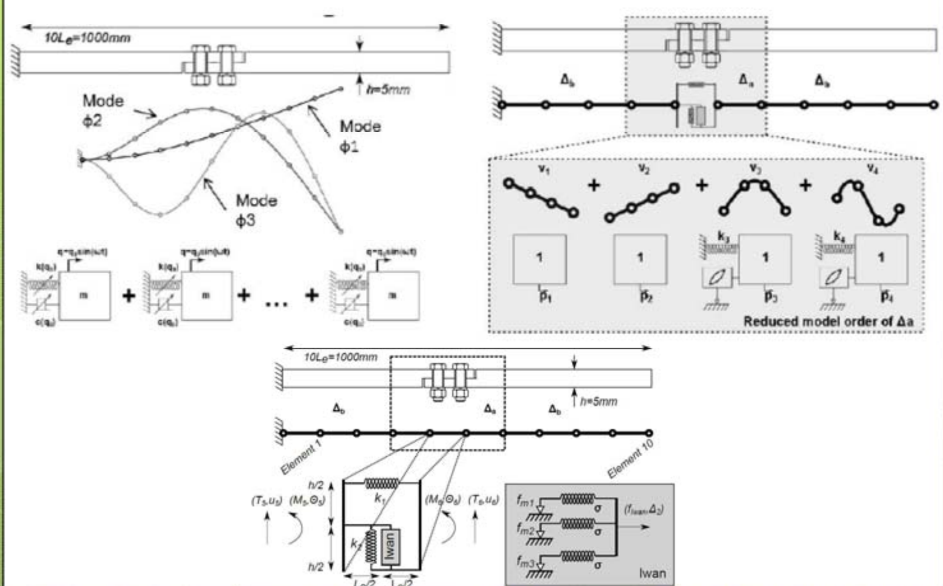

Industrial structure
Application to astronomic measurement devices
600
200
30
3
Φ3.4(M3)
500
Φ144
75.3
Jean-Luc DION
SUPMECA-QUARTZ EA7393


Identification devices
21
Acc7
Acc1
Acc3
Acc2
Acc5
Acc4
Acc6
Application to
joint components
Jean-Luc DION
SUPMECA-QUARTZ EA7393

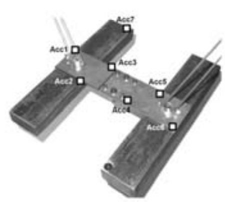

Identification devices
22
Application to
joint components
Torsional mode
Bending mode
|Acceleration (Acc7)| [m.s−2]
Frequency [Hz]
Jean-Luc DION
SUPMECA QUARTZ EA7393

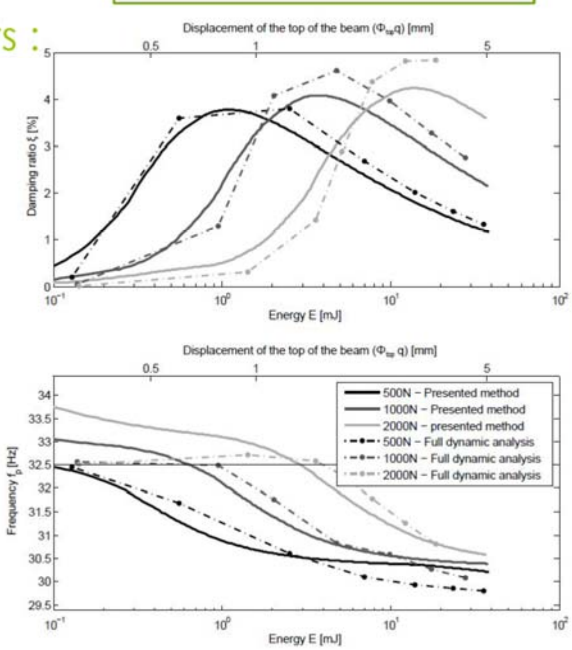

Identification devices
23
Application to
joint components : repeatability
Acc7
Acc1
Acc3
Acc2
Acc5
Acc4
Acc6
Frequence [Hz]
Déplacement maximal au cours d'un cycle [mm]
2000N
1000N
500N
250N
Ratio d'amortissement [%]
Déplacement maximal au cours d'un cycle [mm]
Jean-Luc DION
SUPMECA QUARTZ EA7393

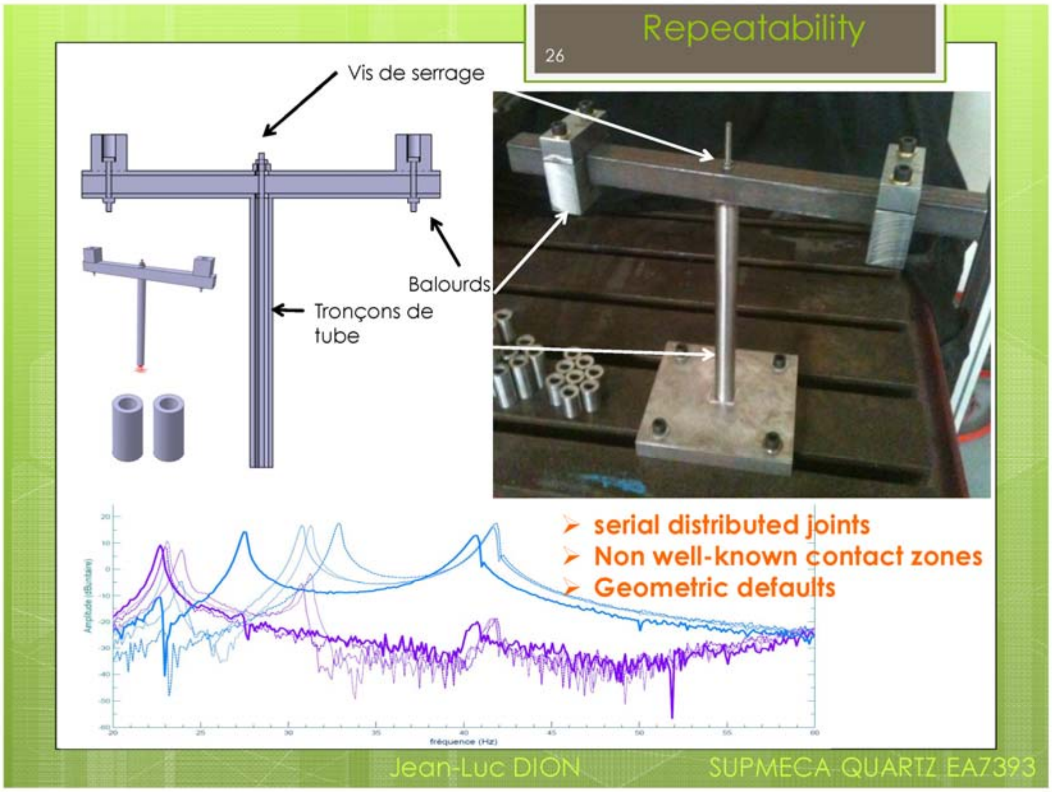
Application to joint components : identified models

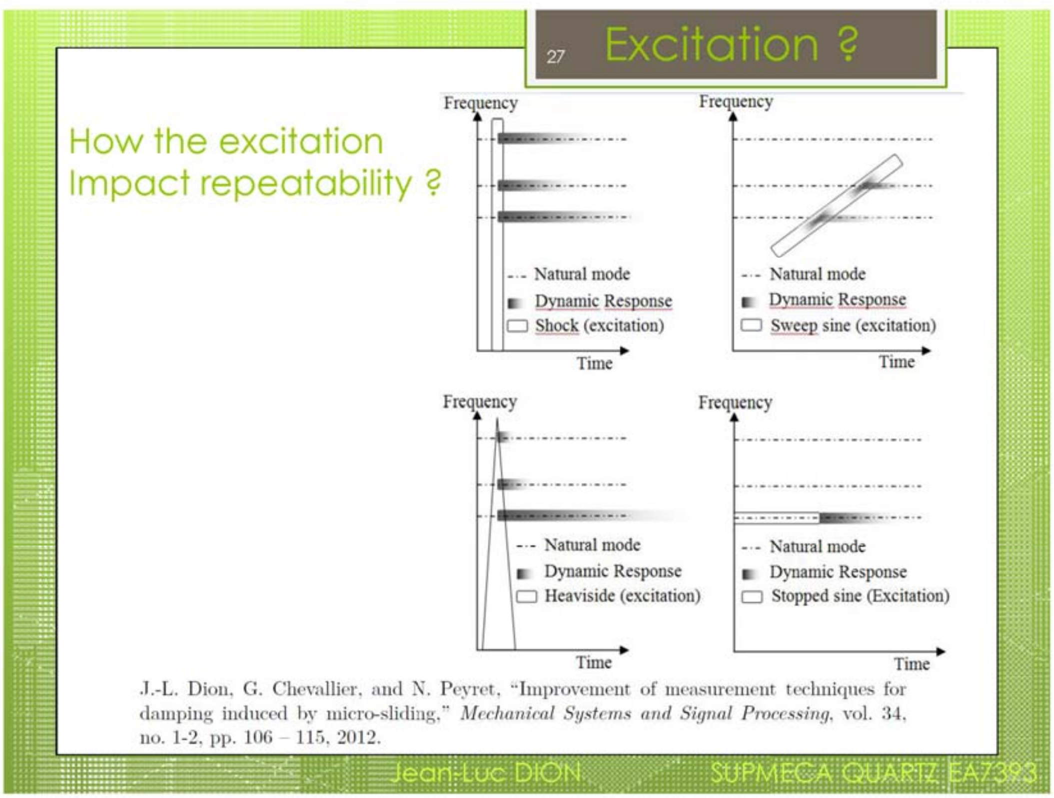


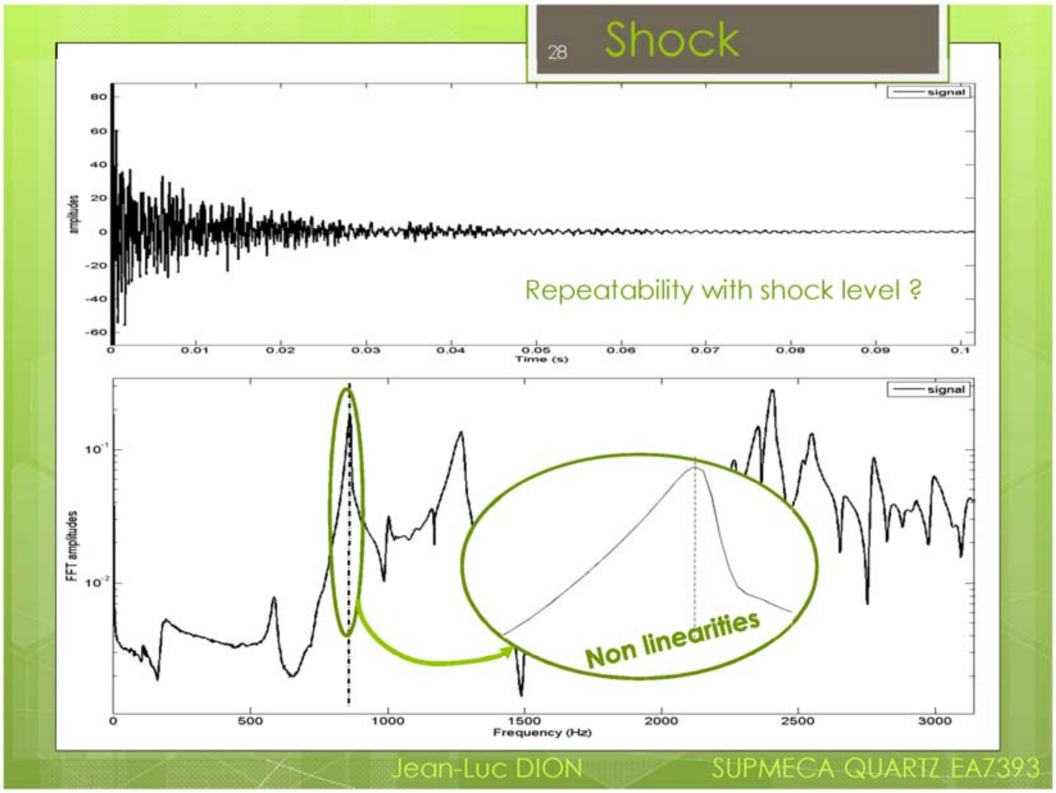
Application to joint components : predictability

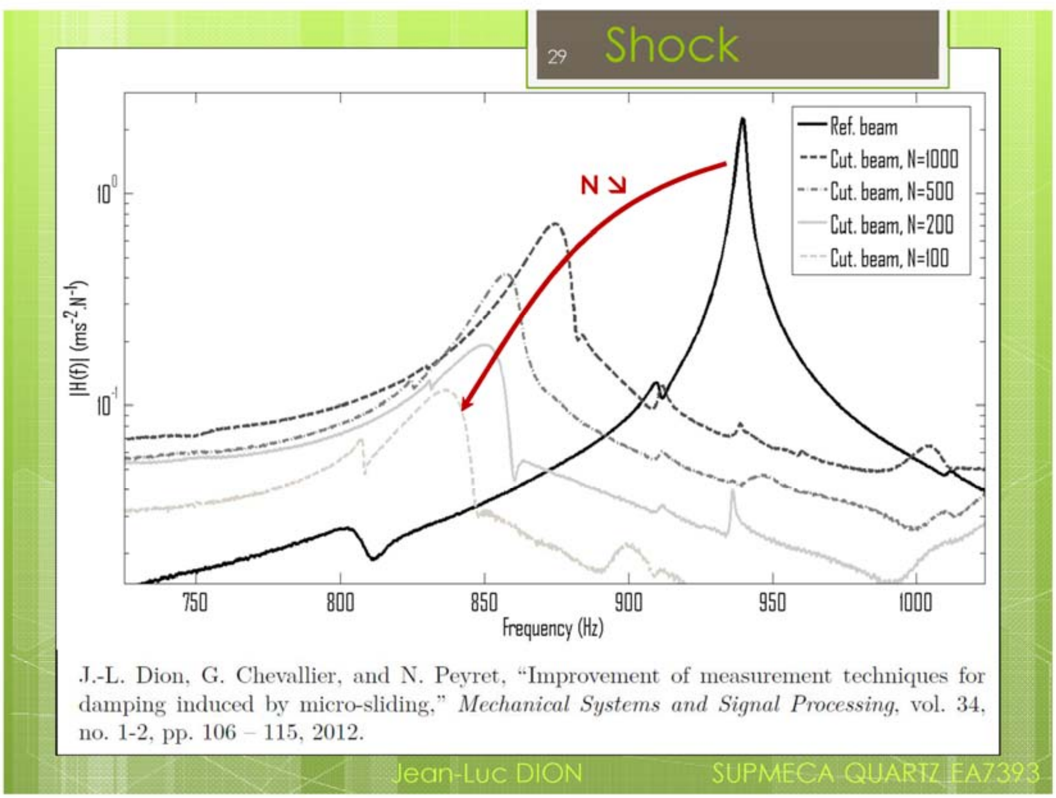


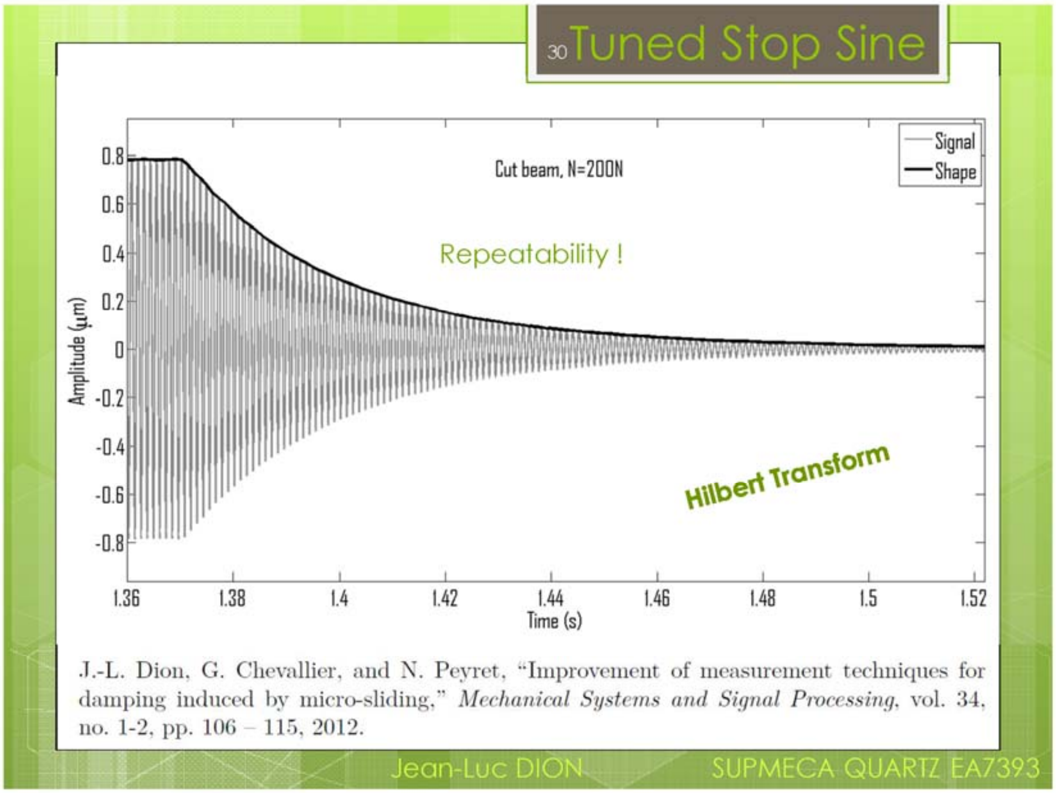


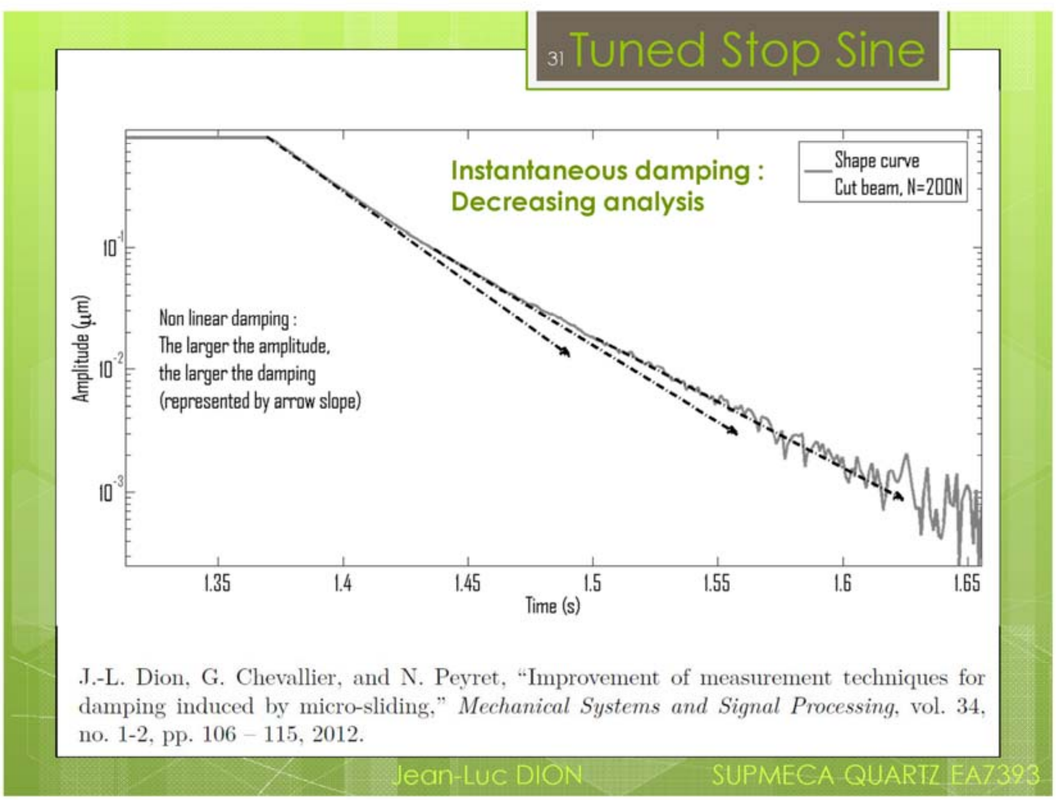

Repeatability
26
Vis de serrage
Balourds
Tronçons de tube
serial distributed joints
Non well-known contact zones
Geometric defaults
Jean-Luc DION
SUPMECA QUARTZ EA7393


Excitation ?
27
How the excitation
Impact repeatability ?
Frequency
Time
Natural mode
Dynamic Response
Shock (excitation)
Frequency
Time
Natural mode
Dynamic Response
Sweep sine (excitation)
Frequency
Time
Natural mode
Dynamic Response
Heaviside (excitation)
Frequency
Time
Natural mode
Dynamic Response
Stopped sine (Excitation)
J.-L. Dion, G. Chevallier, and N. Peyret, "Improvement of measurement techniques for damping induced by micro-sliding," Mechanical Systems and Signal Processing, vol. 34, no. 1-2, pp. 106 – 115, 2012.
Jean-Luc DION
SUPMECA QUARTZ EA7393


28
Shock
signal
amplitudes
Time (s)
Repeatability with shock level ?
signal
FFT amplitudes
Frequency (Hz)
Non linearities
Jean-Luc DION
SUPMECA QUARTZ EA7393


29
Shock
Ref. beam
Cut. beam, N=1000
Cut. beam, N=500
Cut. beam, N=200
Cut. beam, N=100
N ↘
|H(f)| (ms⁻².N⁻¹)
Frequency (Hz)
J.-L. Dion, G. Chevallier, and N. Peyret, "Improvement of measurement techniques for damping induced by micro-sliding," Mechanical Systems and Signal Processing, vol. 34, no. 1-2, pp. 106 – 115, 2012.
Jean-Luc DION
SUPMECA QUARTZ EA7393


30 Tuned Stop Sine
Signal
Shape
Cut beam, N=200N
Repeatability !
Hilbert Transform
Amplitude (μm)
0.8
0.6
0.4
0.2
0
-0.2
-0.4
-0.6
-0.8
1.36
1.38
1.4
1.42
1.44
1.46
1.48
1.5
1.52
Time (s)
J.-L. Dion, G. Chevallier, and N. Peyret, "Improvement of measurement techniques for damping induced by micro-sliding," Mechanical Systems and Signal Processing, vol. 34, no. 1-2, pp. 106 – 115, 2012.
Jean-Luc DION
SUPMECA QUARTZ EA7393


31 Tuned Stop Sine
Instantaneous damping :
Decreasing analysis
Shape curve
Cut beam, N=200N
Non linear damping :
The larger the amplitude,
the larger the damping
(represented by arrow slope)
Amplitude (μm)
10^-1
10^-2
10^-3
1.35
1.4
1.45
1.5
1.55
1.6
1.65
Time (s)
J.-L. Dion, G. Chevallier, and N. Peyret, "Improvement of measurement techniques for damping induced by micro-sliding," Mechanical Systems and Signal Processing, vol. 34, no. 1-2, pp. 106 – 115, 2012.
Jean-Luc DION
SUPMECA QUARTZ EA7393

32 Accurate Frequency & damping tracking

Predicted state $$\hat{X}_{n+1|n} = \hat{F}_{n|n}\hat{X}_{n|n}$$

Predicted estimated covariance $$\hat{P}_{n+1|n} = \tilde{F}_{n|n}\hat{P}_{n|n}\tilde{F}_{n|n}^T + Q$$

Innovation $$\tilde{Y}_{n+1|n} = Z_{n+1} - H\hat{X}_{n+1|n}$$

Innovation covariance $$S_{n+1} = H\hat{P}_{n+1|n}H^T + R$$

Kalman gain $$K_{n+1} = \hat{P}_{n+1|n}H^T\left(S_{n+1}\right)^{-1}$$

Updated state estimate $$\hat{X}_{n+1|n+1} = \hat{X}_{n+1|n} + K_{n+1}\tilde{Y}_{n+1|n}$$

Updated estimate covariance $$\hat{P}_{n+1|n+1} = \left(I - K_{n+1}H\right)\hat{P}_{n+1|n}$$

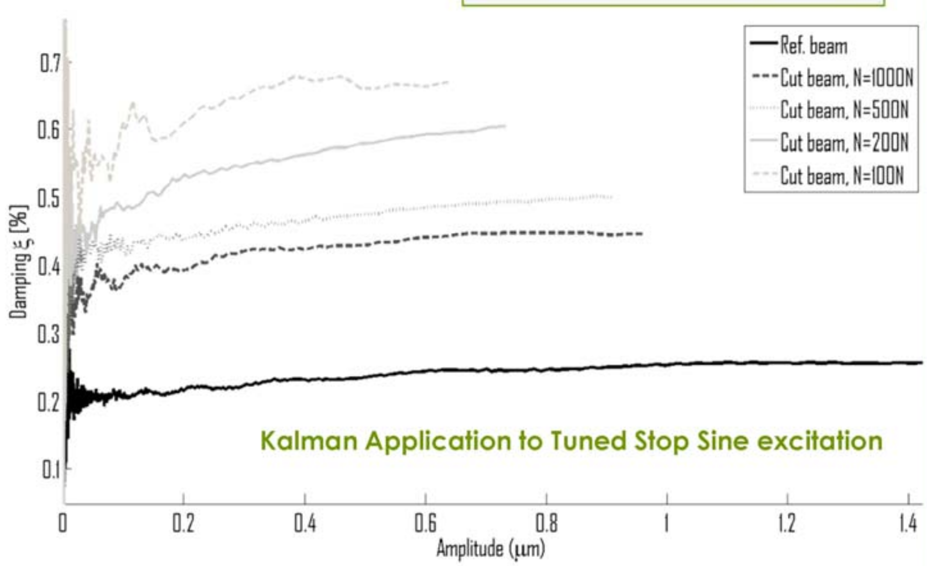
Jacobian: $$\tilde{F}_{n|n} = \left(\nabla_X \left(F(X)X\right)^T\right)^T\Big|_{X=\hat{X}_{n|n}}$$

$$F_n = \begin{pmatrix} cos(x_{3,n}) & -sin(x_{3,n}) & 0 \\ sin(x_{3,n}) & cos(x_{3,n}) & 0 \\ 0 & 0 & 1 \end{pmatrix} \qquad \tilde{F}_{n,i} = \begin{bmatrix} \cos(\hat{x}_{3i,n}) & -\sin(\hat{x}_{3i,n}) & -\hat{x}_{1+3(i-1),n}\sin(\hat{x}_{3i,n}) - \hat{x}_{2+3(i-1),n}\cos(\hat{x}_{3i,n}) \\ \sin(\hat{x}_{3i,n}) & \cos(\hat{x}_{3i,n}) & \hat{x}_{1+3(i-1),n}\cos(\hat{x}_{3i,n}) - \hat{x}_{2+3(i-1),n}\sin(\hat{x}_{3i,n}) \\ 0 & 0 & 1 \end{bmatrix}$$

J.-L. Dion, C. Stephan, G. Chevallier, and H. Festjens, "Tracking and removing modulated sinusoidal components : A solution based on the kurtosis and the extended kalman filter," *Mechanical Systems and Signal Processing*, vol. 38, no. 2, pp. 428 – 439, 2013.

Jean-Luc DION SUPMECA QUARTZ EA7393

33 Improved TSS



J.-L. Dion, C. Stephan, G. Chevallier, and H. Festjens, "Tracking and removing modulated sinusoidal components : A solution based on the kurtosis and the extended kalman filter," *Mechanical Systems and Signal Processing*, vol. 38, no. 2, pp. 428 – 439, 2013.

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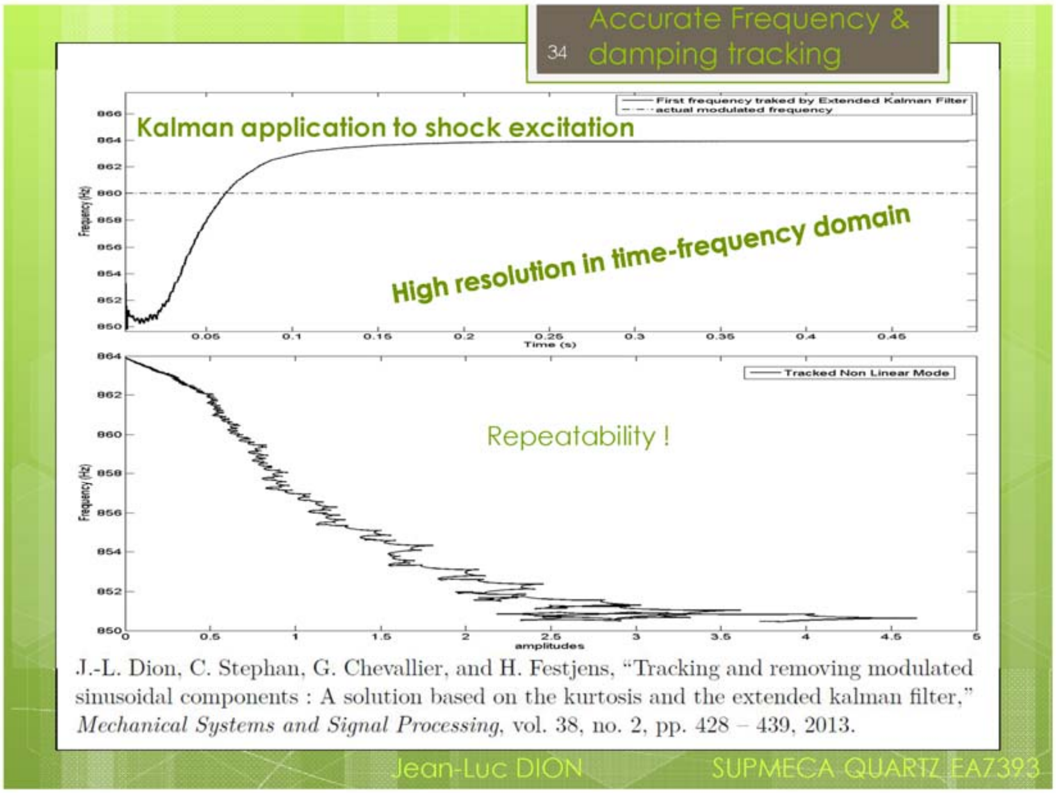
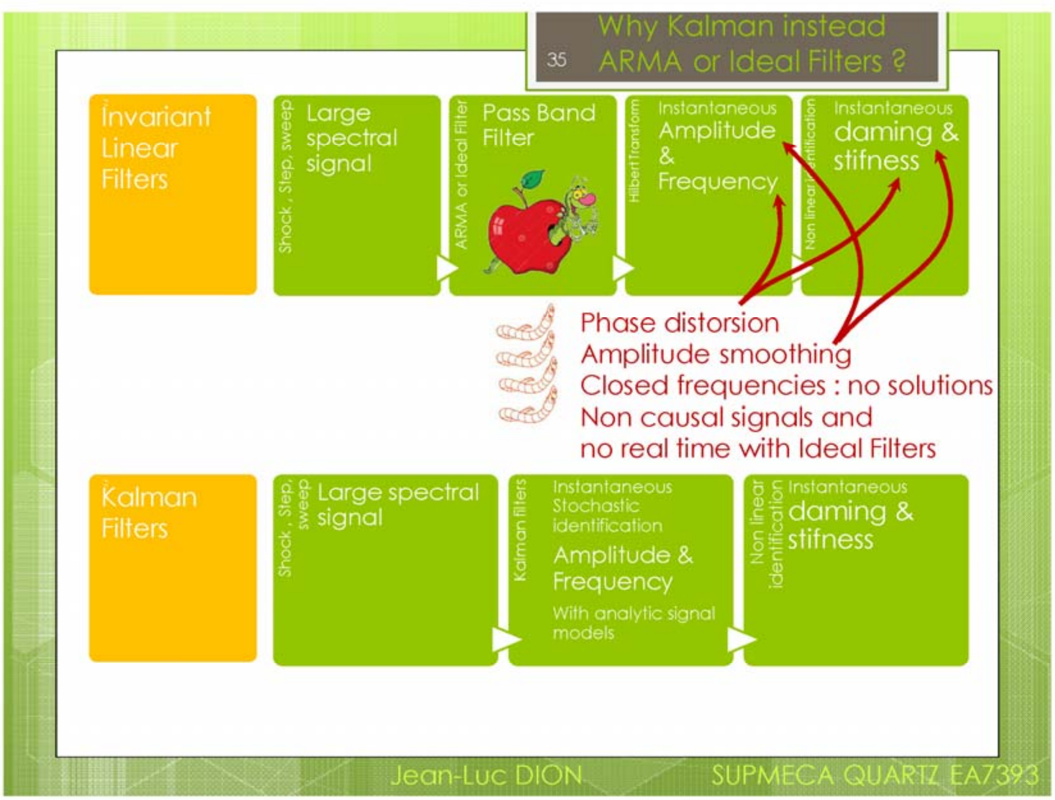
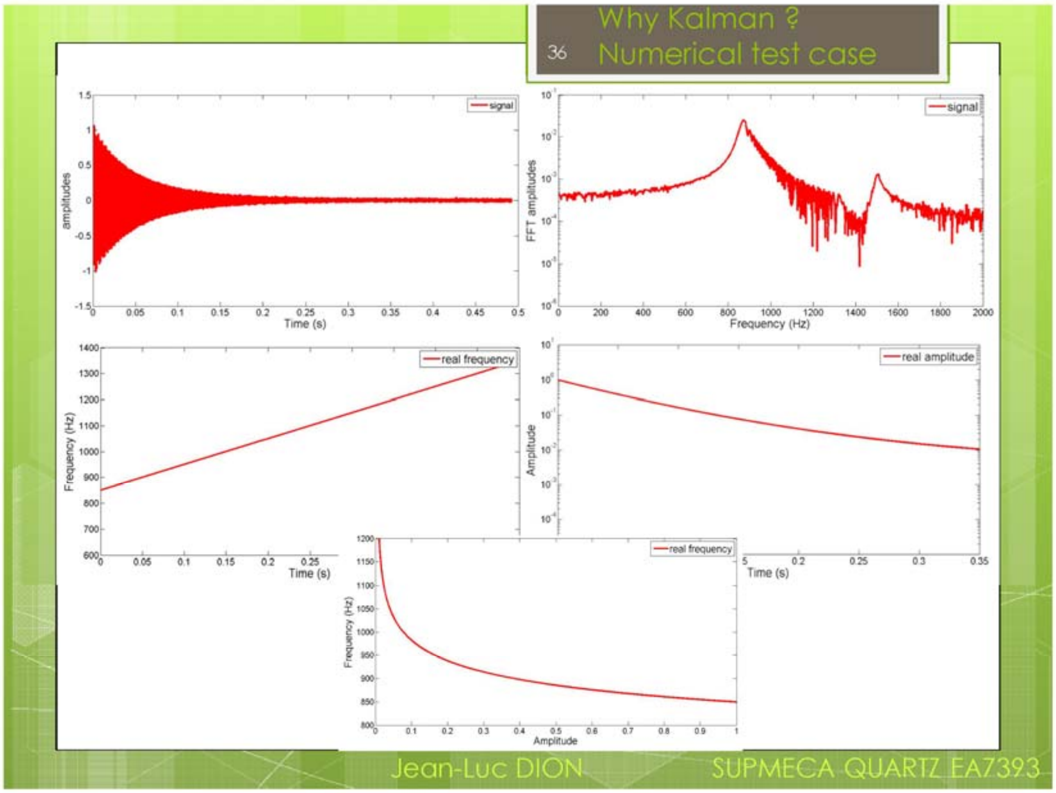
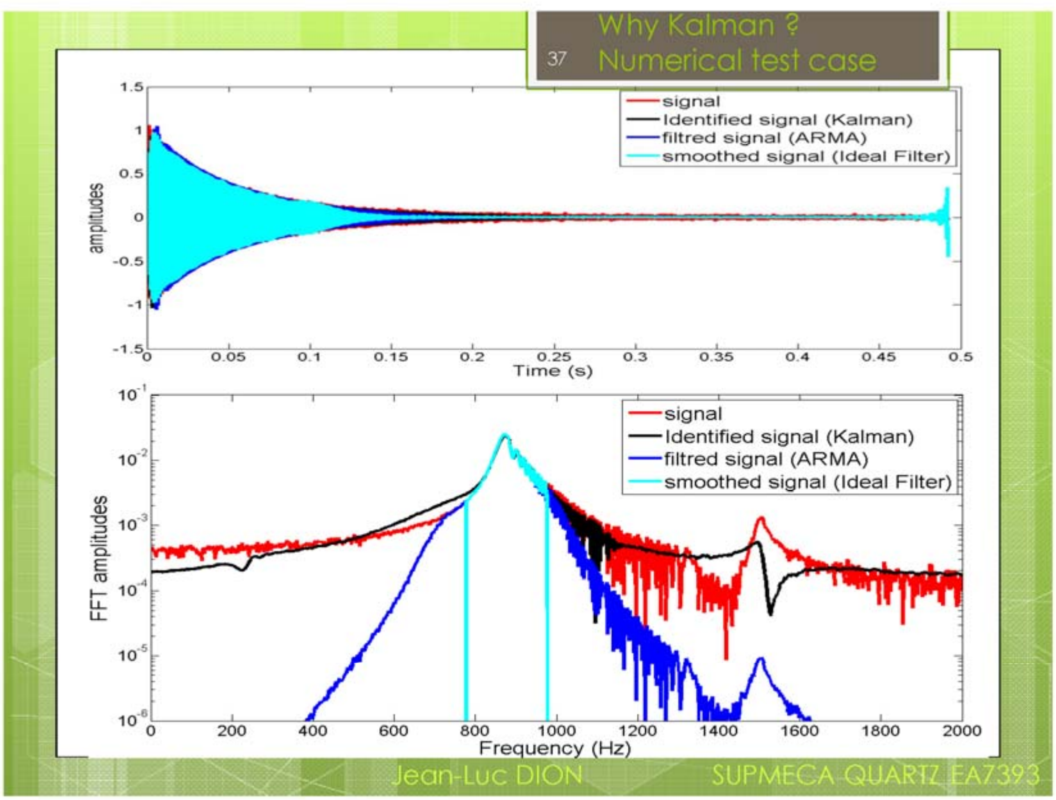
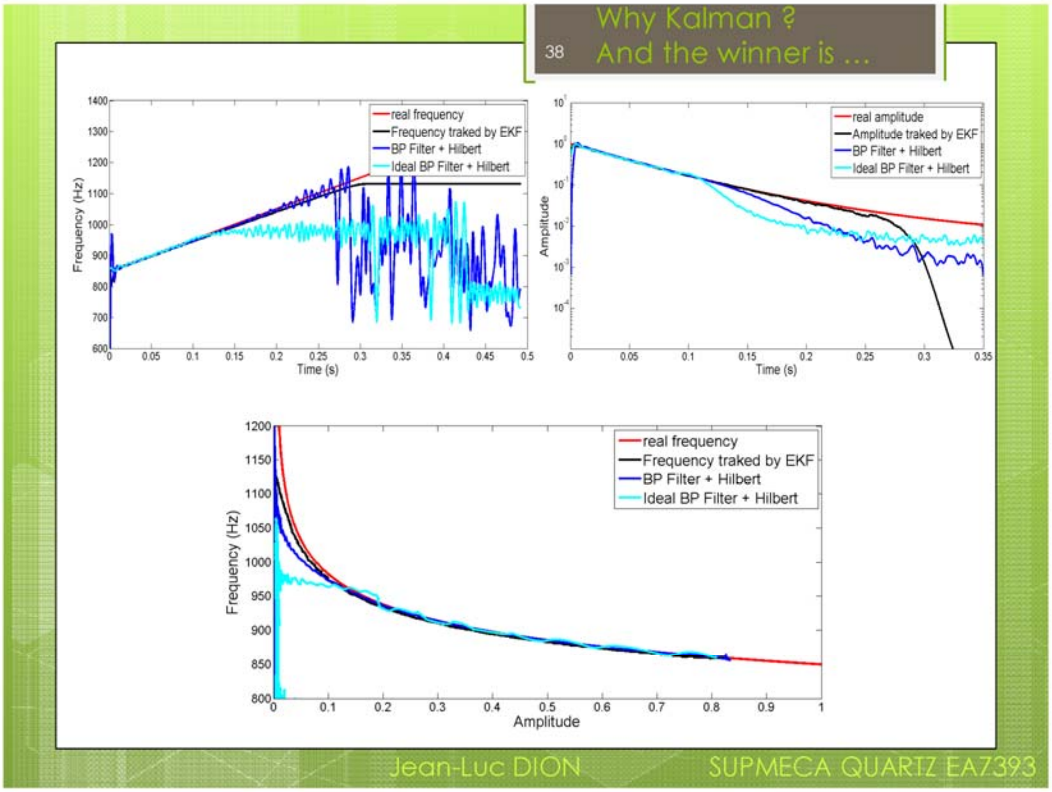
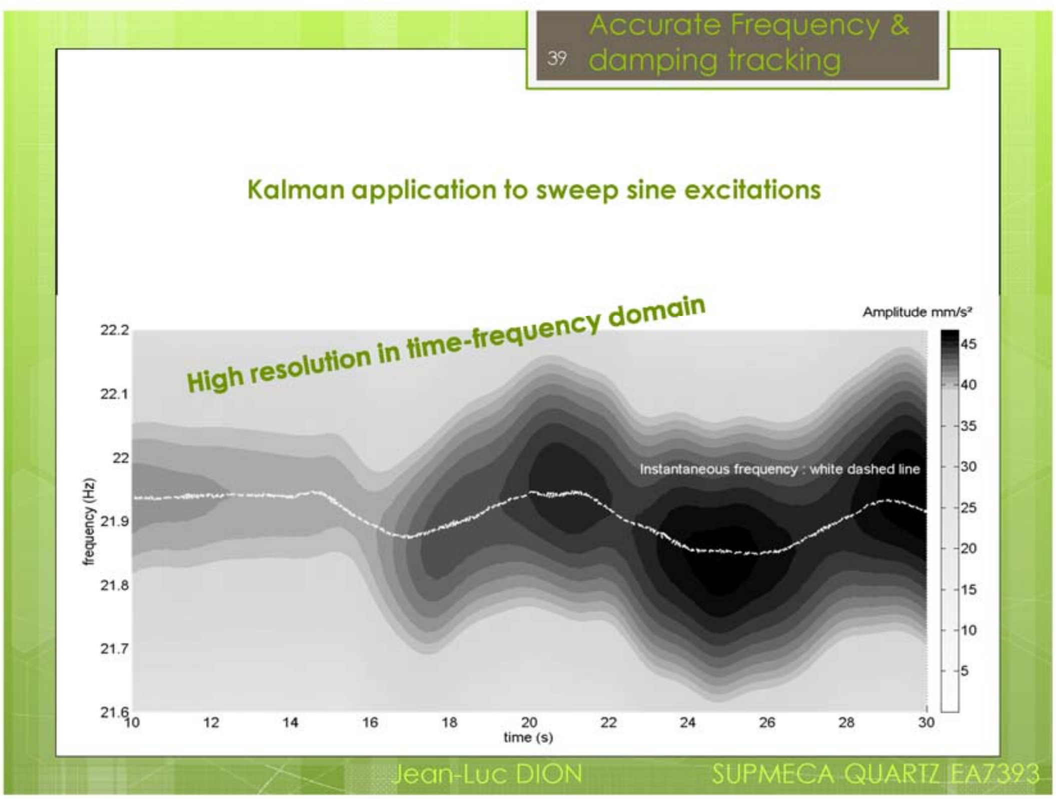

Accurate Frequency &
34 damping tracking
Kalman application to shock excitation
First frequency traked by Extended Kalman Filter
actual modulated frequency
High resolution in time-frequency domain
Frequency (Hz)
Time (s)
Tracked Non Linear Mode
Repeatability !
amplitudes
J.-L. Dion, C. Stephan, G. Chevallier, and H. Festjens, "Tracking and removing modulated sinusoidal components : A solution based on the kurtosis and the extended kalman filter," Mechanical Systems and Signal Processing, vol. 38, no. 2, pp. 428 – 439, 2013.
Jean-Luc DION
SUPMECA QUARTZ EA7393


Why Kalman instead
35 ARMA or Ideal Filters ?
Invariant Linear Filters
Shock , Step, sweep
Large spectral signal
ARMA or Ideal Filter
Pass Band Filter
Hilbert Transform
Instantaneous Amplitude & Frequency
Instantaneous daming & stifness
Phase distorsion
Amplitude smoothing
Closed frequencies : no solutions
Non causal signals and
no real time with Ideal Filters
Kalman Filters
Shock , Step, sweep
Large spectral signal
Kalman filters
Instantaneous Stochastic identification
Amplitude & Frequency
With analytic signal models
Non linear identification
Instantaneous daming & stifness
Jean-Luc DION
SUPMECA QUARTZ EA7393


Why Kalman ?
36 Numerical test case
signal
amplitudes
Time (s)
signal
FFT amplitudes
Frequency (Hz)
real frequency
Frequency (Hz)
real amplitude
Amplitude
real frequency
Frequency (Hz)
Amplitude
Jean-Luc DION
SUPMECA QUARTZ EA7393

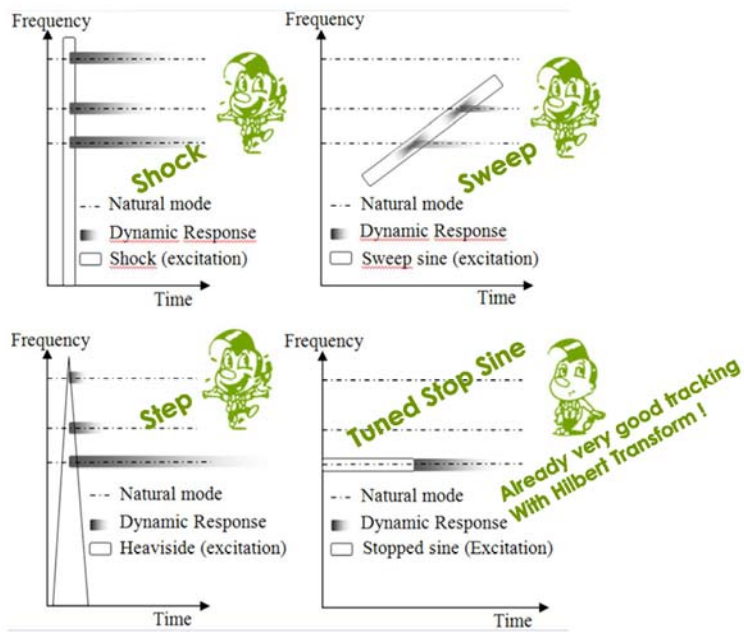

Why Kalman ?
37 Numerical test case
signal
Identified signal (Kalman)
filtred signal (ARMA)
smoothed signal (Ideal Filter)
amplitudes
Time (s)
signal
Identified signal (Kalman)
filtred signal (ARMA)
smoothed signal (Ideal Filter)
FFT amplitudes
Frequency (Hz)
Jean-Luc DION
SUPMECA QUARTZ EA7393


Why Kalman ?
38 And the winner is …
real frequency
Frequency traked by EKF
BP Filter + Hilbert
Ideal BP Filter + Hilbert
Frequency (Hz)
Time (s)
real amplitude
Amplitude traked by EKF
BP Filter + Hilbert
Ideal BP Filter + Hilbert
Amplitude
Time (s)
real frequency
Frequency traked by EKF
BP Filter + Hilbert
Ideal BP Filter + Hilbert
Frequency (Hz)
Amplitude
Jean-Luc DION
SUPMECA QUARTZ EA7393


Accurate Frequency &
39 damping tracking
Kalman application to sweep sine excitations
High resolution in time-frequency domain
Amplitude mm/s²
Instantaneous frequency : white dashed line
frequency (Hz)
time (s)
Jean-Luc DION
SUPMECA QUARTZ EA7393

40

Kalman filters for …



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41

Conclusion

- The linearity assumption is no longer valid for accurate measurements
- The usual descriptors are related to the linearity assumption

Improved Tuned Stop Sine

2 solutions to increase the accuracy of damping measurements :

- The implementation of Tuned Stop Sine excitation
- The use of Kalman filters provides an accurate parametric Identification of nonlinear models based on usual linear descriptors

Repeatability

- Parallel Distribution of joints
- Well known contact zones
- Geometric defaults under control

N. PEYRET, J-L. DION, G. CHEVALLIER « A framework for backbone experimental tracking : piezoelectric actuators, stop-sine signal and kalman filtering" MSSP 2015

Jean-Luc DION SUPMECA QUARTZ EA7393

Defence Academy of the United Kingdom

Cranfield UNIVERSITY

Some Measurements Illustrating Repeatability in Jointed Structures

Hugh Goyder
Cranfield University
United Kingdom

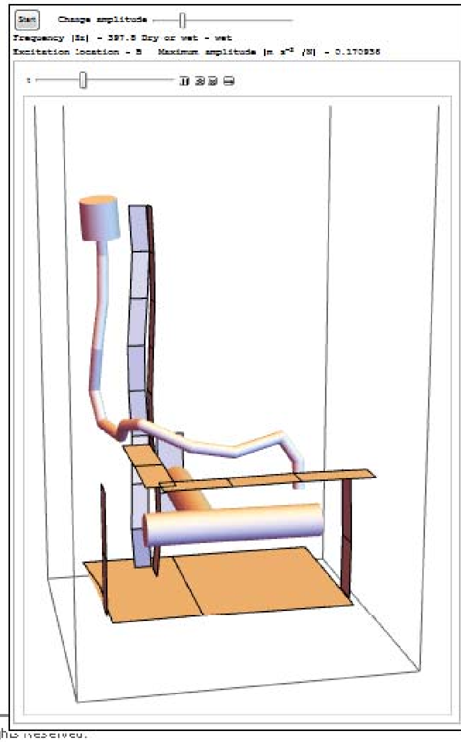
www.da.mod.uk

Vibration Test of Pipework

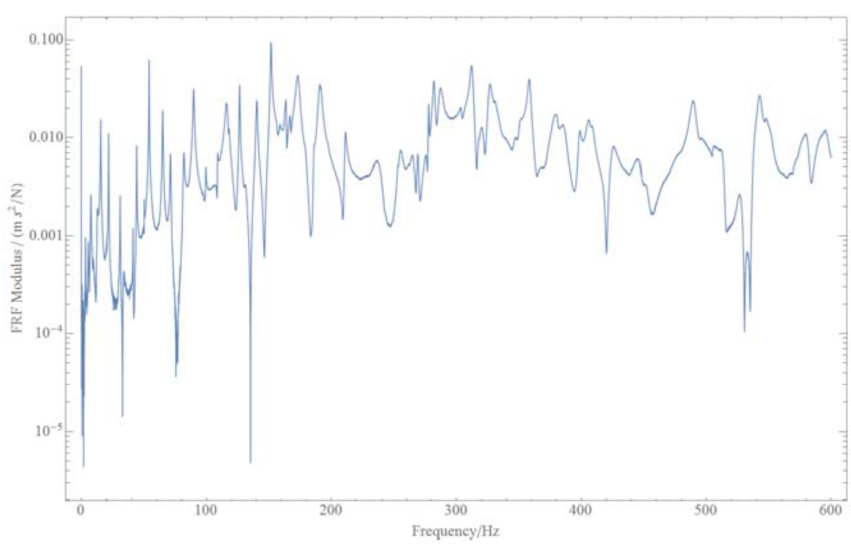
www.da.mod.uk

Pipework Joints

www.da.mod.uk



Example FRF (1 of 780)



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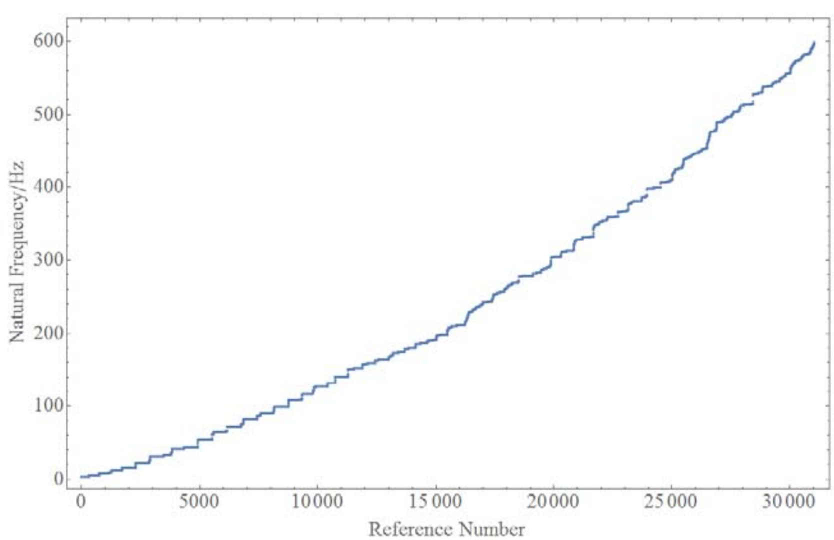
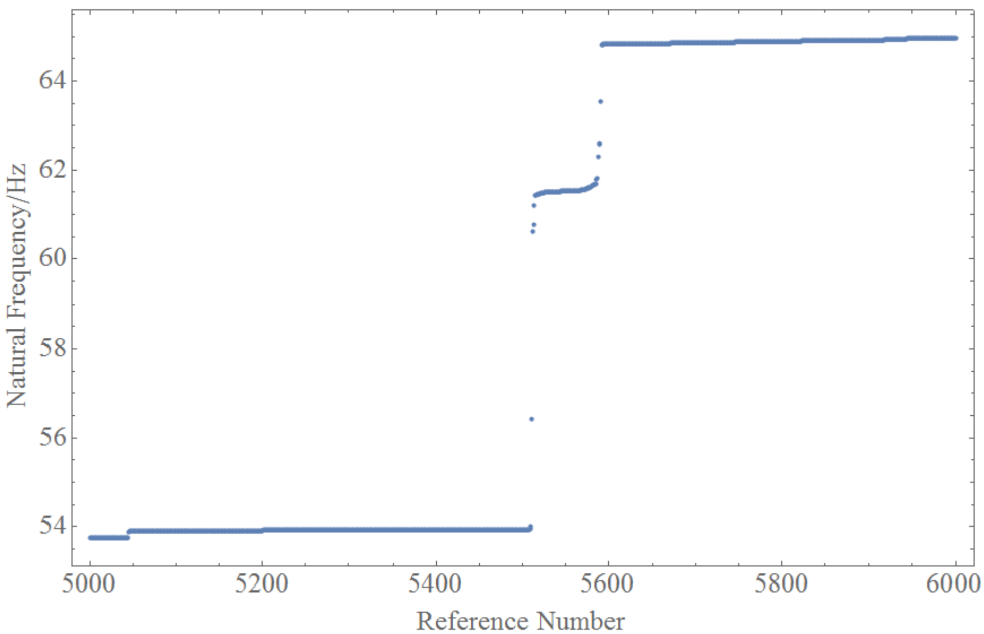
Model Fitting

- Fit all FRFs individually to determine natural frequencies and damping ratios.
- Reject poor fits
- 31037 values remain
- Sort in order of natural frequencies

Natural Frequencies





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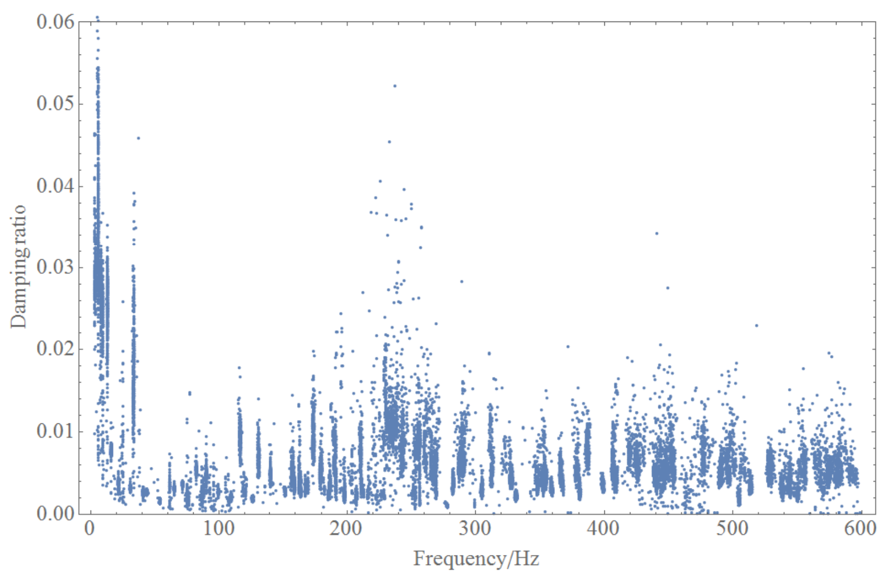
Damping and Natural Frequency



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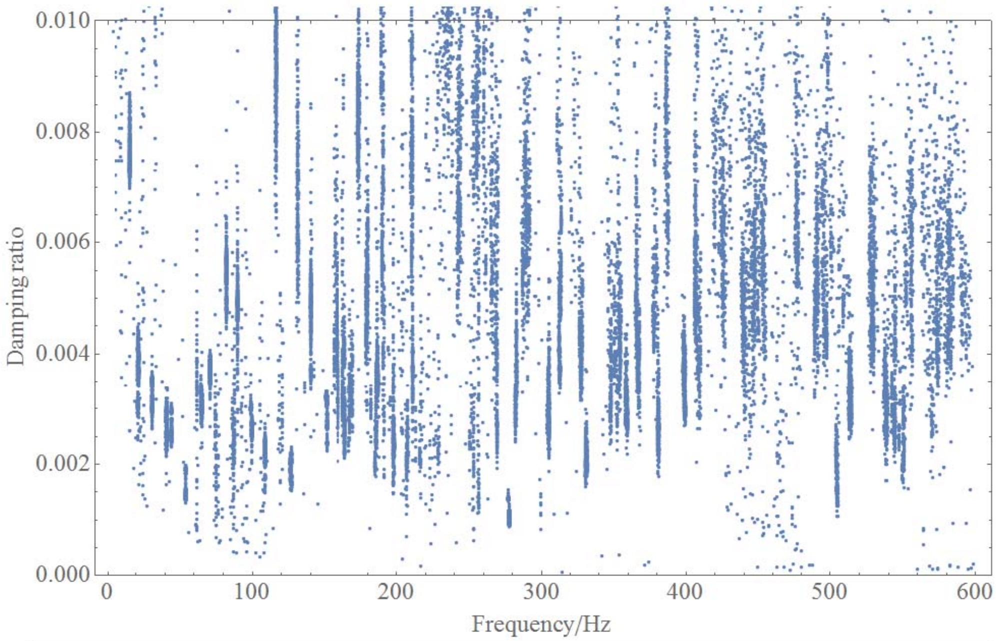
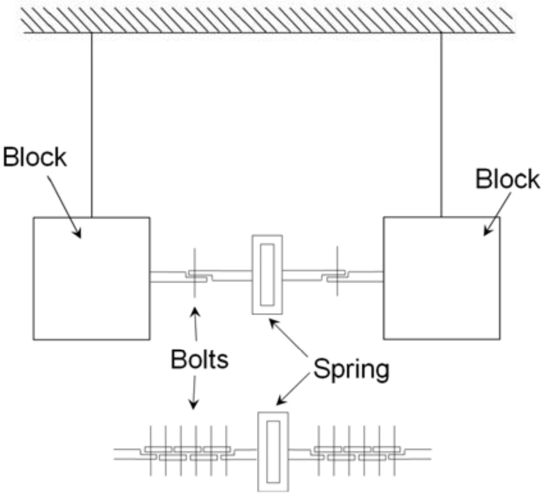


0.010
0.008
0.006
0.004
0.002
0.000
Damping ratio
0
100
200
300
400
500
600
Frequency/Hz

Damping in a Controlled Experiment



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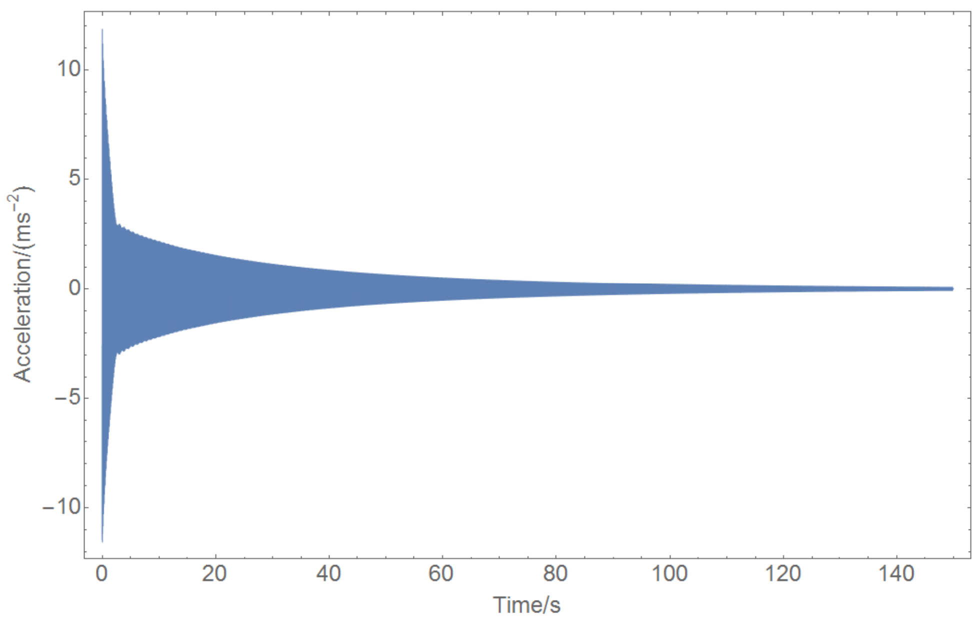
Vibration Decay



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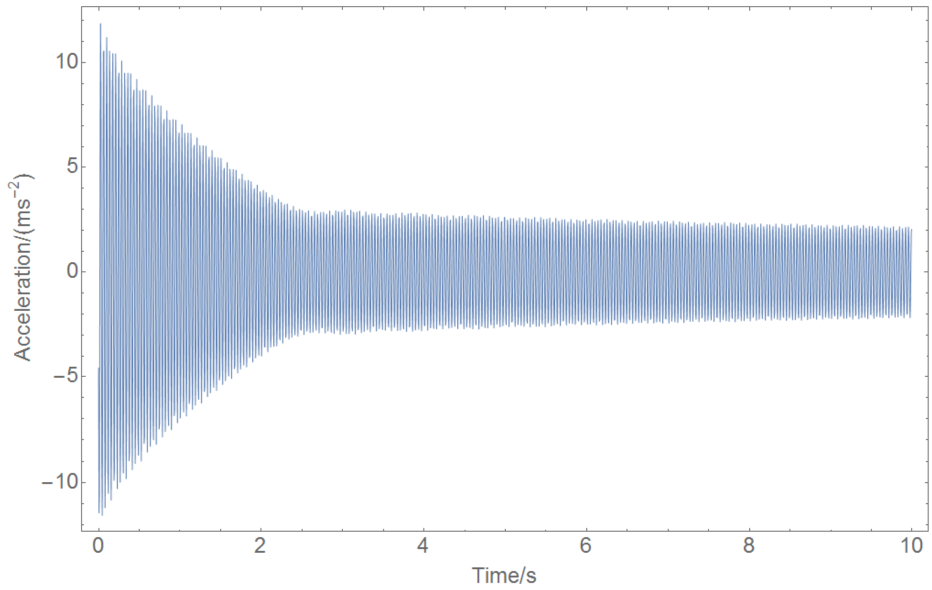
Vibration Decay



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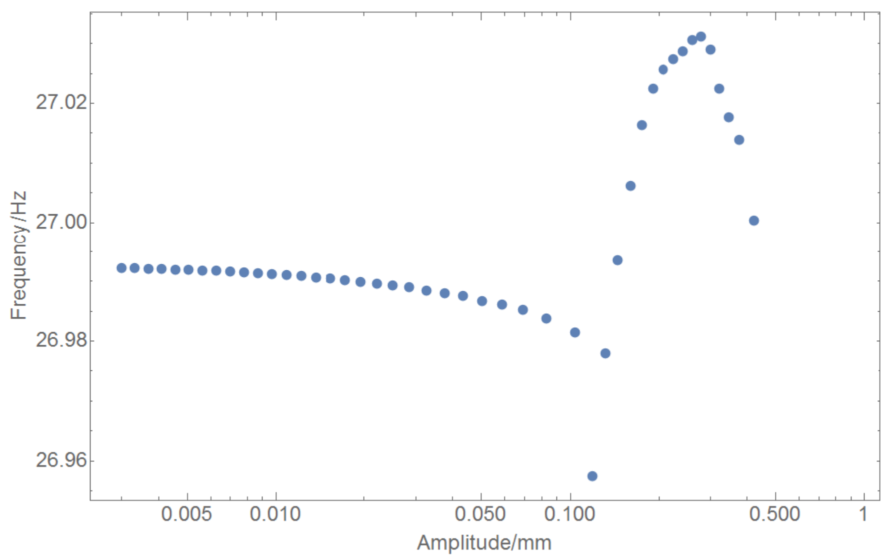
Change in frequency during decay



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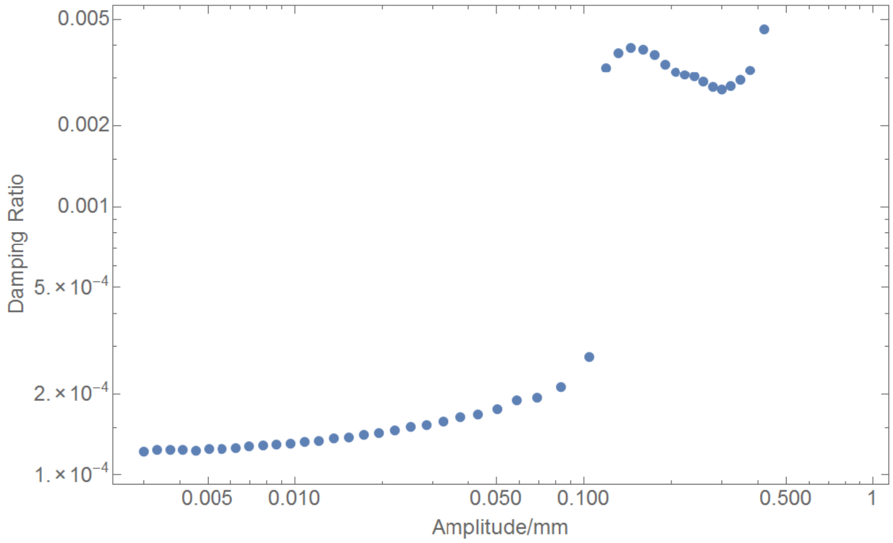
Change in damping during decay



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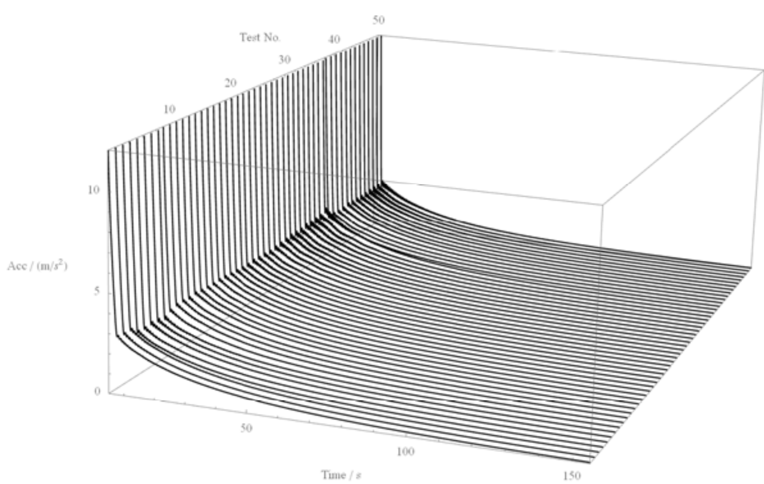
Repeatability



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Conclusions

- Non-repeatability of a built-up structure, particularly damping.
- Are we missing something – non-linear?
- Special methods needed to extract non-linear dependence of frequency and damping from joints.

Experimental Characterization of Joints in Aircraft Structures

Gaëtan Kerschen

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Workshop on Nonlinear System Identification Benchmarks

Brussels, Belgium, April 25-27, 2016

Home	Program	Bouc-Wen	Wiener-Hammerstein	Cascaded Tanks

Scope and Objectives

The objective of this workshop is to advance the current knowledge in nonlinear system identification by encouraging the exchange of ideas and the establishment of formal collaborations between the systems and control, mechanical and machine learning communities.

These three communities have developed over the years various and numerous nonlinear modeling approaches driven by the different backgrounds, constraints and end-uses. Moreover, they generally focus on different aspects of the modeling problem as they face different limiting factors in terms of model quality and identification cost. This is why we believe that, by promoting interaction, significant benefit can be mutually gained.

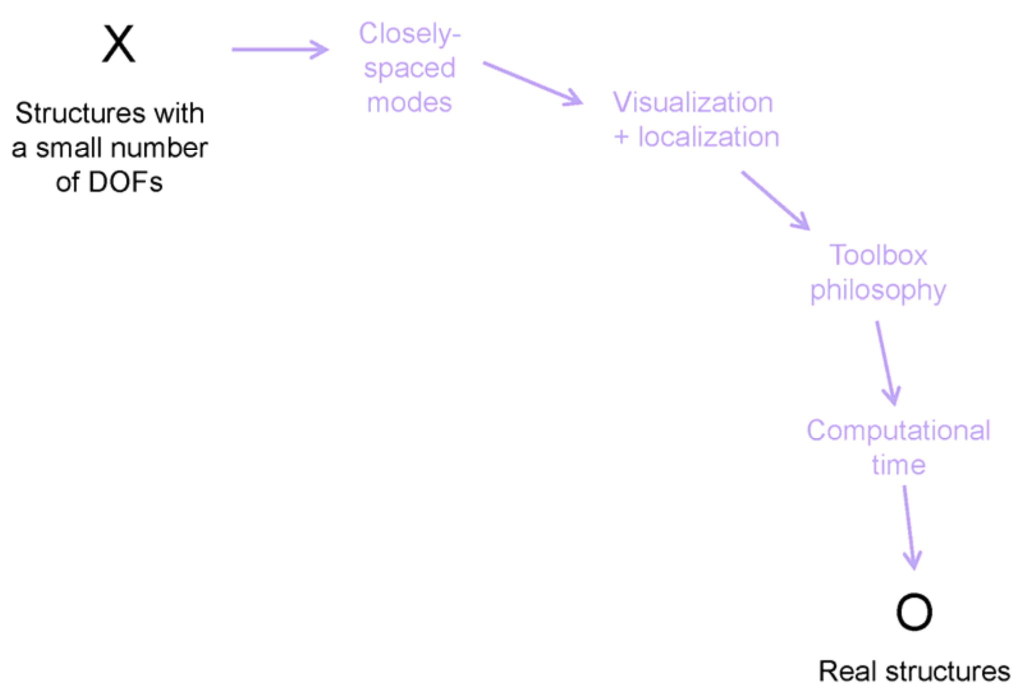
This workshop will be structured around three benchmark systems featuring state-of-the-art challenges in nonlinear system identification, namely dynamic nonlinearity, process noise, and short data record. They are a Bouc-Wen hysteretic system, a Wiener-Hammerstein system with process noise, and a cascaded tanks setup, respectively.

Solicited contributions should describe solutions to one or several of these benchmark problems. In particular, comparative overviews of methods would be particularly appreciated.

Participant Registration and Deadlines

Researchers wishing to participate in the workshop are invited to signal their interest as soon as possible via email. They will be kept informed of various benchmark developments and updates, in particular the Wiener-Hammerstein measurement opportunities.

Our Previous Strategy for Nonlinear System ID



3

We Are Missing Something

X

Structures with a small number of DOFs

Closely-spaced modes

Visualization + localization

Toolbox philosophy

Computational time

O

Real structures

4

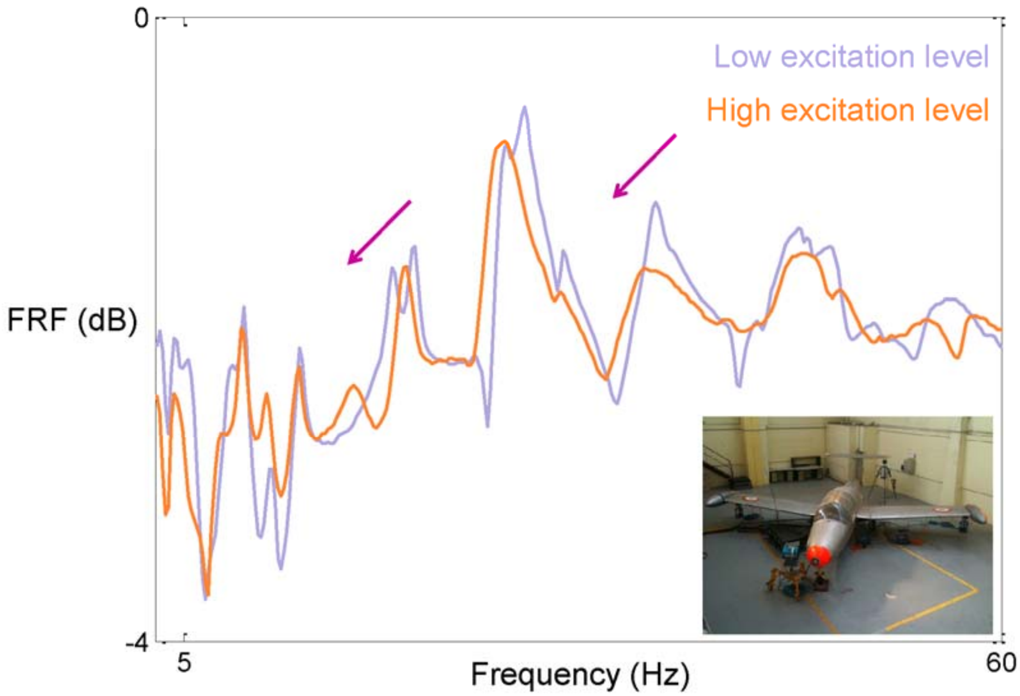
Two Full-Scale Aircraft Structures

Morane Saulnier 760 (ONERA, France)

F-16 (Saffraanberg, Belgium)

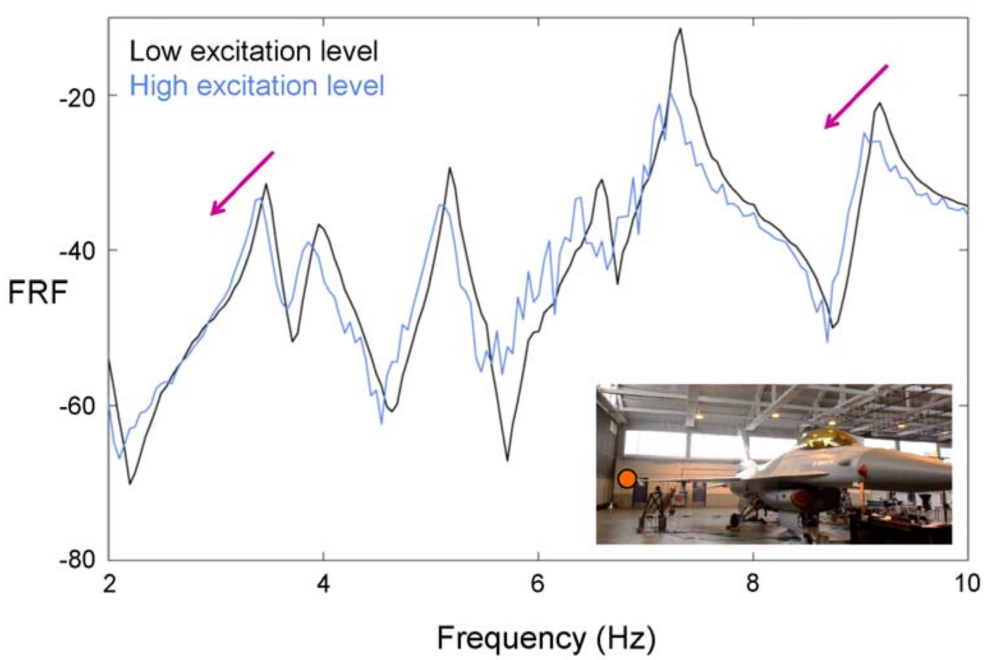
5

Evidence of Softening Nonlinear Behavior: MS760



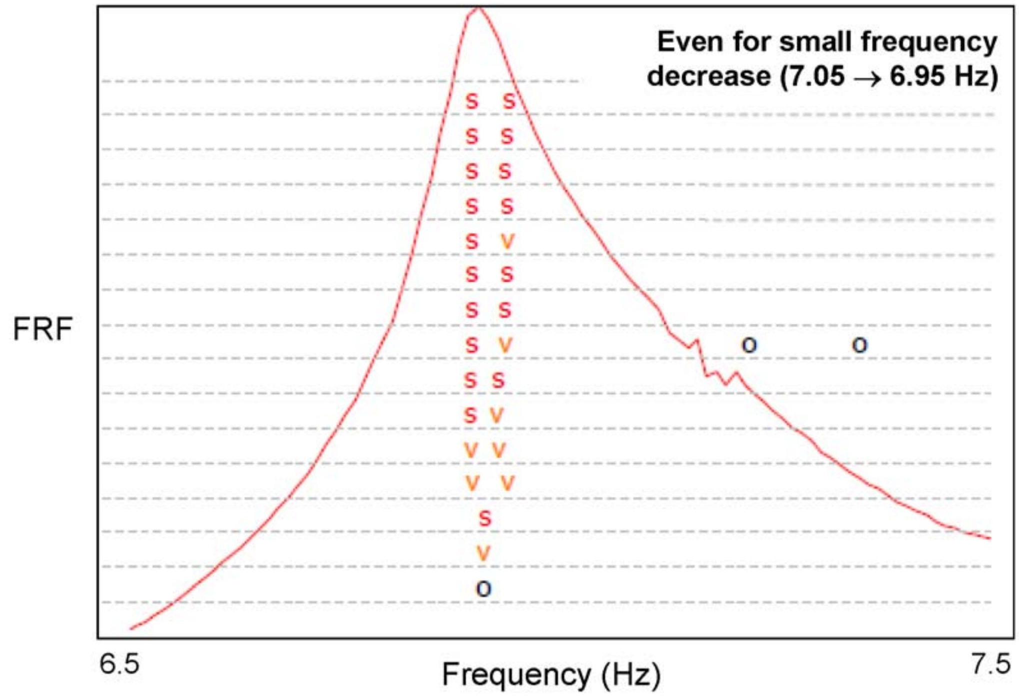
6

Evidence of Softening Nonlinear Behavior: F-16



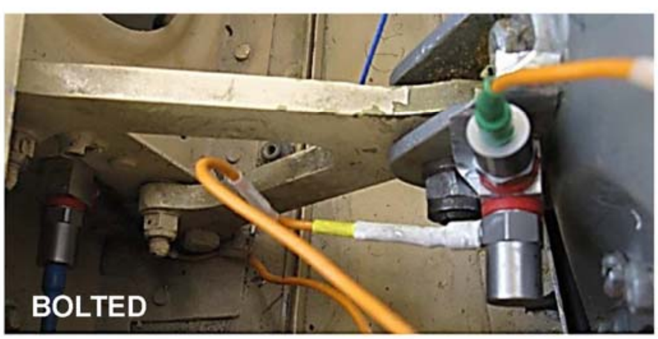
7

Failure of Classical Modal Analysis: F-16



8

What Are the Underlying Nonlinearities ?



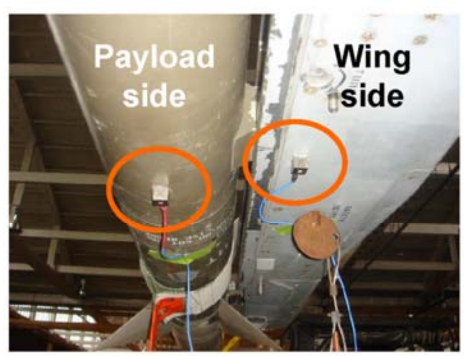
9

What Are the Underlying Nonlinearities ?

10

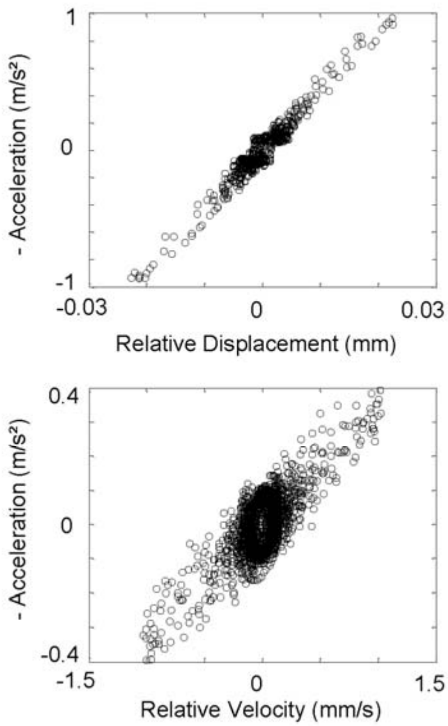
Can We Visualize the Nonlinearities ?

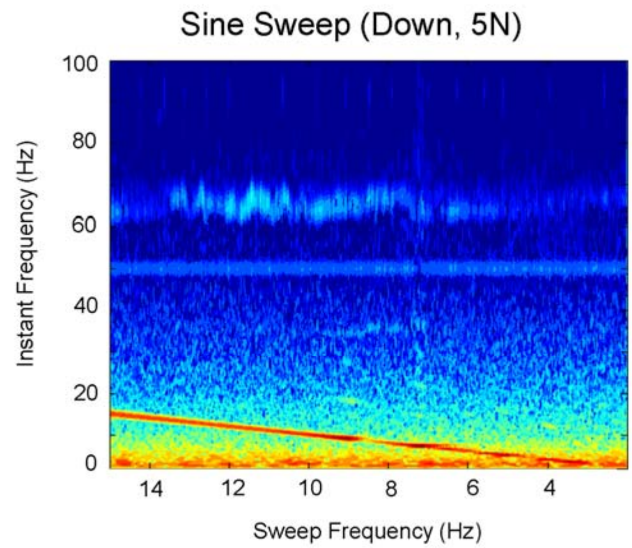
Accelerometers across the assumed nonlinear components + sine sweep excitation + modified restoring force surface.



11

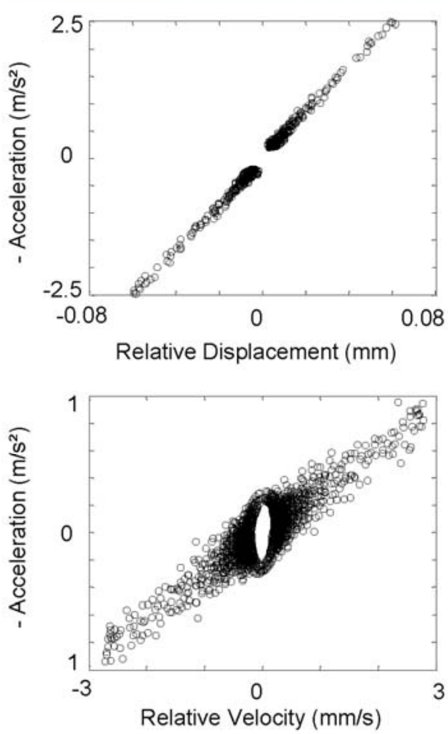
Linear Behavior at Low Excitation Level (F-16)

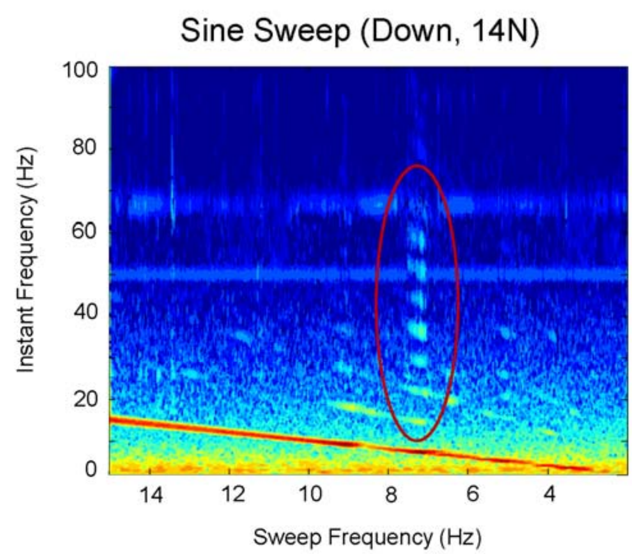




12

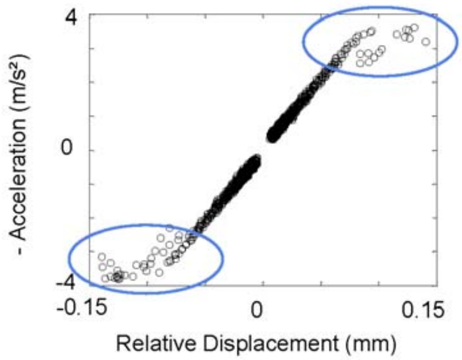
Richer Frequency Content at Moderate Levels

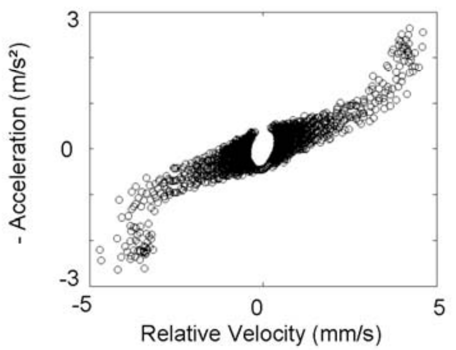


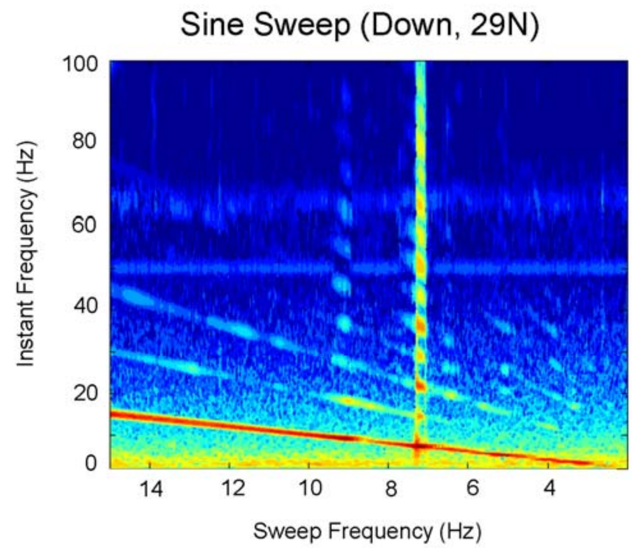


13

Richer Frequency Content at Moderate Levels

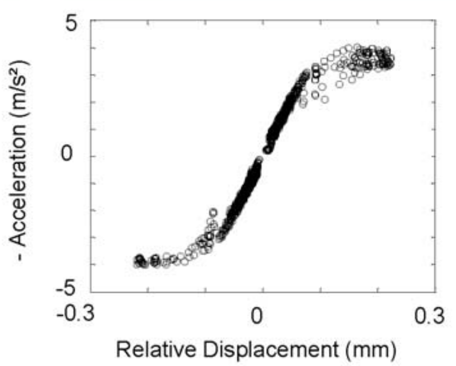


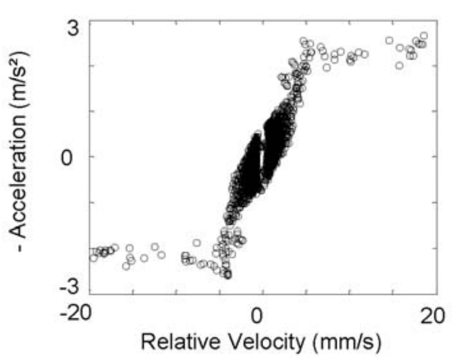


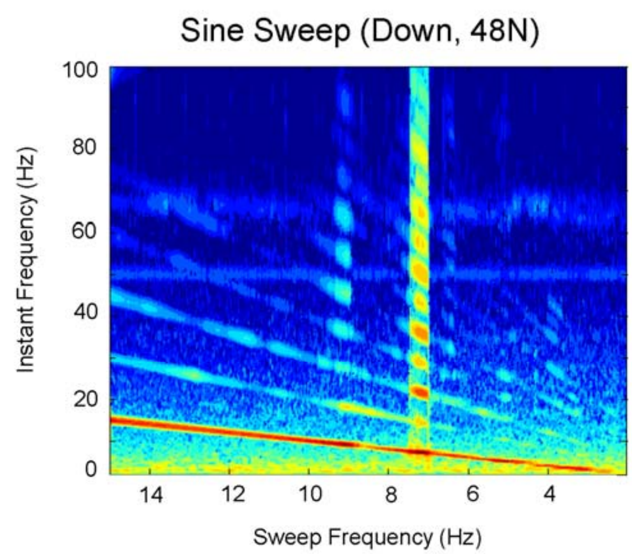


14

Softening and Coulomb Friction at Higher Levels







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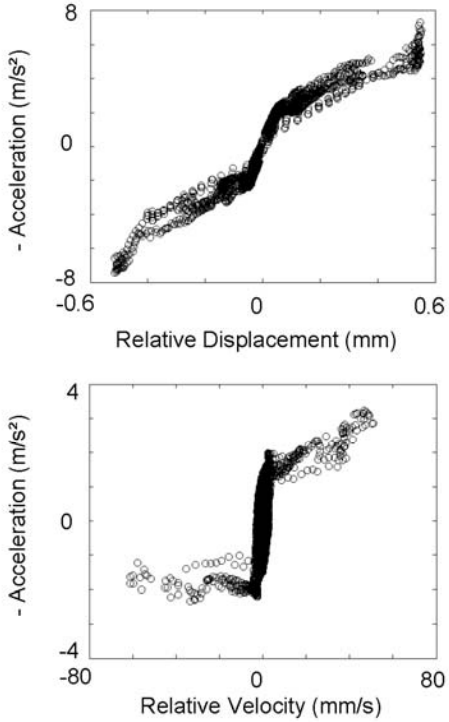
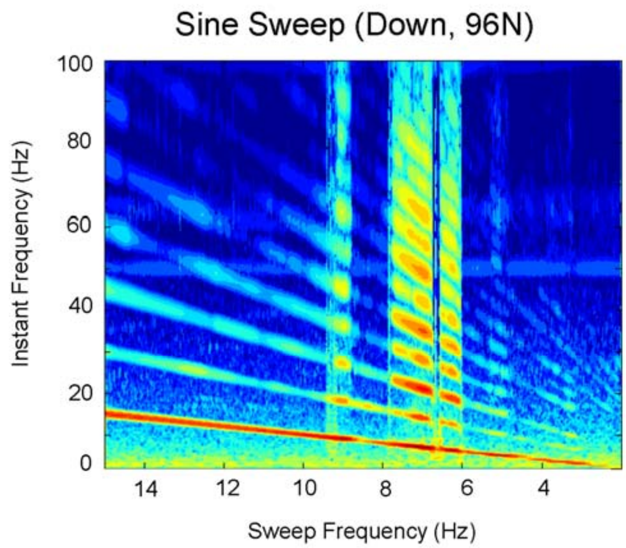
Softening-Hardening and Coulomb Friction

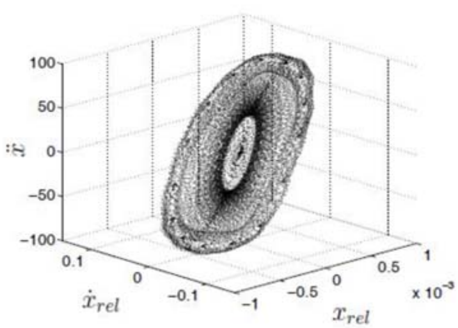


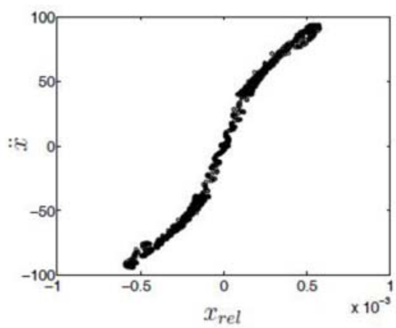


16

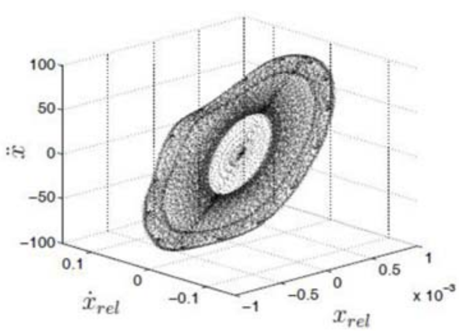
Nonsmooth, Softening Behavior (MS760)

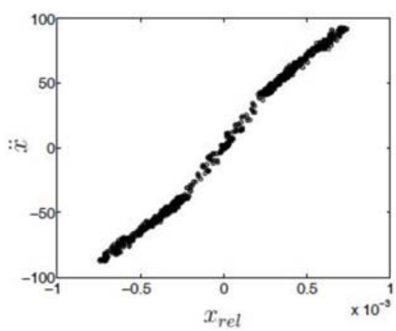
LEFT WING





RIGHT WING





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Nonlinear System ID (Black-box Type of Approach)

	Linear Id. at Low Level		Linear Id. at High Level		Discrepancy		
Mode	f_n (Hz)	ζ_n (%)	f_n (Hz)	ζ_n (%)	δ_f (%)	δ_ζ (%)	MAC (%)
1	9.02	0.30	8.92	0.44	1.06	46.49	99.1
2	10.64	0.78	10.48	1.30	1.51	68.13	98.8
3	12.75	0.67	12.60	0.71	1.12	6.59	39.8
4	18.72	1.33	17.25	0.99	7.85	25.28	75.8
5	20.23	0.88	19.85	1.13	1.91	28.43	96.7

Before vs. After ID

BEFORE

	Linear Id. at Low Level		Linear Id. at High Level		Discrepancy		
Mode	f_n (Hz)	ζ_n (%)	f_n (Hz)	ζ_n (%)	δ_f (%)	δ_ζ (%)	MAC (%)
1	9.02	0.30	8.92	0.44	1.06	46.49	99.1
2	10.64	0.78	10.48	1.30	1.51	68.13	98.8
3	12.75	0.67	12.60	0.71	1.12	6.59	39.8
4	18.72	1.33	17.25	0.99	7.85	25.28	75.8
5	20.23	0.88	19.85	1.13	1.91	28.43	96.7

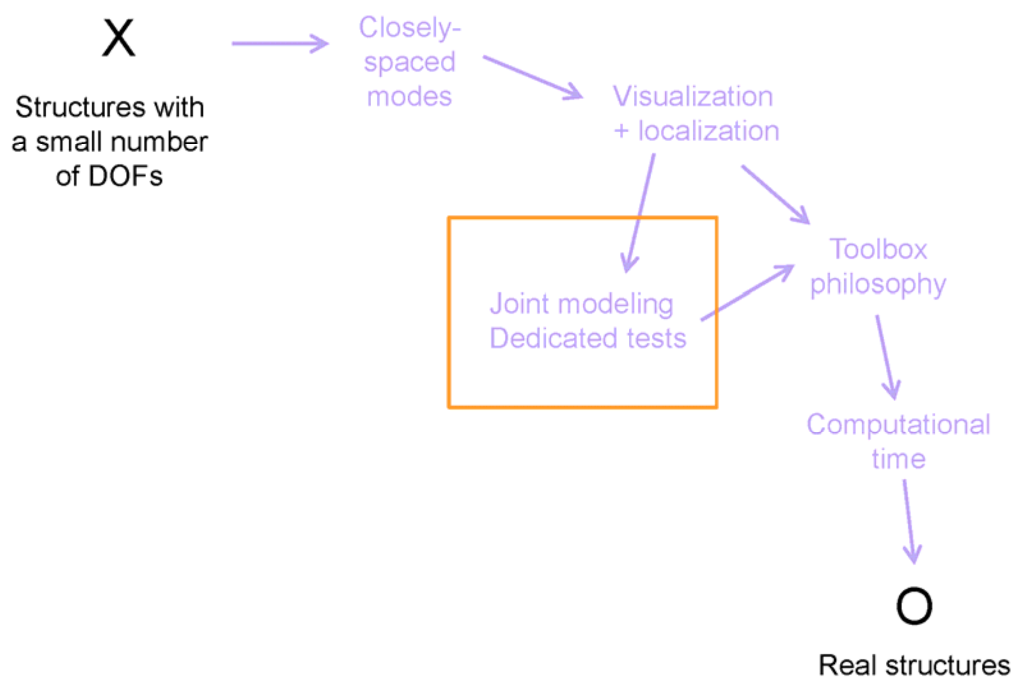
Much better
(our target)

Much
worse !

AFTER

	Linear Id. at Low Level		Nonlinear Id. at High Level		Error		
Mode	f_n (Hz)	ζ_n (%)	f_n (Hz)	ζ_n (%)	ϵ_f (%)	ϵ_ζ (%)	MAC (%)
1	9.02	0.30	8.93	0.49	0.95	64.21	99.2
2	10.64	0.78	10.70	0.94	0.50	21.16	96.3
3	12.75	0.67	12.62	0.67	0.97	0.45	36.4
4	18.72	1.33	18.75	0.18	0.17	86.72	11.3
5	20.23	0.88	20.05	0.20	0.92	77.74	56.4

I Am Bringing Coal to Newcastle



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Concluding Remarks

1. Different joints (bolted vs. sliding), yet same overall impact, i.e., softening nonlinearity, increase in damping.

2. System ID is helpful, if not mandatory, but probably not viable on its own. (black box approach)

3. Joint modeling should be included in the dynamicist's toolbox if real structures are to be addressed. (white box approach)

4. We need a grey box approach (physics + flexibility)

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Thank you for your attention.

Gaëtan Kerschen

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Belgium

5.5: Session 3: Predictability

David Hills: Bringing the Joints and Contacts Communities Together

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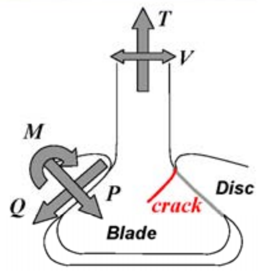
Bringing the Joints and Contacts Communities together

David A. Hills

October 2015

Some Gas turbine contacts (or Joints)

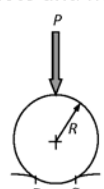
Dovetail joint | Fir-tree joint | Spline joint



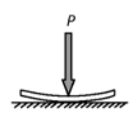
D.A. Hills 2

Idealized types of contacts

1. Incomplete and non-conformal

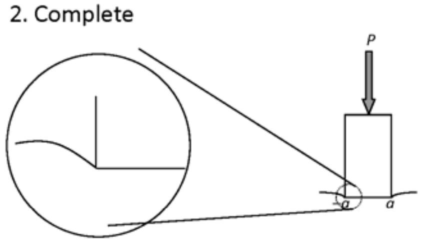


3. Receding

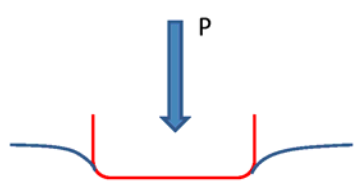


4. 'Common Edge'

2. Complete



wedge theory



D.A. Hills 3

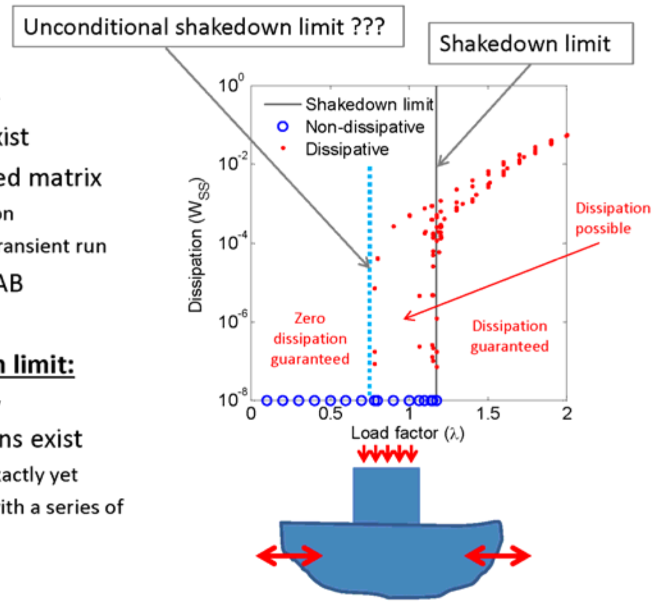
Frictional Shakedown

Shakedown limit:

- Load above which only dissipative solutions exist
- Calculated from reduced matrix
  - Frame as an optimization
  - Calculated as fast as a transient run
- Implemented in MATLAB

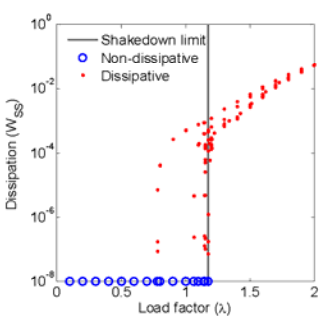
Unconditional shakedown limit:

- Load below which only non-dissipative solutions exist
  - Cannot be calculated exactly yet
  - Can be approximated with a series of transient runs

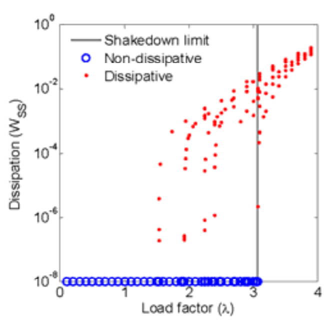


Frictional Energy Dissipation

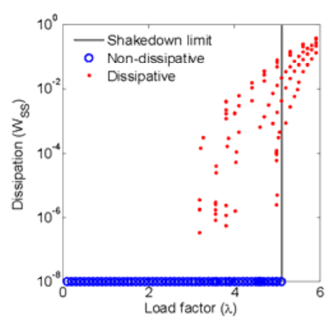
- Steady state dissipation is sensitive to the initial residual stress state



f = 0.3



f = 0.6



f = 0.9

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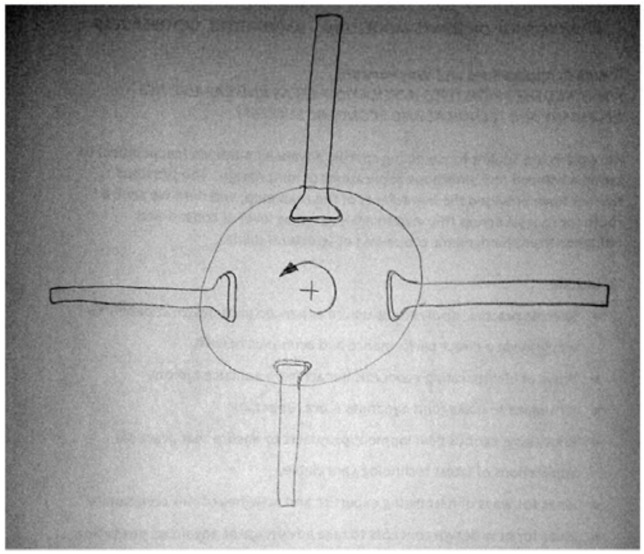
Four Fanblades in a disk



D.A. Hills 6

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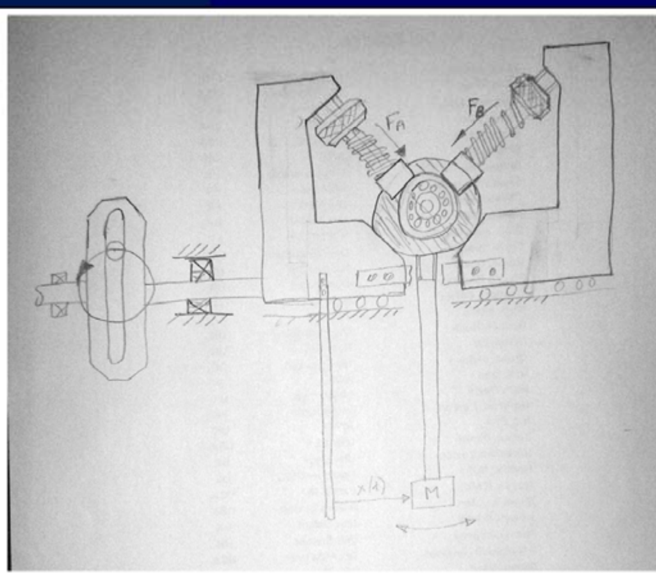
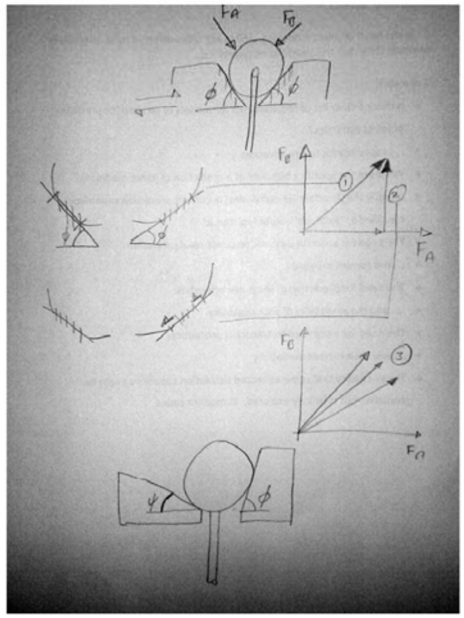
A Test Apparatus ?



D.A. Hills 7

Effect of Clamping Sequence

Shakedown:
Do experiments to measure damping and see if it *is* very dependent on prior conditions.



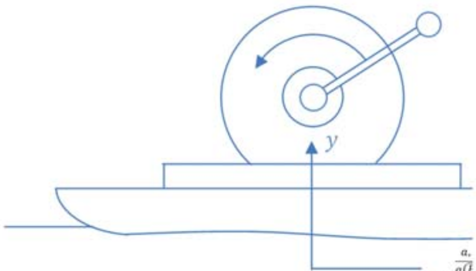
Damping

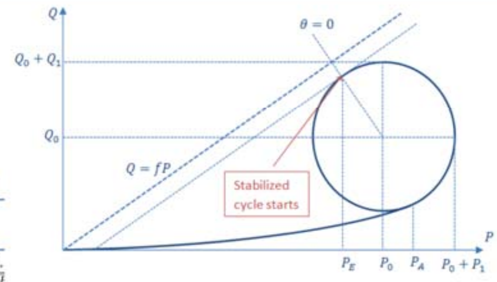
1. In a coupled, frictionally damped system, the 'damping coefficient' is *not* a system property.
2. It depends on the nature of the loading;

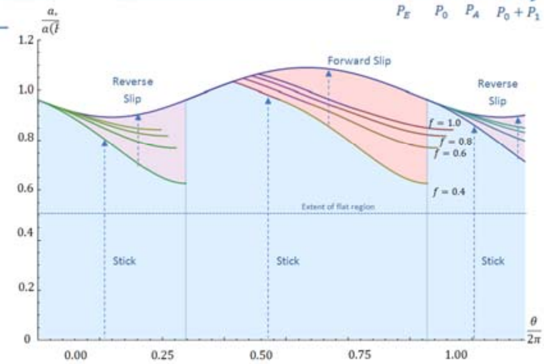
Here we want it for periodic excitation.

It is *not* related to the damping 'coefficient' revealed by a 'striking' test.

Introduction: Mat Davies' Solution







November 2014

PREDICTING UNCERTAINTY IN JOINTS BEHAVIOR

Marc P. Mignolet
SEMTE, Faculties of Mechanical and Aerospace Engineering
Arizona State University

Ira A. Fulton Schools of Engineering
SEMTE, Faculties of Mechanical and Aerospace Engineering
Structural Dynamics Group

Predicting Uncertainty ?

* Lack of repeatability in joints behavior is well recognized.
Cast doubts on being able to obtain *single* predictions…

* What is the next best thing?
Predicting the *domain* (band) in which the predictions are. This is predicting the uncertainty…

* How is that done?
(i) define band directly (e.g., from measurement data)
(ii) use a joint model (e.g., Iwan model) in which uncertainty is introduced (input). Here through randomness (probabilistic approach). Band on behavior is output from model.

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EPISTEMIC VS. ALEATORIC UNCERTAINTY

<u>Epistemic (or model or reducible) uncertainty</u>
Observed when the response of the system cannot be matched by the model predictions irrespectively of the model parameters, e.g. curved beam modeled by a straight one, nonlinear system represented by a linear one, …

<u>Aleatoric (or parameter or irreducible) uncertainty</u>
Observed when the response of the system can be matched by the model predictions for an appropriate choice of the parameters which is different for different structures, e.g. random Young' modulus

Improving the model tends to reduce epistemic uncertainty but increase aleatoric uncertainty

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ON MODELING AND UNCERTAINTY

I. Model "type" affects the balance of epistemic/aleatoric uncertainty

Detailed (finite element) model:

Aleatoric uncertainty can be introduced only in the "mechanical/ material" properties and/or geometry

Global/reduced order/phenomenological model:

Parameters regroup many features of the problem and geometry. Aleatoric uncertainty includes some uncertainty epistemic in the detailed model

II. A very refined model may not be necessary in the presence of aleatoric uncertainty. Epistemic uncertainty can be present as long as it does not affect (increase) significantly the band of predictions.

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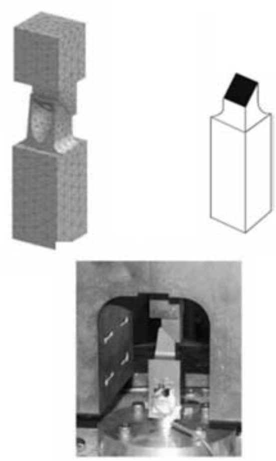
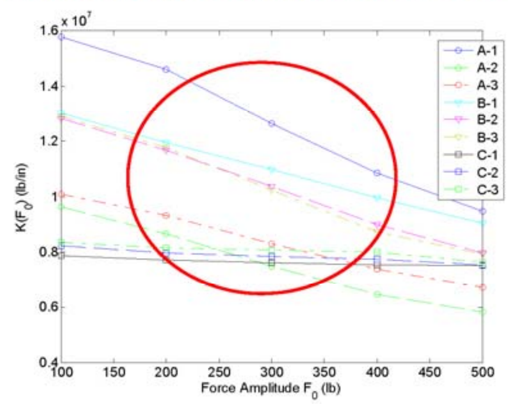
A DISCUSSION PROBLEM

Test and Data from Sandia Joints Handbook: 9 "identical" bolted joints



ALEATORIC UNCERTAINTY!!



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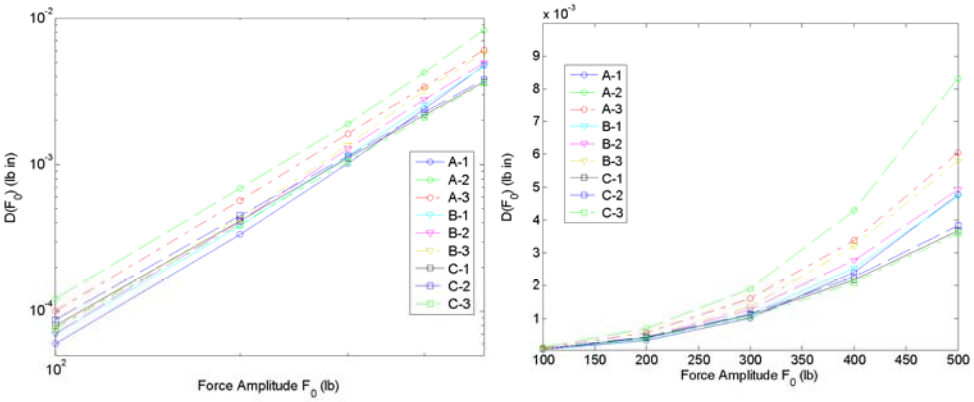
A DISCUSSION PROBLEM

Test and Data from Sandia Joints Handbook: 9 “identical” bolted joints

Dissipation data also recorded and also shows aleatoric uncertainty



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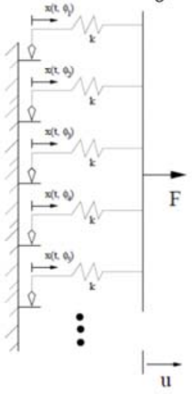
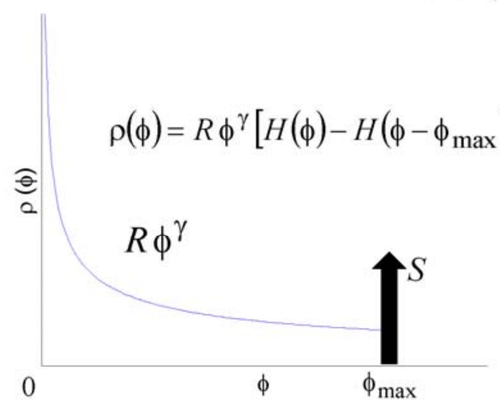
4-PARAMETER IWAN

$$F(t) = \int_0^\infty \rho(\phi)\left[u(t) - x(t,\phi)\right] d\phi$$

$$\dot{x}(t,\phi) = \dot{u}(t) \text{ if } \left\| u(t) - x(t,\phi) \right\| = \phi \text{ and } \dot{u}\left(u(t) - x(t,\phi)\right) > 0$$

$$\rho(\phi) = R\,\phi^{\gamma}\left[H(\phi) - H(\phi - \phi_{max})\right] + S\,\delta(\phi - \phi_{max})$$





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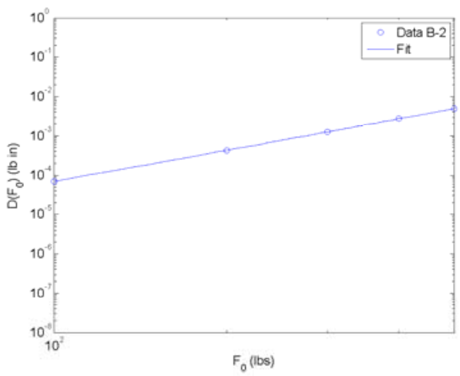
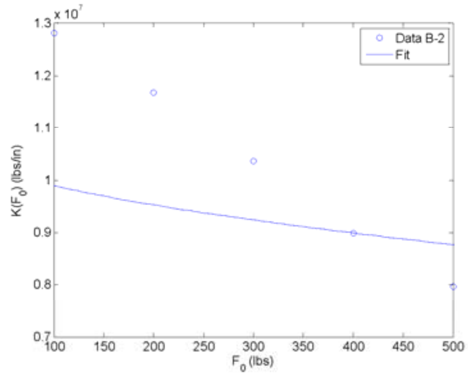
STOCHASTIC MODELING

IS the model correct? Identify the parameters of the model and check prediction. Use stiffness and dissipation data





NO, the model is not correct (not for all samples)

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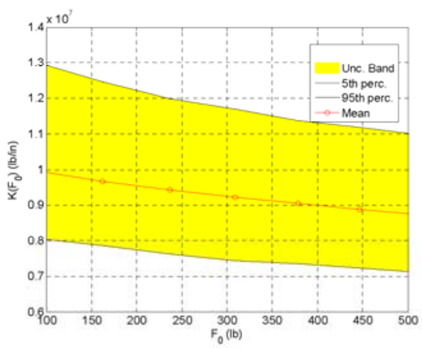
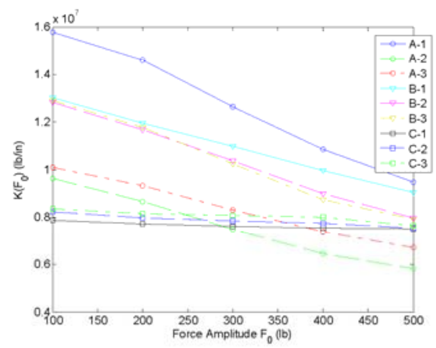
ALEATORIC STOCHASTIC MODELING

IF the model had been correct, it would have "sufficed" to make the 4 parameters of the model $(\gamma, \phi_{max}, R, S)$ random variables. (7% standard deviations for all). Band would end up prohibitively large.





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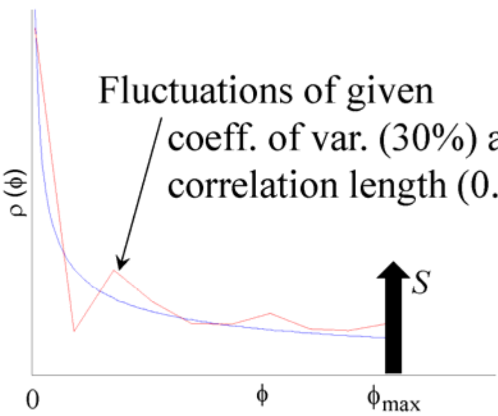
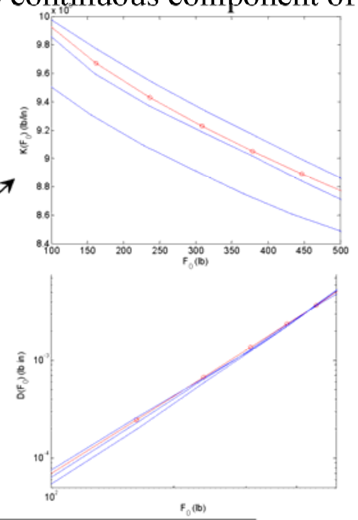
EPISTEMIC STOCHASTIC MODELING

<u>OPTION #1:</u> Allow smooth variations of the continuous component of $\rho(\phi)$ around the mean function $R\,\phi^{\gamma}$





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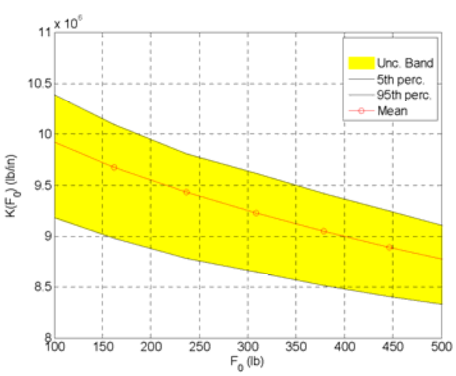
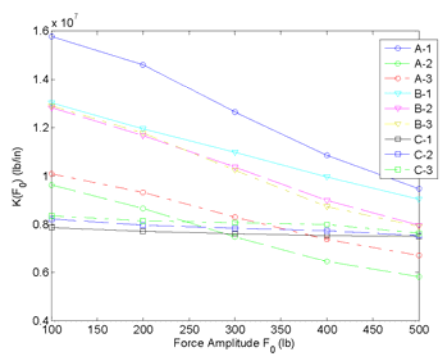
EPISTEMIC STOCHASTIC MODELING

<u>OPTION #1:</u> Allow smooth variations of the continuous component of $\rho(\phi)$ around the mean function $R\,\phi^{\gamma}$. Band too large!





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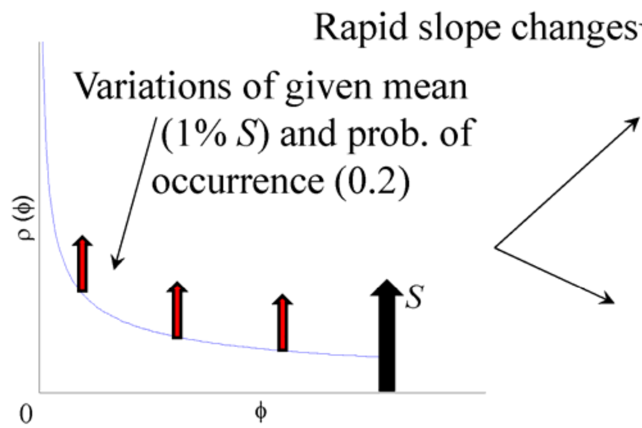
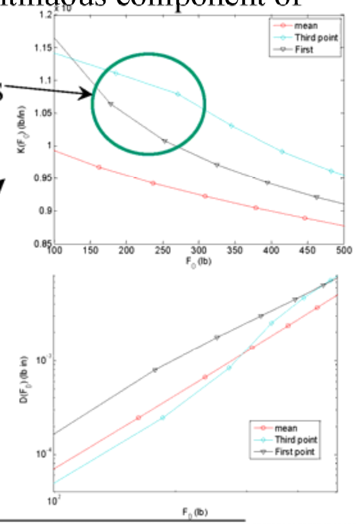
EPISTEMIC STOCHASTIC MODELING

<u>OPTION #2:</u> Dirac delta variations of the continuous component of $\rho(\phi)$ around the mean function $R\phi^{\gamma}$





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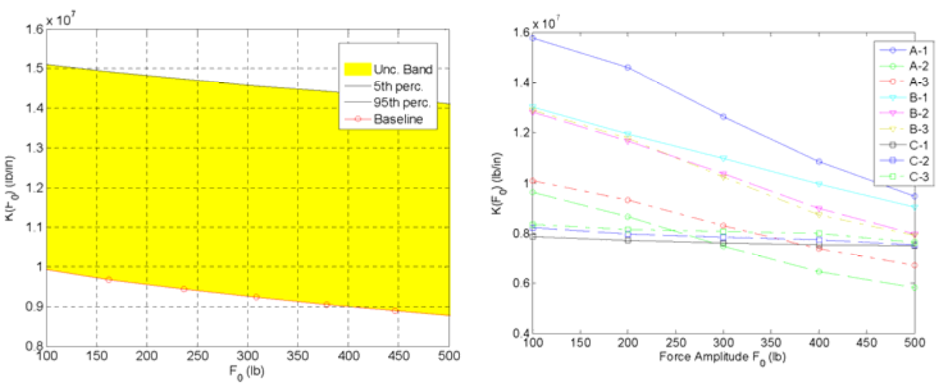
EPISTEMIC STOCHASTIC MODELING

<u>OPTION # 2:</u> Dirac delta variations of the continuous component of $\rho(\phi)$ around the mean function $R\phi^{\gamma}$. Band still too large, mean too high!



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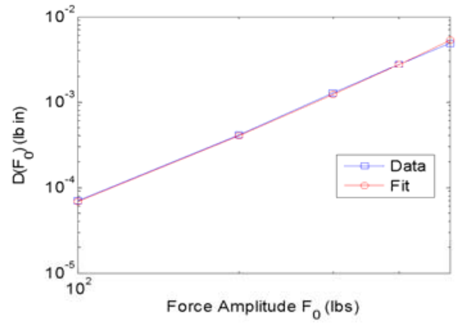
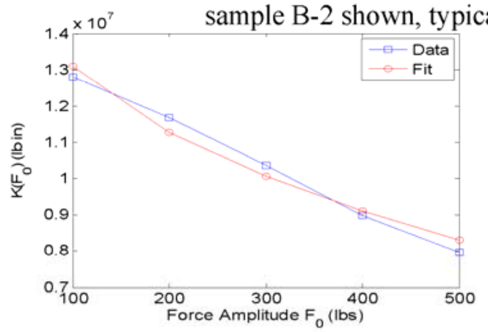
5-PARAMETER IWAN MODEL

5-parameter Iwan:

* Take $\mu_S \neq \mu_D$ and introduce $\theta = \mu_D / \mu_S \leq 1$. Much better!



sample B-2 shown, typical



Small epistemic uncertainty present! Model aleatoric uncertainty

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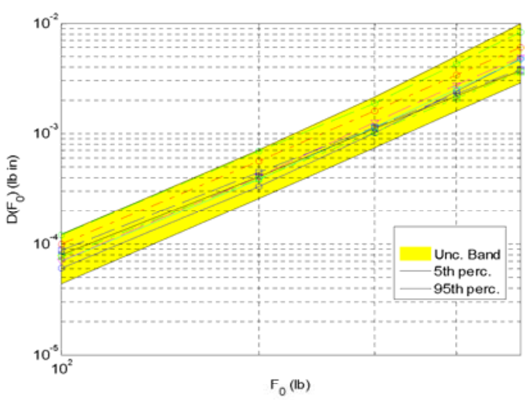
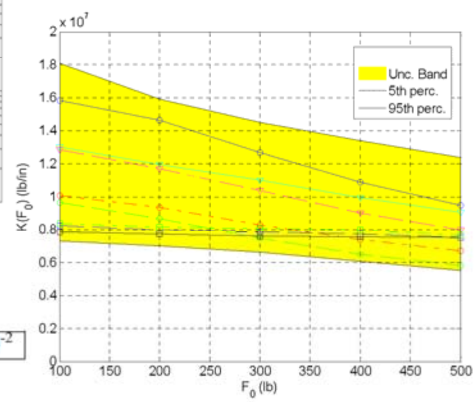
UNCERTAIN 5-PARAMER IWAN MODEL VALIDATION



Predicted uncertainty bands well include experiments.



θ values for all 9 samples:

$3.74\ 10^{-2}$	$5.21\ 10^{-2}$	$7.12\ 10^{-2}$	$9.78\ 10^{-2}$	$3.54\ 10^{-2}$
$4.69\ 10^{-2}$	1.000	1.000	1.000	

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DISCUSSION

Uncertain 4-parameter Iwan:

* Epistemic uncertainty clear and significant on some samples
* Band of uncertainty prohibitively large to match data (due to epistemic uncertainty)

Uncertain 5-parameter Iwan:

* Small, but present, epistemic uncertainty
* Band of uncertainty matches data
* … at least in this case.
* θ values belong to 2 distinct groups, reflects some physics?

NEEDS: More data/analyses of this type and applications

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On the predictive analysis of dynamics in complex structures with joints

E.P. Petrov, University of Sussex, UK

Predictive analysis of dynamics of complex machinery structures

Majority of practical structures are assembled, jointed structures or structures interacting through contact interfaces

Analysis of assembled systems with gaps, impacts, friction damping and other types of nonlinear interaction is required

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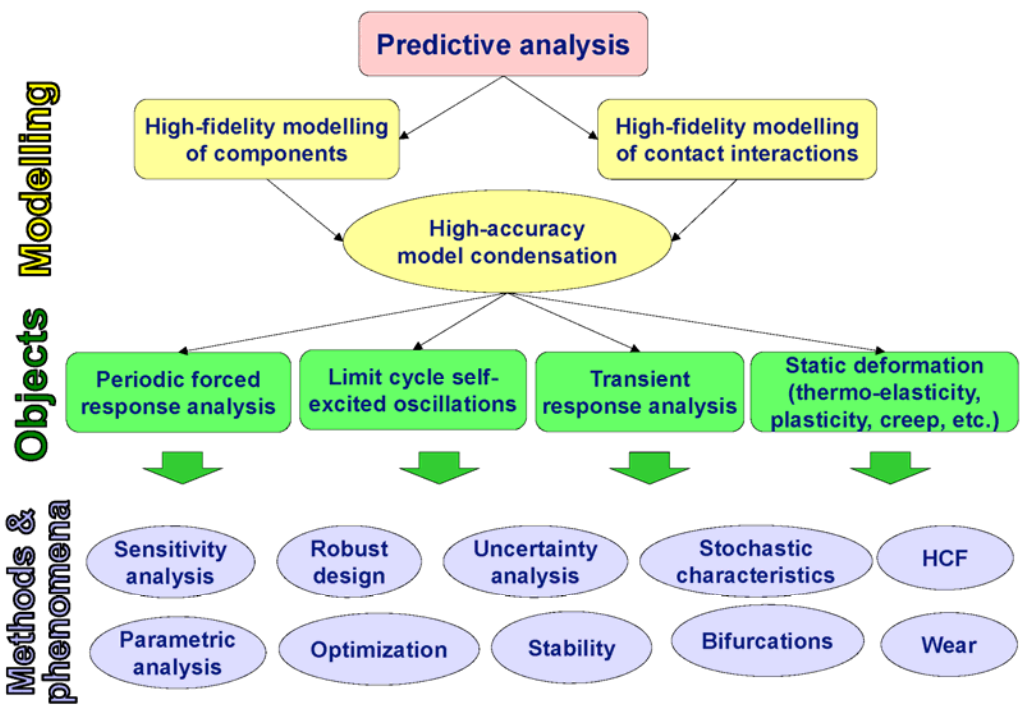

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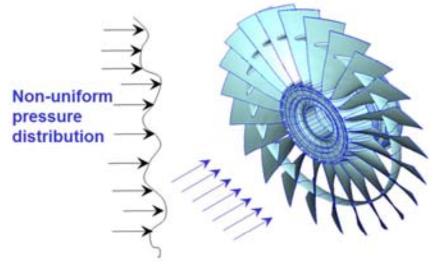
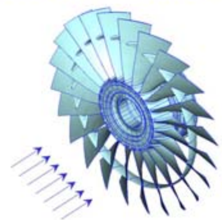
Major types of periodic vibrations

Forced response: vibrations are excited by external forces which vary in times: aerodynamic forces, unbalances, etc.



Self-excited vibrations: excited and maintained by energy source due to its interaction with a structure: e.g. flutter, rubbing at contact interfaces



Limit cycle oscillations (LCO) are periodic, the principal frequency is dependent on the contacts

$$\underbrace{\boldsymbol{M}\ddot{\boldsymbol{x}} + \boldsymbol{C}\dot{\boldsymbol{x}} + \boldsymbol{K}\boldsymbol{x}} + \boldsymbol{f}_{contact}(\boldsymbol{x}, \dot{\boldsymbol{x}}) + \boldsymbol{f}_{aero}(\boldsymbol{x}, \dot{\boldsymbol{x}}) = \boldsymbol{p}(t)$$

Elastic, damping & inertia forces of the FE model — Nonlinear contact interface forces — Other forces: aeroelastic or self-exciting friction — Excitation forces

Multiharmonic equation for forced response & LCO

Time domain equation

$$\boldsymbol{K}\boldsymbol{x} + \boldsymbol{C}\dot{\boldsymbol{x}} + \boldsymbol{M}\ddot{\boldsymbol{x}} + \boldsymbol{f}_c(\boldsymbol{x}, \dot{\boldsymbol{x}}) + \boldsymbol{f}_{aero}(\boldsymbol{x}, \dot{\boldsymbol{x}}) = \boldsymbol{p}(t)$$

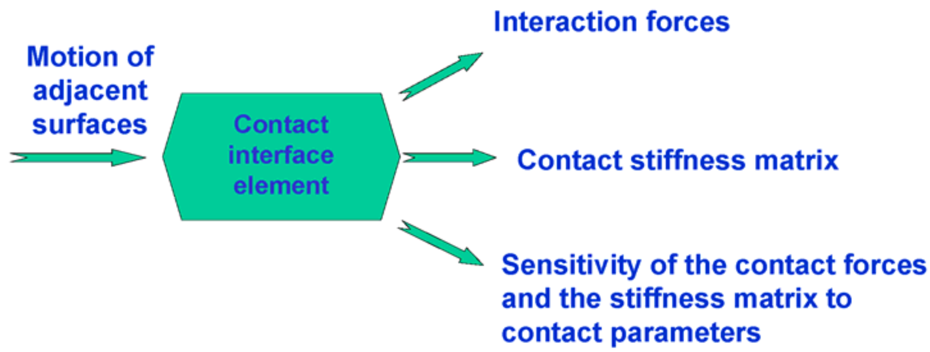
$$\boldsymbol{x} = \sum_{j=1}^{n} \boldsymbol{X}_j^c \cos m_j \omega t + \boldsymbol{X}_j^s \sin m_j \omega t$$

Frequency domain equation (with the reduced size of the equations)

$$\boldsymbol{R}(X,\omega) = \begin{Bmatrix} X_1^c \\ X_1^s \\ \cdots \\ X_n^s \end{Bmatrix} + \begin{bmatrix} \mathrm{Re}(A_1(\omega)) & \mathrm{Im}(A_1(\omega)) & \cdots & \boldsymbol{0} \\ -\mathrm{Im}(A_1(\omega)) & \mathrm{Re}(A_1(\omega)) & \cdots & \boldsymbol{0} \\ \cdots & \cdots & \cdots & \cdots \\ \boldsymbol{0} & \boldsymbol{0} & \cdots & \mathrm{Re}(A_n(\omega)) \end{bmatrix} \begin{Bmatrix} F_1^c(X,\omega) \\ F_1^s(X,\omega) \\ \cdots \\ F_n^s(X,\omega) \end{Bmatrix} = \begin{Bmatrix} P_1^c(\omega) \\ P_1^s(\omega) \\ \cdots \\ P_n^s(\omega) \end{Bmatrix}$$

Displacement amplitudes — FRF matrix with aeroeffects for each harmonic — Contact force amplitudes — Excitation force amplitudes

Analytically derived contact interface elements



Expressions are obtained in analytical form: exact + fast calculations

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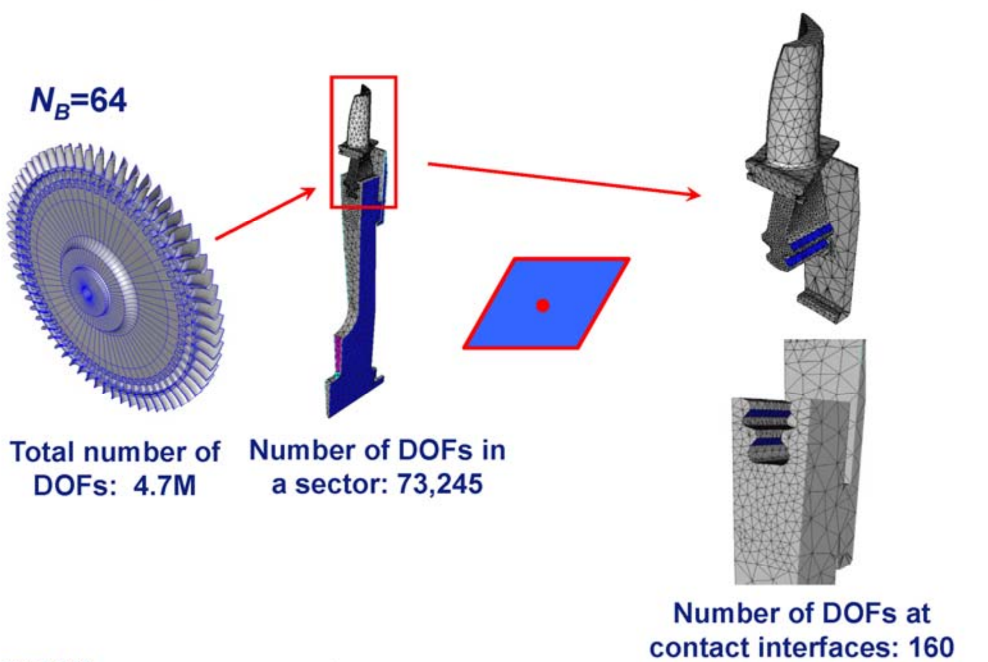
An example of contact interface modelling: a blade-disc joint



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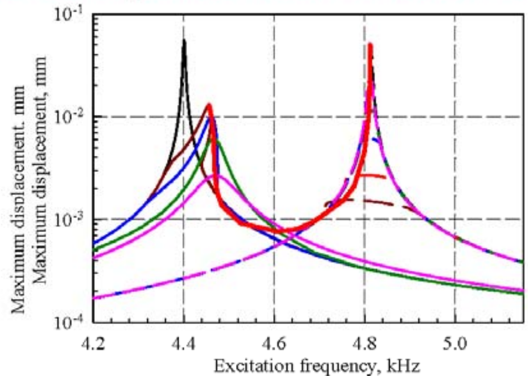
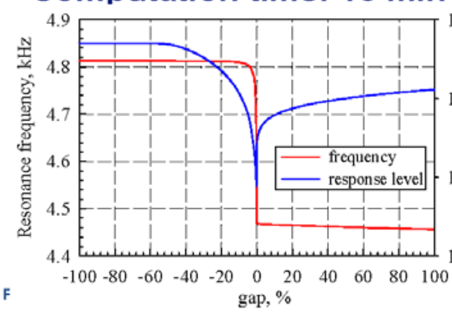
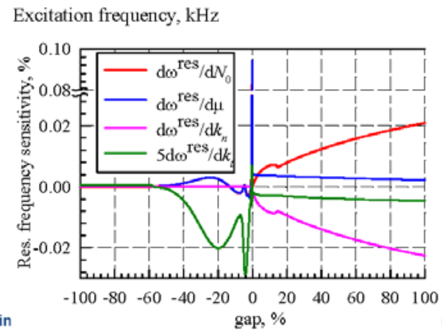
The resonance peak response and its sensitivity



Computation time: 10 min





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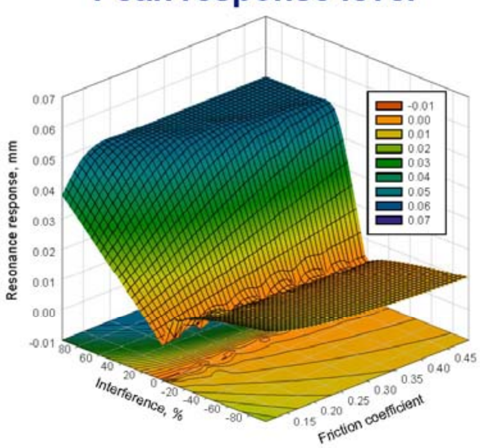
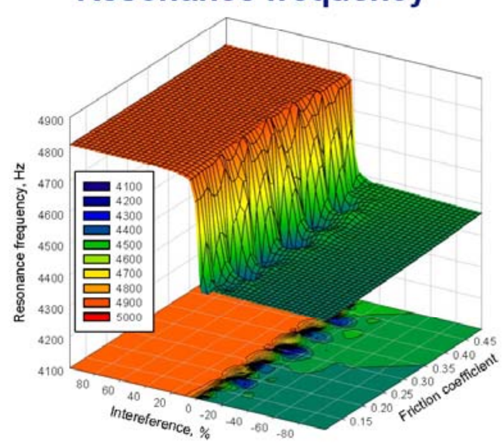
Resonance frequency and response dependency on contact interface parameters: friction coefficient & interferences

Peak response level

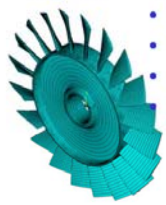


Resonance frequency



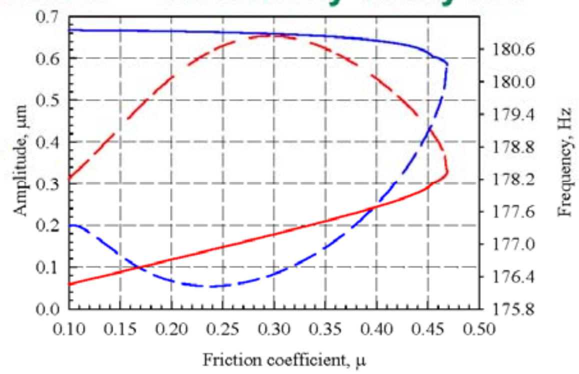
LCO analysis: parametric + sensitivity analyses

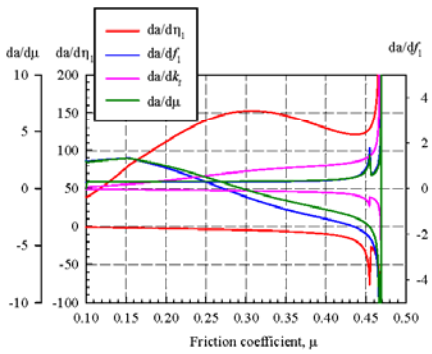
N_B=20

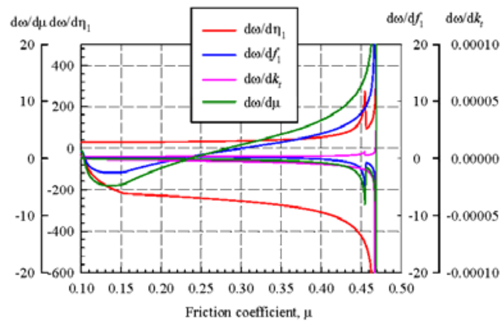


Parameters analysed:
- friction coefficient
- contact stiffness
- flutter intensity
- fluttering frequency

Number of DOFs in a sector: 54,000

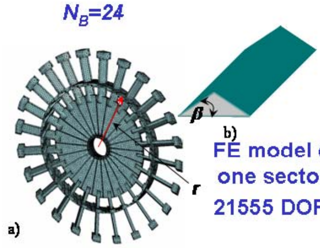




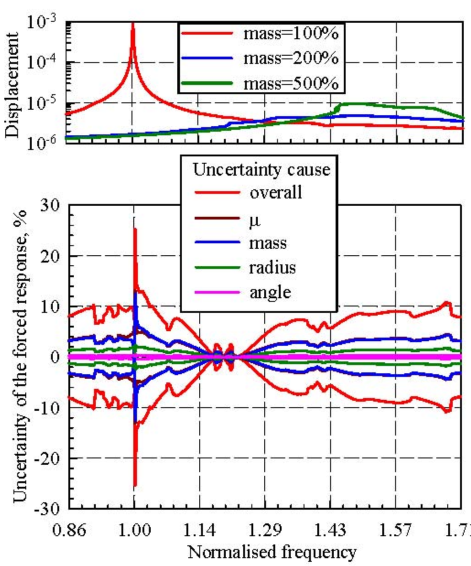


Example of analysis for ranges of uncertainty of the forced response

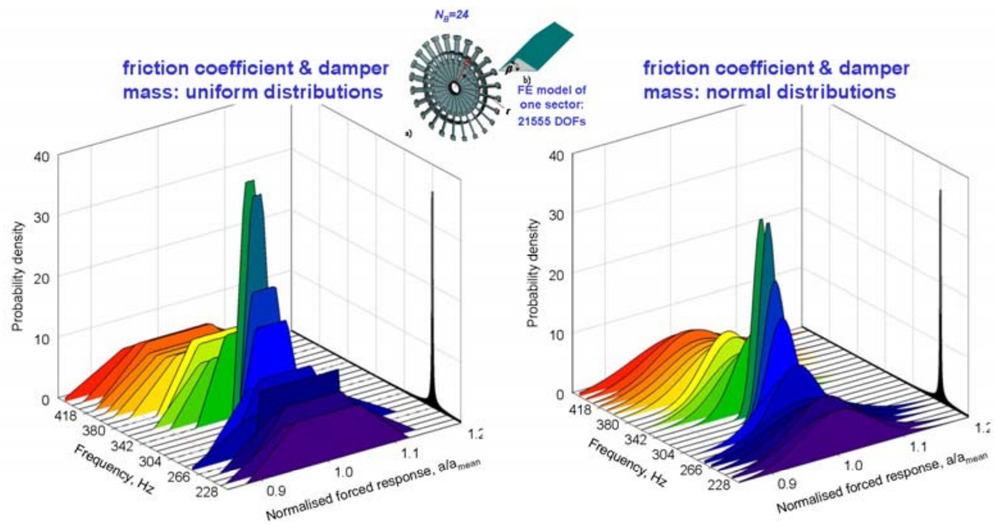
N_B=24



FE model of one sector: 21555 DOFs

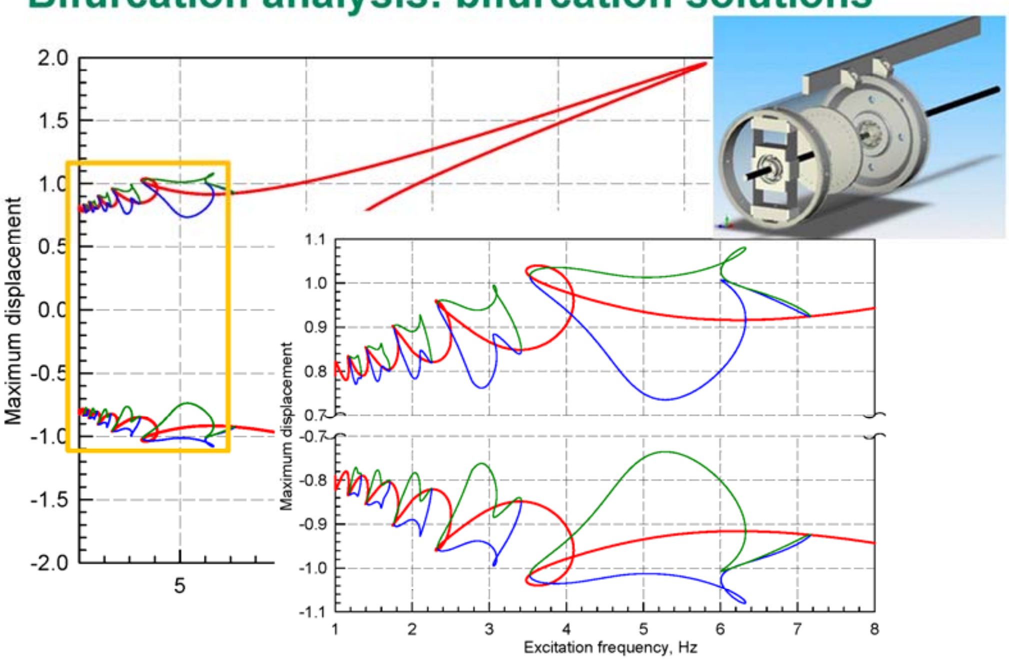


Example: probability density functions of the forced response level for the whole frequency range of interest



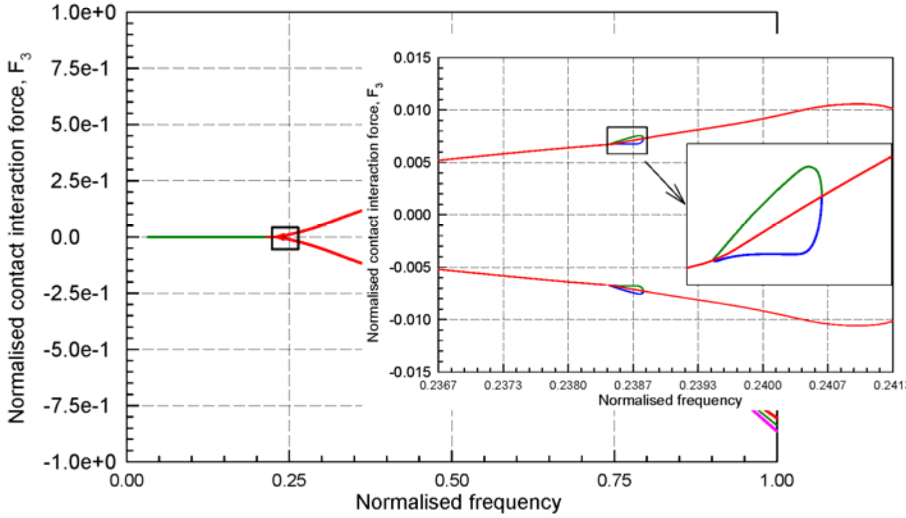
Probability density functions (PDFs) for forced response levels are derived analytically through PDFs of design parameters

Bifurcation analysis: bifurcation solutions



A gas-turbine engine model: branching vibration regimes

Contact force at 3rd rubbing contact



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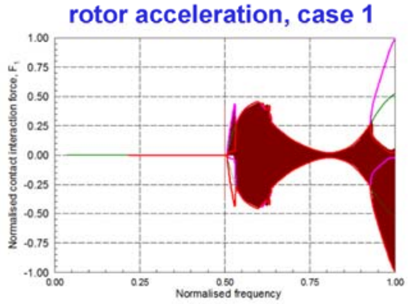
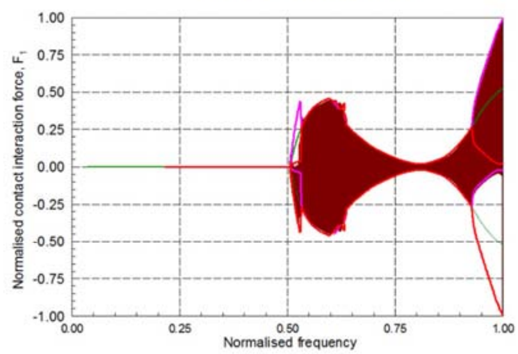
Comparison with the time domain: possible bifurcation scenarios

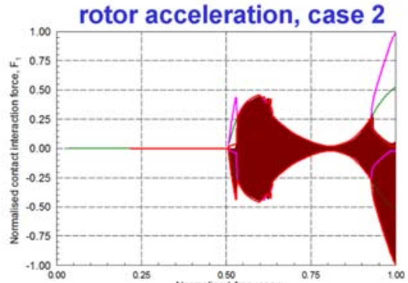
rotor acceleration, case 1



rotor deceleration



rotor acceleration, case 2



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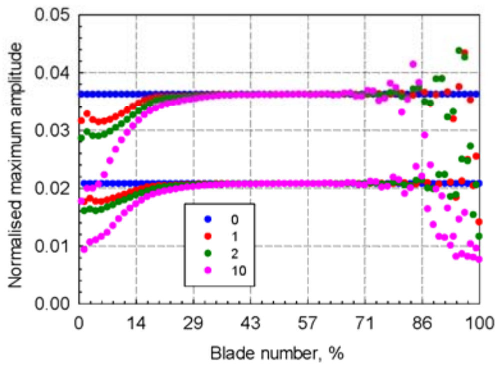
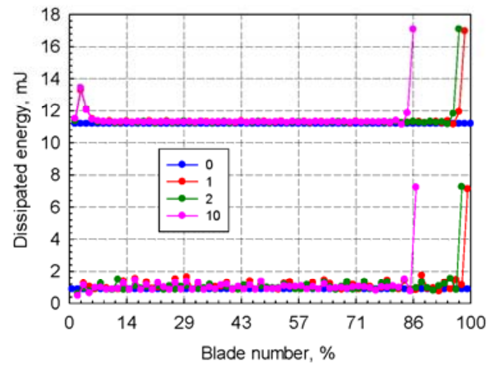
Evolution of the blade amplitudes and energy dissipated by dampers with the sequential damper loss

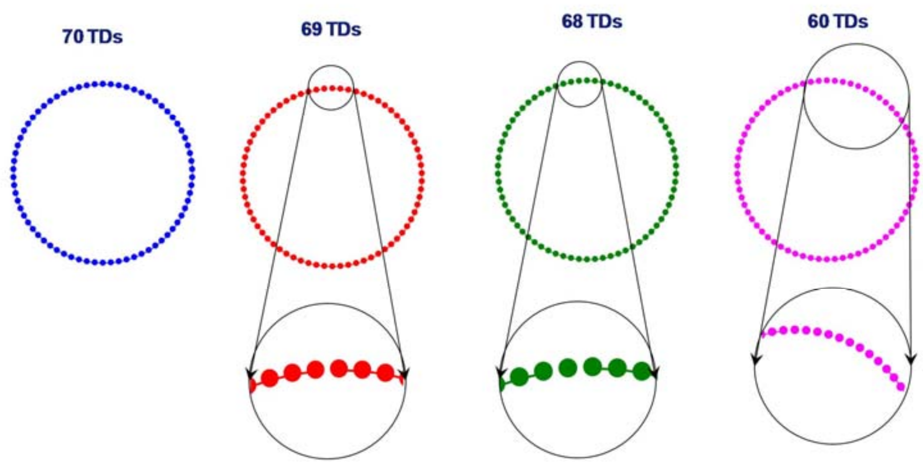
Maximum blade amplitudes



Energy dissipated by dampers over vibration period



The scheme of a progressive loss of tip dampers due to wear



Alex Vakakis: Methodologies for Nonlinearity Quantification and Nonlinear System Identification

Methodologies for Nonlinearity Quantification and Nonlinear System Identification

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

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1

Nonlinear System Identification

- Aim: NSI methodology of broad applicability. Issues:
  - Weak / strong nonlinearities (geometric, kinematic, boundary conditions, friction, clearance, vibro-impact, fluid/structure interactions, self-excitation, relaxation…)
  - Smooth / non-smooth dynamics
  - Uncertainties, incomplete data, multi-physics
  - Repeatability
  - Possibility of multiple co-existing responses (periodic, quasi-periodic or chaotic), domains of attraction, dependence on initial conditions, energy, rate of forcing
  - Multi-scale effects (e.g., waves at small time scales/high frequencies,…), mid-frequency range – mixing of scales
- Predictive design, model updating

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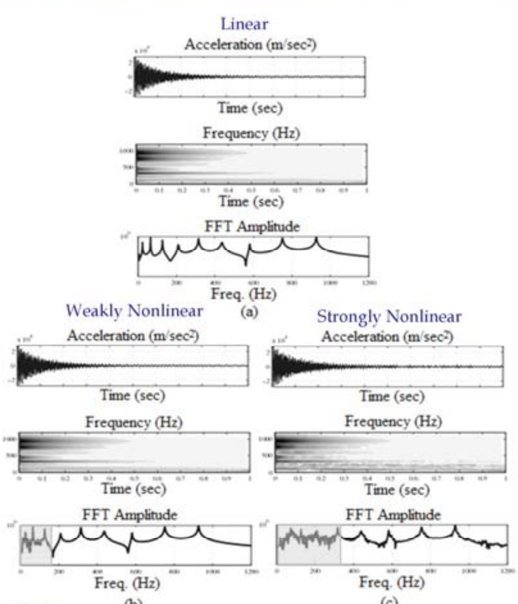
2

Example 1: Beam with local strong stiffness nonlinearity



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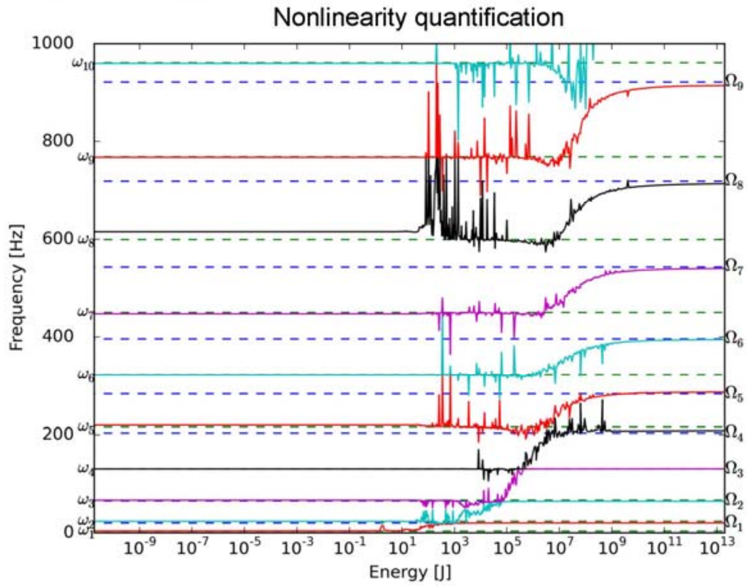
3

Example 1: Beam with local strong stiffness nonlinearity



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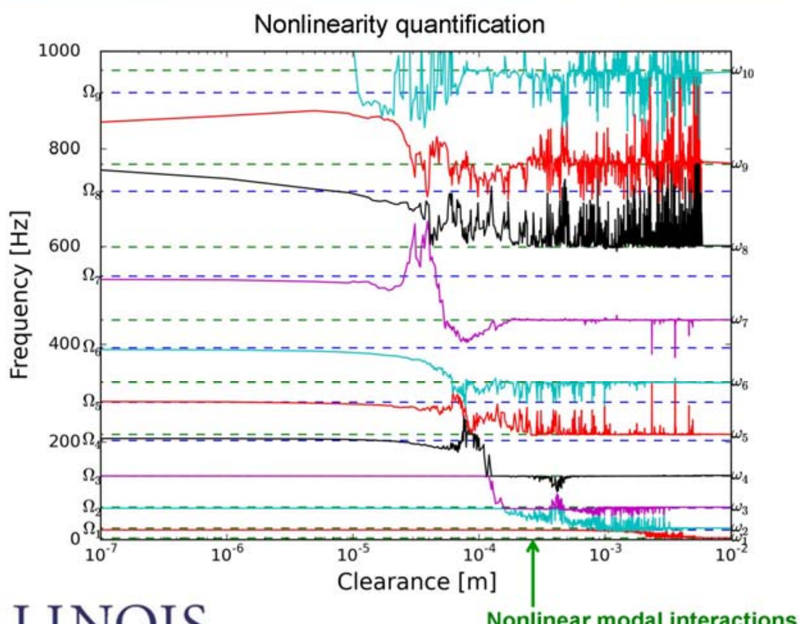
Example 2: Beam undergoing vibro-impacts



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5

Looking forward

- Potential for data-driven reduced-order nonlinear models capable of capturing even strong and time-varying nonlinear effects
- Open issues:
  - Nonlinear model updating
  - Predictable models of complex systems (e.g., structures with multiple mechanical joints)
  - Dry friction, plasticity effects
  - Modal interactions leading nonlinear beat phenomena
  - …

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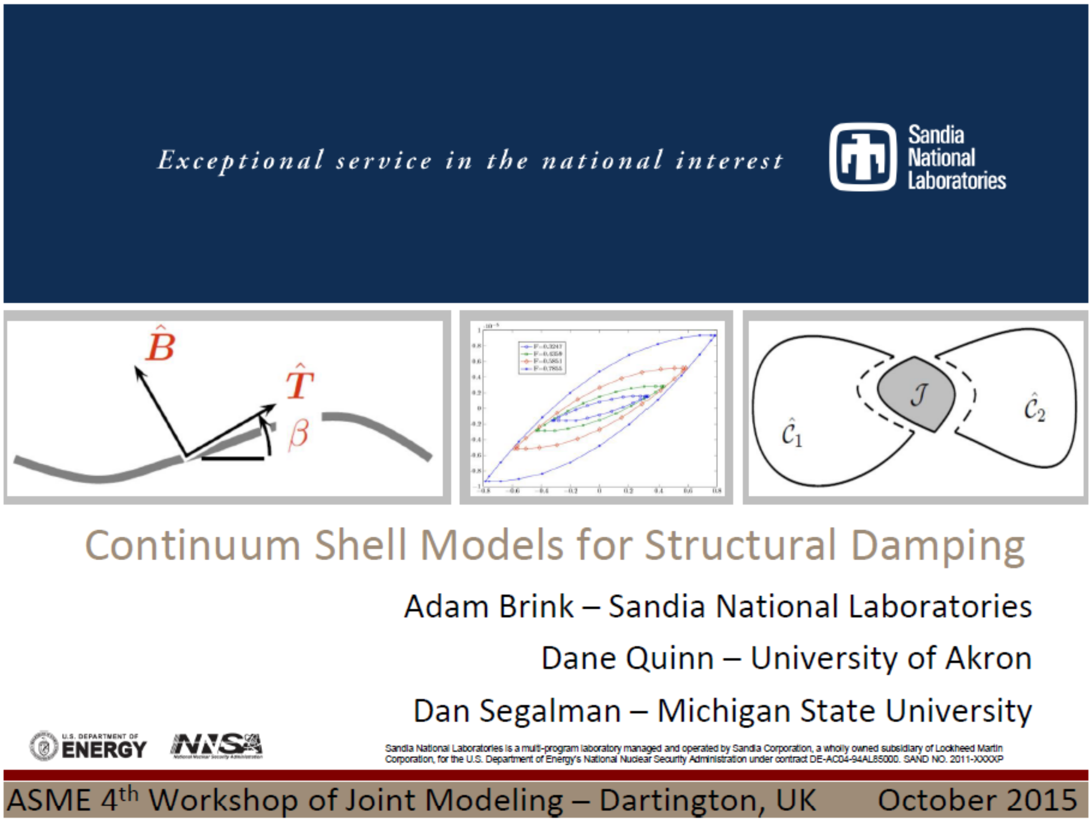
5.6. Submitted Short Talks from the Second Evening

Adam Brink: Continuum Shell Models for Structural Damping

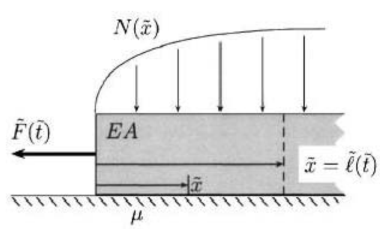


Research Motivation

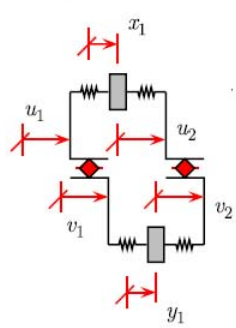
- Develop a shell model to intrinsically capture joint nonlinearities
- Why shells?
  - Beams and shells are well understood both experimentally and numerically
  - Analysts are traditionally comfortable using with beams and shells
  - Already captures most of the mechanics needed
    - Bending, shear and axial loading

History

- Quinn and Segalman: bar on rigid foundation formulation (series-series Iwan) [1]

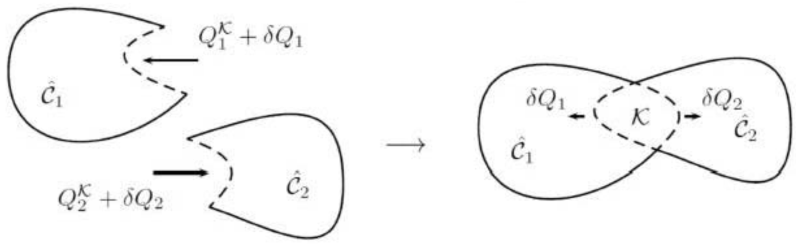


- Miller and Quinn: discretized two-sided interface model [2]



Modal Analysis

- Quinn solves modal equations of motion for a monolithic structure, then adds the effect of the joint back in.



$$\left[\int_{\mathcal{M}} \phi_i(x)\, \rho_{\mathcal{M}}(x)\, \phi_i(x)\, dx\right] \ddot{A}_i(t) + \left[\int_{\mathcal{M}} \frac{\partial \phi_i(x)}{\partial x}\, EA_{\mathcal{M}}(x)\, \frac{\partial \phi_i(x)}{\partial x}\, dx\right] A_i(t)$$

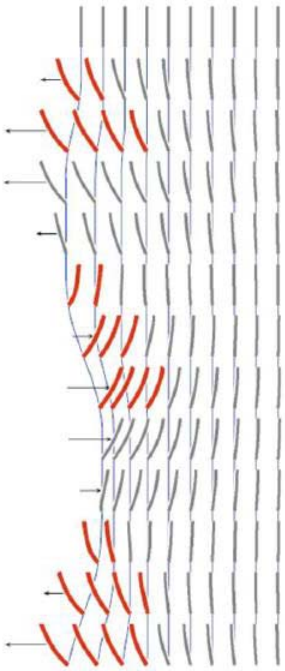
Monolithic Response

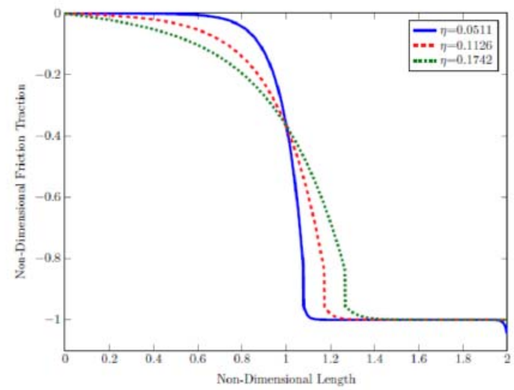
$$= -\left(\phi_i(s_1)\, \delta Q_1(t) + \phi_i(s_2)\, \delta Q_2(t)\right).$$

Forces Arising From Joint at Interfaces

Sandia National Laboratories

Why Include Shear?



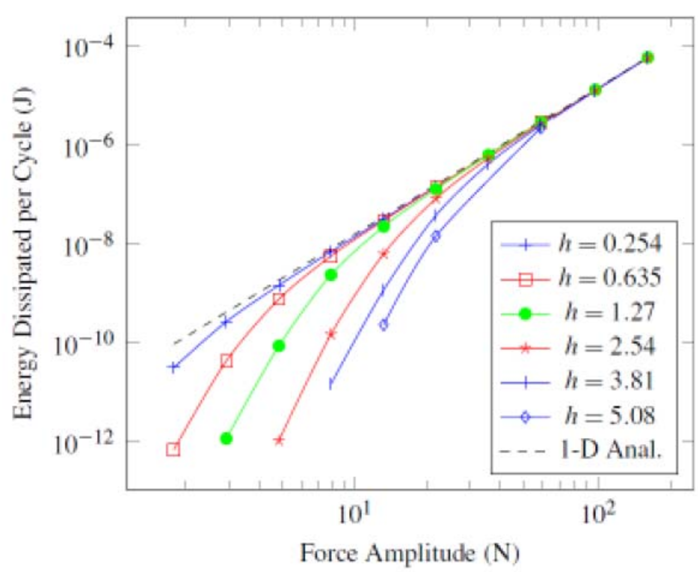


- Shearing allows forces to transmit beyond the slip initiation length.
- The shorter the cross-section, the more rod-like the behavior becomes.

Sandia National Laboratories

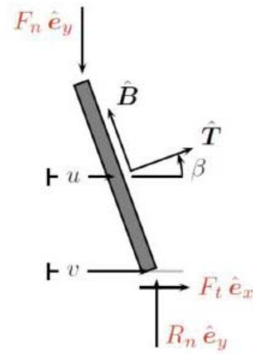
Why Include Shear?

- The presence of shear deformation allows for a precipitous drop off from the cubic power law bar solution



Shearable Shell Derivation

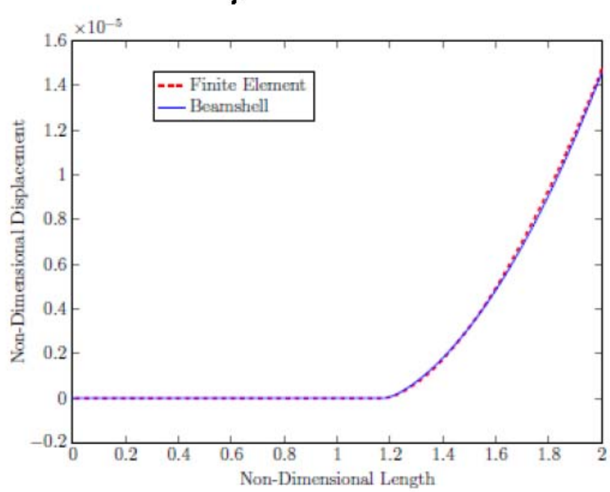
- Using kinematic arguments and geometrically exact shell theory [3], the shell equations of motion are:



$$
\begin{aligned}
(N' - \beta' Q) + F_t \cos\beta - (R_n - F_n)\sin\beta &= 0, \\
(Q' + \beta' N) + F_t \sin\beta + (R_n - F_n)\cos\beta &= 0, \\
M' + (1+e)\,N - g\,Q + h\,(F_t \cos\beta + (R_n + F_n)\sin\beta) &= 0.
\end{aligned}
$$

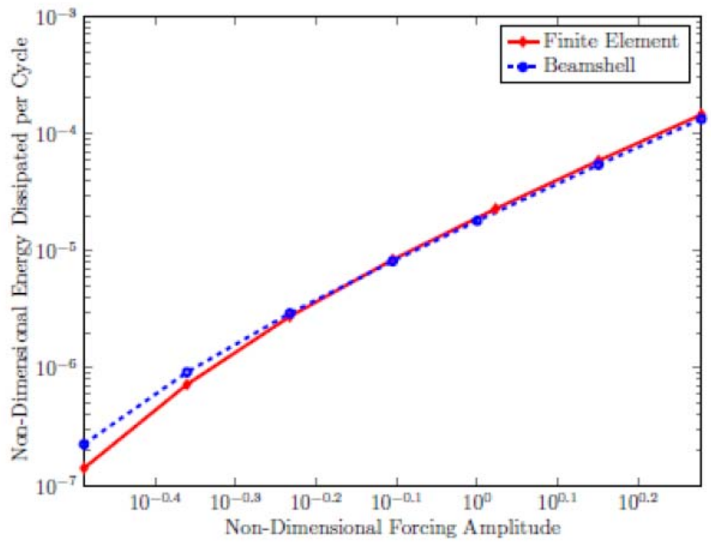
Results – Beam on Rigid Foundation

- Displacement solution and slip zone length prediction is nearly identical



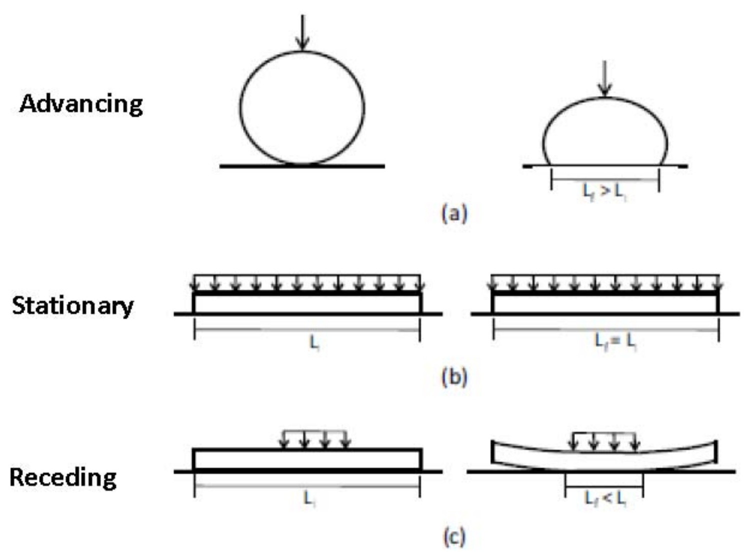
Results – Beam on Rigid Foundation

- Energy dissipated per cycle shows good agreement.
- The effect of the overly stiff shell cross-section is seen at low forcing amplitudes.



Other Cool Things…

- Estimation of the contact patch dimensions is crucial to good predictions.
- Contact behavior is categorized as:



Sandia National Laboratories

Receding Contact

- For current shell theory to capture receding contact, an through thickness stress state is assumed (Airy stress function).
- Without an assumption the shell does not respond to equal squeezing pressure (see [4]).

$$\int_{-t}^{t}\left[\int_{-H}^{H}\mathbf{S}^{\sigma}\mu d\zeta\big|_a^b + \int_a^b\left(\int_{-H}^{H}\mathbf{f}\mu d\zeta + \mathbf{S}^{\zeta}\mu\big|_{-H}^{H}\right)d\sigma\right]dt - \int_a^b\left(\int_{-H}^{H}\rho\dot{\mathbf{r}}\mu\zeta\right)\Big|_{-t}^{t} d\sigma = 0$$

Integrates away through thickness effect

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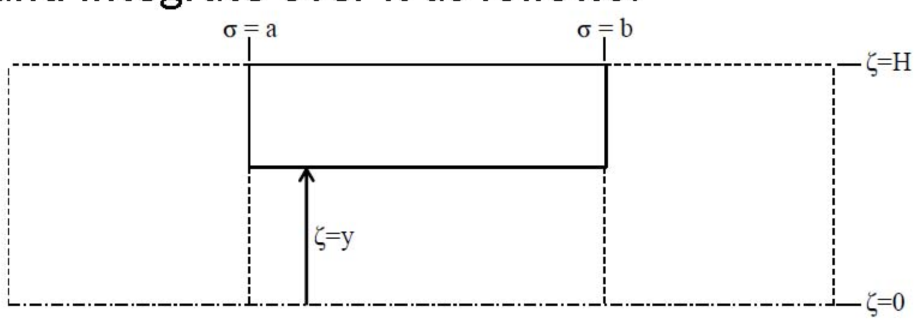
Receding Contact

- Split the through thickness control volume and integrate over it as follows:



$$\left[\int_{-H}^{H}\left(\int_{y}^{H}\tau_{12}\,dx_2 - \int_{-H}^{-y}\tau_{12}\,dx_2\right)dy\right]_a^b + \int_a^b\left[2H\left(\sigma_{22}(H)+\sigma_{22}(-H)\right) - \int_{-H}^{H}\left(\sigma_{22}(y)+\sigma_{22}(-y)\right)dy\right]dx_1 = 0$$

Long Story Short...

$$\xi' - \frac{1}{2H}\chi + p_z = 0$$

New state equation for through thickness stress.

Squeezing Stress

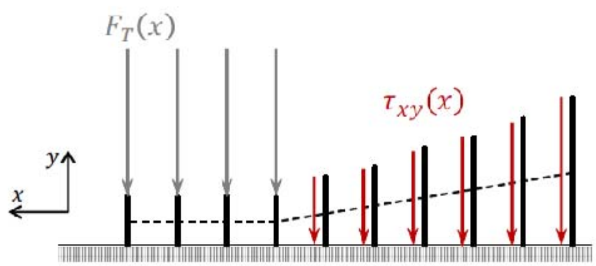
$$\chi = \int_{-H}^{H} \sigma_{22}(y)\, dy$$

Applied External Squeezing Traction

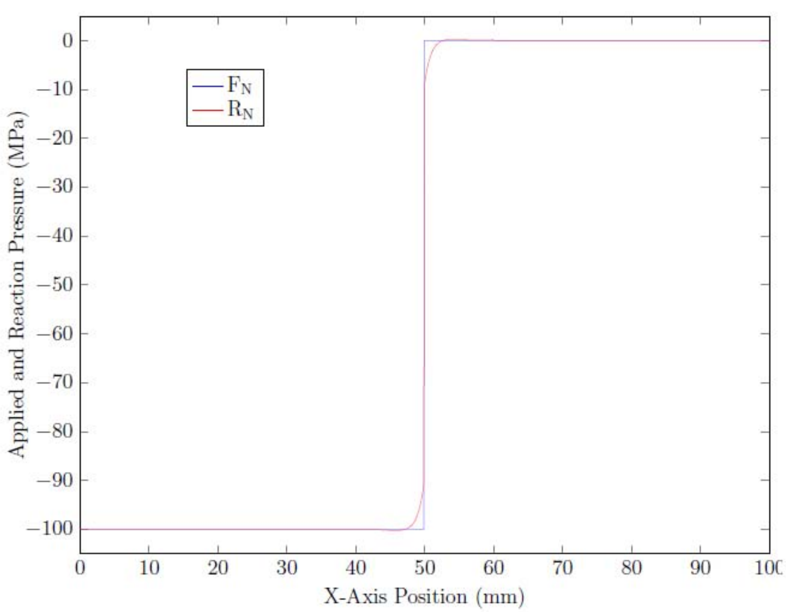
$$p_z = \frac{(\sigma_{22}(H) + \sigma_{22}(-H))}{2}$$

Shear Distribution

$$\xi = \frac{1}{4H}\left[\int_{-H}^{H}\left(\int_{y}^{H} \tau_{12}\, dx_2 - \int_{-H}^{-y} \tau_{12}\, dx_2\right) dy\right]$$

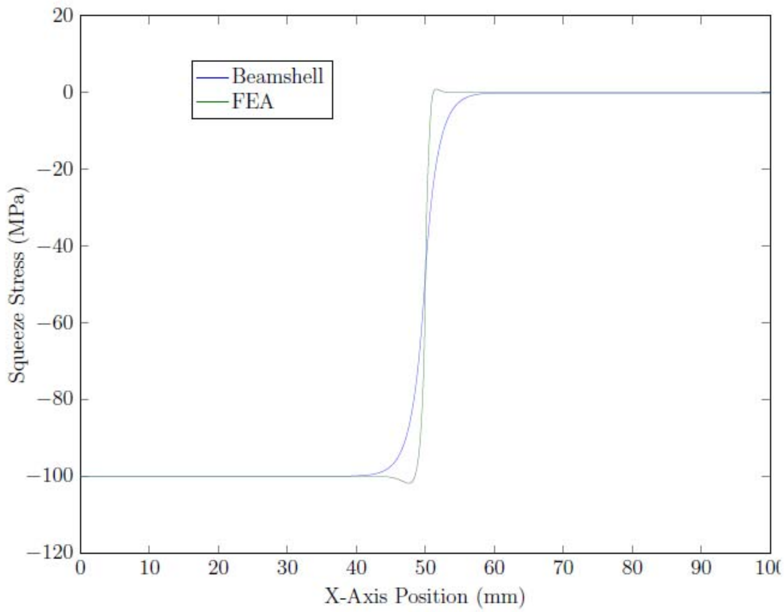


Receding Results



Reaction traction extends beyond load application

Receding Results



Receding contact capture by shell

Conclusions

- Nonlinear shells present an excellent platform to study jointed connection problems.
- New physical constitutive laws easily incorporated into the shell framework.
- Captures frictional energy dissipation as well as predicting contact patch geometry.

Citations

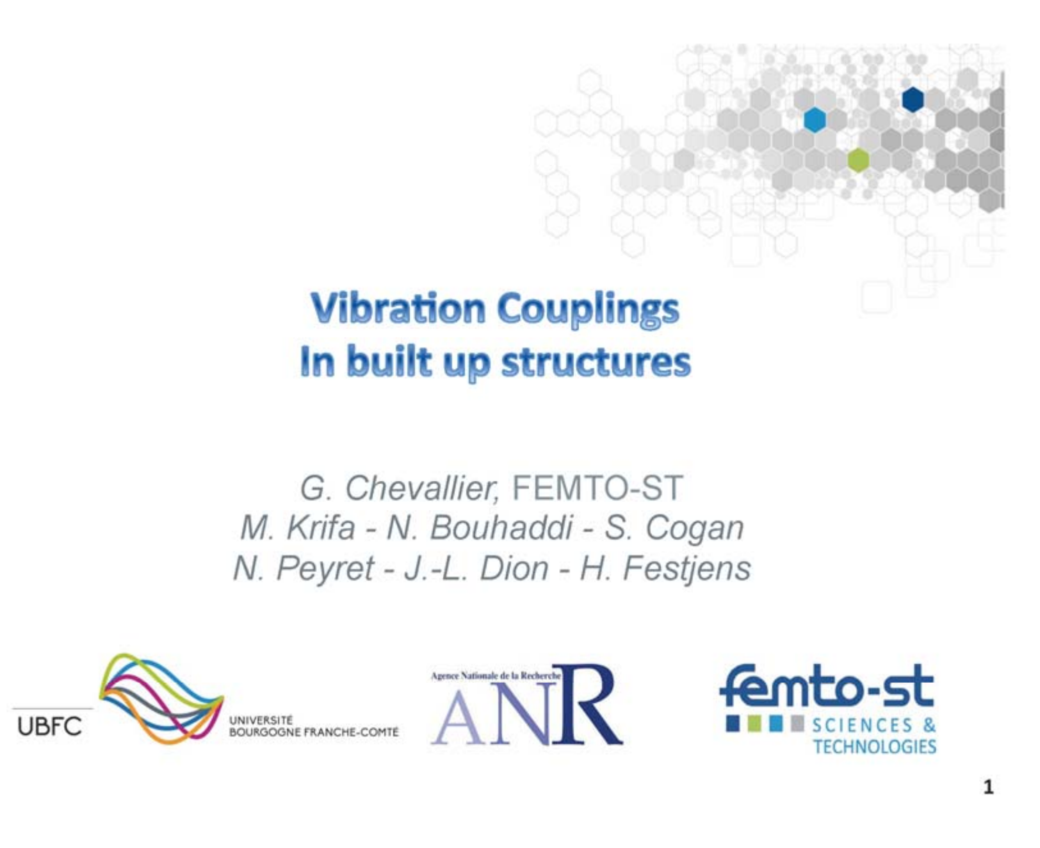
[1] D. Dane Quinn and D.J. Segalman. Using series-series iwan-type models for understanding joint dynamics. *Journal of Applied Mechanics*, 72:778–784, 2005.

[2] Jason D. Miller and D. Dane Quinn. A two-sided interface model for dissipation in structural systems with frictional joints. *Journal of Sound and Vibration*, 321:201–219, 2008.

[3] A. Libai and J.G. Simmonds. *The Nonlinear Theory of Elastic Shells*. Cambridge University Press, Cambridge, UK, 1998.

[4] L.N.G. Filon. On an approxiamte solution for the bending of a beam of rectangular cross-section under any system of load. *Philosophical Transactions of the Royal Society of London*, 206A:63–155, 1903.

Gael Chevallier: Vibration Couplings in Built Up Structures



Bio and Topics

Biography :

- Bachelor in Aix –Marseille University, FR
- Master Degree in ENS Cachan, FR
- PhD in Supméca
- Assistant Prof in Supméca, FR from 2005 to 2014
- **Prof in FEMTO-ST – University of Franche-Comte, Besancon, FR from 2014 - ...**

Topics :

- Damping :
  - Viscoelasticity, polymers
  - Friction
  - Piezo-electricity
- Squeal Noise
- Experiments : new testing devices,
- Simulations : model order reduction

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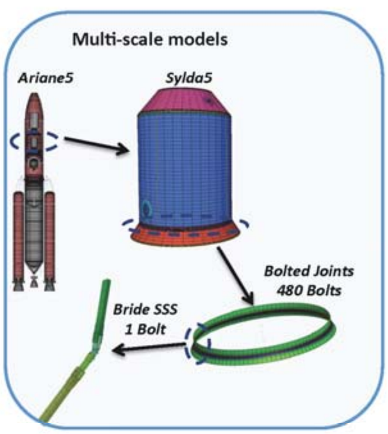
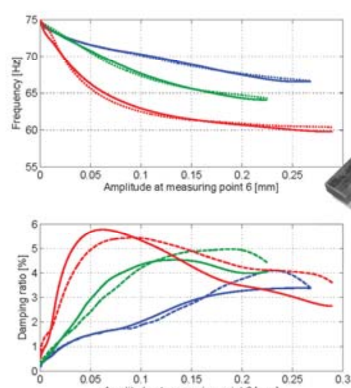
Introduction

- **Localized Dissipations**
- **Non Linearities**

- **Modal Density**
- **Large frequency Bandwidth**

- **Modal Couplings**
- **Hard to measure/compute damping**





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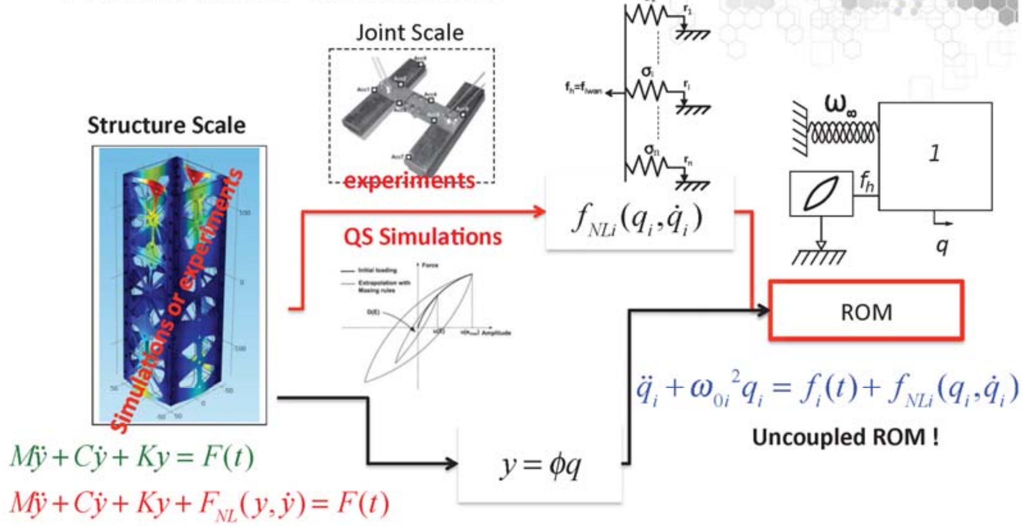
Traditional workflow



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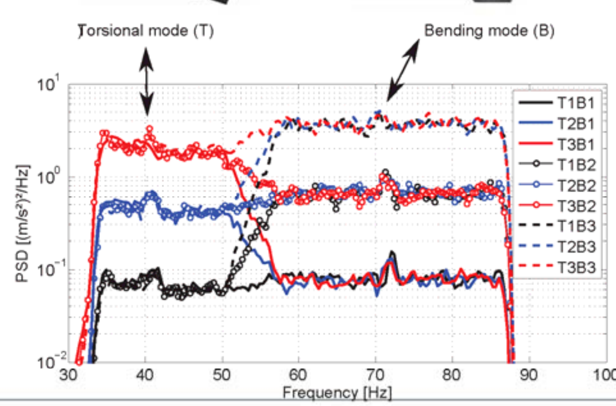
Experimental observations



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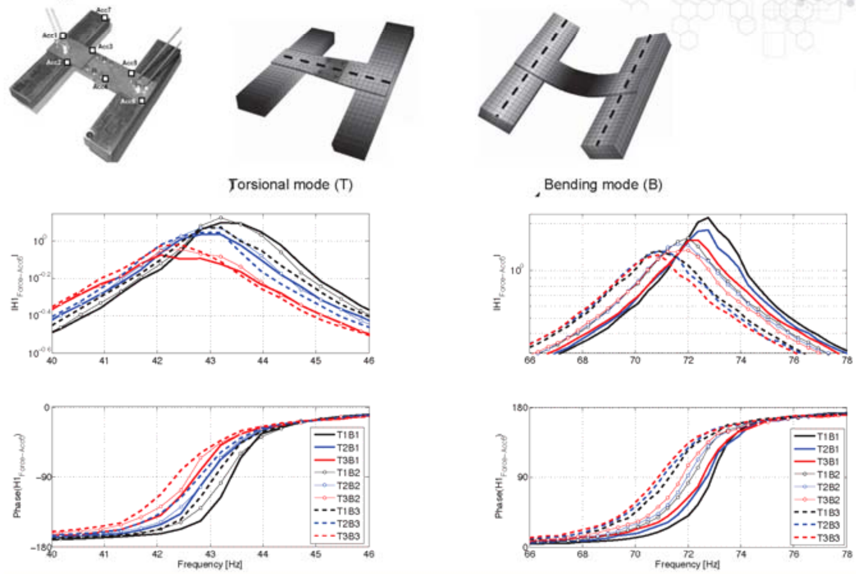
Experimental observations



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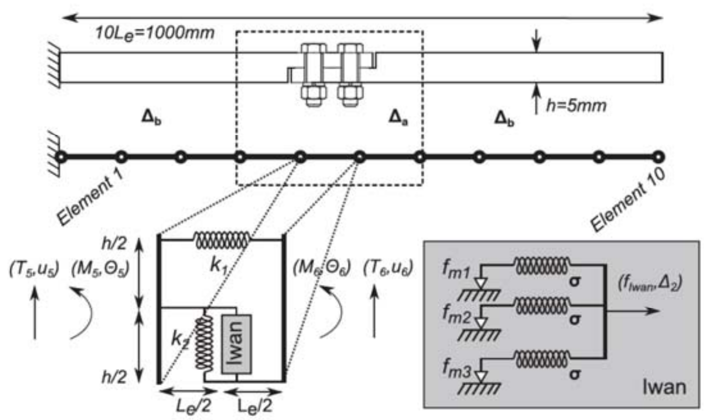
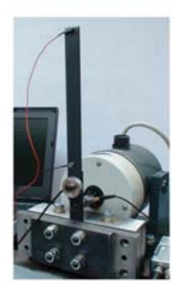
Numerical Observations





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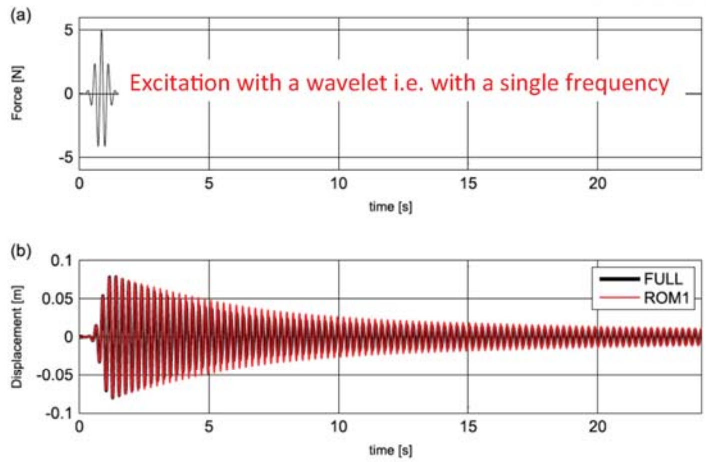
Excitation mono-fréquentielle



FULL $M\ddot{y}+C\dot{y}+Ky=F(t)$ **ROM1** $\ddot{q}_1+\omega_{01}{}^2 q_i=f_1(t)+f_{NL1}(q_1,\dot{q}_1)$

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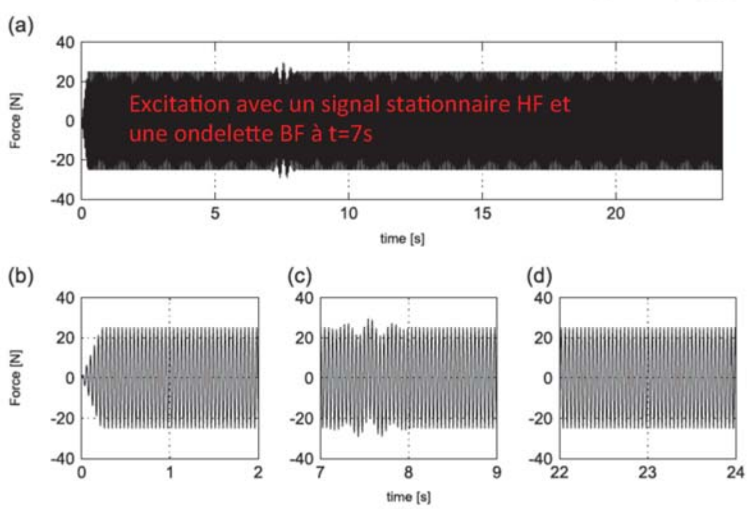
Excitation multi-fréquentielle



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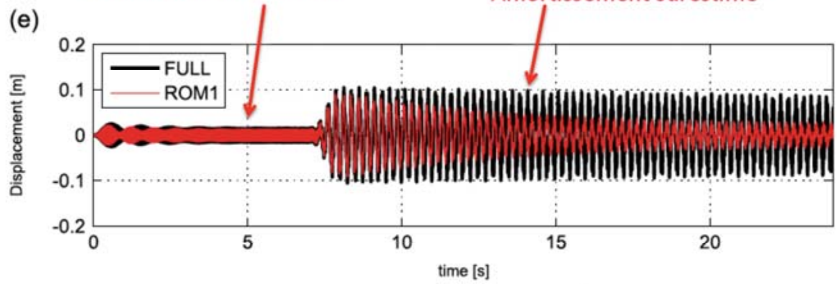
Résultats



FULL $M\ddot{y} + C\dot{y} + Ky = F(t)$

ROM1

$$\ddot{q}_1 + \omega_{01}{}^2 q_i = b_1 f(t) + f_{NL1}(q_1, \dot{q}_1)$$
$$\ddot{q}_2 + \omega_{02}{}^2 q_2 = b_2 f(t) + f_{NL2}(q_2, \dot{q}_2)$$

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Coupling computations

Let's go back to a linear problem, with non-proportional damping, joint damping for instance :

$$M\ddot{y} + C\dot{y} + Ky = f(t)$$

assuming ϕ is a modal basis constituted with p normal modes
thus $\beta = \phi^T C \phi$: **generalized damping matrix**

We decompose the generalized damping matrix into a diagonal matrix and non diagonal one.

$$\beta = \beta_1 + \varepsilon\beta_2 \quad \beta_1 = \begin{pmatrix} . & 0 & 0 \\ 0 & . & 0 \\ 0 & 0 & . \end{pmatrix} \quad \beta_2 = \begin{pmatrix} 0 & . & . \\ . & 0 & . \\ . & . & 0 \end{pmatrix}$$

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Asymptotic expansion

The stationnary response function is assumed to be of the form

$$y_n = y^{(0)} + \varepsilon y^{(1)} + \varepsilon^2 y^{(2)} + \ldots + \varepsilon^n y^{(n)}$$

with

$$y^{(i)} = \phi q^{(i)}$$

Thus

$$y_n = \phi(q^{(0)} + \varepsilon q^{(1)} + \varepsilon^2 q^{(2)} + \ldots + \varepsilon^n q^{(n)})$$

Finally

$$\underbrace{\phi^T M \phi}_{I}(\ddot{q}^{(0)} + \varepsilon \ddot{q}^{(1)} + \varepsilon^2 \ddot{q}^{(2)} + \cdots) + \underbrace{\phi^T C \phi}_{\beta=\beta_1+\varepsilon\beta_2}(\dot{q}^{(0)} + \varepsilon \dot{q}^{(1)} + \varepsilon^2 \dot{q}^{(2)} + \cdots) +$$
$$\underbrace{\phi^T K \phi}_{\Lambda}(q^{(0)} + \varepsilon q^{(1)} + \varepsilon^2 q^{(2)} + \cdots) = \phi^T f$$

Asymptotic expansion

Grouping the terms according to ε powers

$$\ddot{q}^{(0)} + \beta_1 \dot{q}^{(0)} + \Lambda q^{(0)} + \varepsilon(\ddot{q}^{(1)} + \beta_1 \dot{q}^{(1)} + \Lambda q^{(1)} + \beta_2 \dot{q}^{(0)}) +$$
$$\varepsilon^2(\ddot{q}^{(2)} + \beta_1 \dot{q}^{(2)} + \Lambda q^{(2)} + \beta_2 \dot{q}^{(1)}) + \varepsilon^3 \beta_2 \dot{q}^{(2)} + \cdots = \phi^T f$$

From the order 0

$$\ddot{q}^{(0)} + \beta_1 \dot{q}^{(0)} + \Lambda q^{(0)} = \phi^T f \quad \rightarrow \quad q^{(0)} = (-\omega^2 I + j\omega\beta_1 + \Lambda)^{-1} \phi^T f$$

From the order 1

$$\ddot{q}^{(1)} + \beta_1 \dot{q}^{(1)} + \Lambda q^{(1)} = -\beta_2 \dot{q}^{(0)} \quad \rightarrow \quad q^{(1)} = -j\omega(-\omega^2 I + j\omega\beta_1 + \Lambda)^{-1} \beta_2 q^{(0)}$$

From the order 2

$$\ddot{q}^{(2)} + \beta_1 \dot{q}^{(2)} + \Lambda q^{(2)} = -\beta_2 \dot{q}^{(1)} \quad \rightarrow \quad q^{(2)} \underbrace{= -j\omega(-\omega^2 I + j\omega\beta_1 + \Lambda)^{-1}}_{\psi} \beta_2 q^{(1)}$$

Méthode d'estimation du couplage a posteriori

A partir de l'ordre n en ε on déduit l'expression de $q^{(n)}$

$$\ddot{q}^{(n)} + \beta_1 \dot{q}^{(n)} + \Lambda q^{(n)} = -\beta_2 \dot{q}^{(n-1)} \rightarrow q^{(n)} = -j\omega(-\omega^2 I + j\omega\beta_1 + \Lambda)^{-1}\beta_2 q^{(n-1)}$$

On peut facilement démontrer l'expression du $q^{(n)}$ en fonction du $q^{(0)}$

$$q^{(n)} = \psi q^{(n-1)} = \psi^n q^{(0)}$$

$$Avec \begin{cases} q^{(0)} = (-\omega^2 I + j\omega\beta_1 + \Lambda)^{-1}\phi^T f \\ \psi = -j\omega(-\omega^2 I + j\omega\beta_1 + \Lambda)^{-1}\beta_2 \end{cases}$$

PERSPECTIVE :
Développement et adaptation de la méthode aux cas NL

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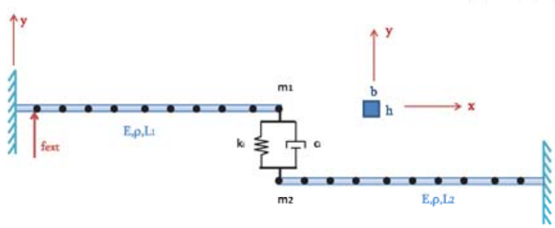
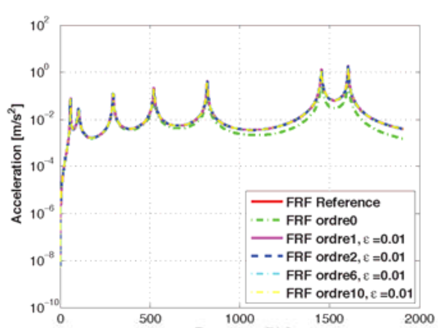
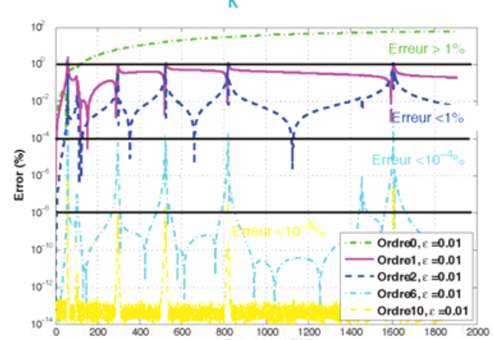
Example







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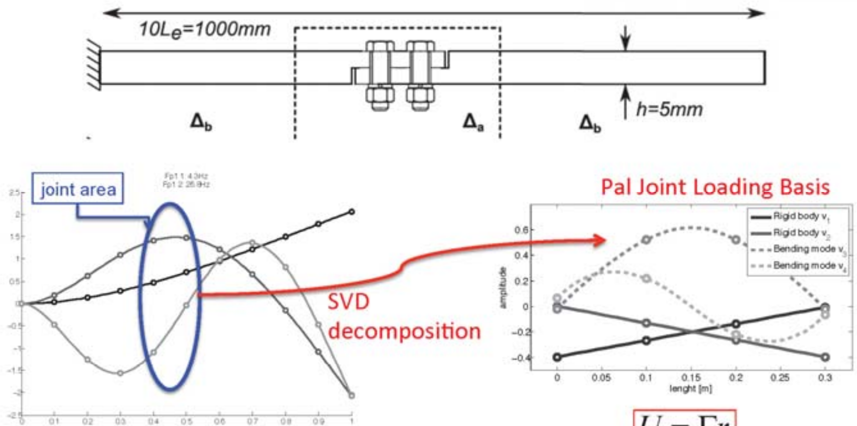
Principal Joint Loading Basis



$$U = \Gamma r$$

EigenModes

$$U = \Phi q$$

$$\ddot{q}_1 + \omega_{01}{}^2 q_i = b_1 f(t) + f_{NL1}(r_1, \dot{r}_1)$$
$$\ddot{q}_2 + \omega_{02}{}^2 q_2 = b_2 f(t) + f_{NL2}(r_2, \dot{r}_2)$$

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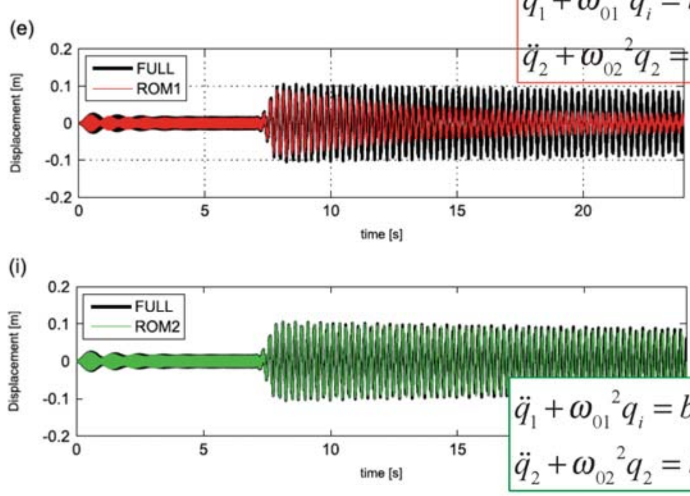
Back to the 2nd example

$$\ddot{q}_1 + \omega_{01}{}^2 q_i = b_1 f(t) + f_{NL1}(q_1, \dot{q}_1)$$
$$\ddot{q}_2 + \omega_{02}{}^2 q_2 = b_2 f(t) + f_{NL2}(q_2, \dot{q}_2)$$



$$\ddot{q}_1 + \omega_{01}{}^2 q_i = b_1 f(t) + f_{NL1}(r_1, \dot{r}_1)$$
$$\ddot{q}_2 + \omega_{02}{}^2 q_2 = b_2 f(t) + f_{NL2}(r_2, \dot{r}_2)$$

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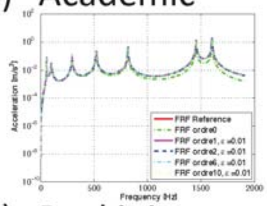
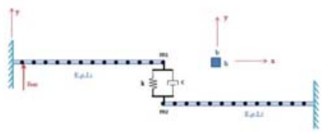
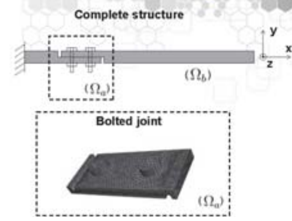
Work in Progress

1) Academic

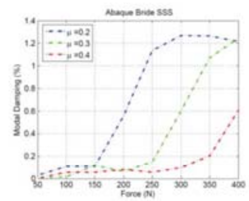






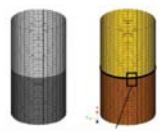
2) Real joint

Bride SSS



3) Complete Structure

Sylda5



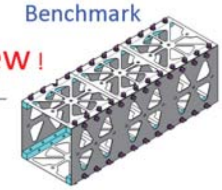
Bride T90

Modèle complet

FEMTO-Supmeca

Benchmark

New !



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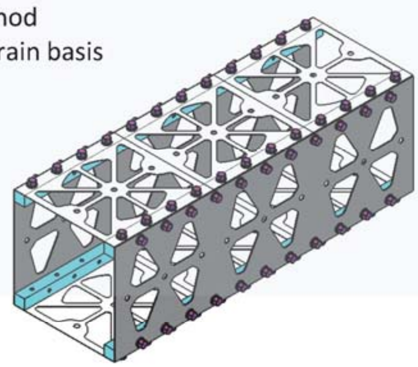
Conclusions

- Couplings induced by :
  - Localized joints
  - Non-linearities
- Two Ideas :
  - Asymptotic method
  - Principal joint strain basis
- Benchmark



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Chiara Gastaldi: Modeling and Model Tuning in Under-Platform Damper Dynamics

4TH WORKSHOP ON JOINTS MODELLING **DARTINGTON, OCTOBER 2015**

Modeling and model tuning in under-platform damper dynamics

Chiara M. GASTALDI - Ph.D. student

Politecnico di Torino
DIMEAS Dept . Mechanical and Aerospace Engineering
AERMEC laboratory http://www.aermec-dimec.polito.it

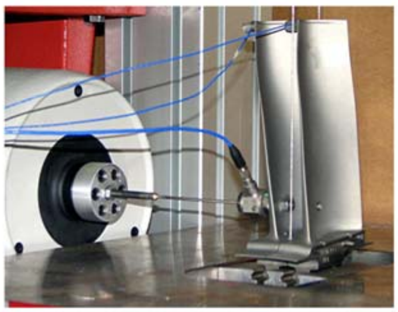
M.M. Gola C. Gastaldi October 2015
DIMEAS - POLITO

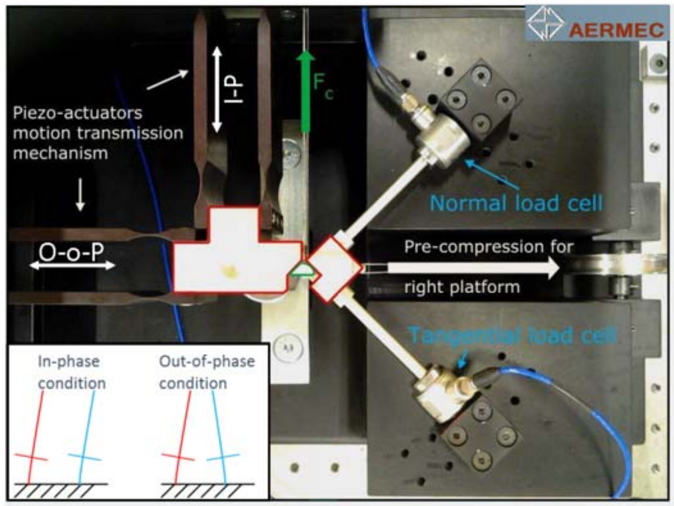
1 - Purpose and contents

The need of **reliable models** for the design of **Under-platform Dampers (UPDs)** for turbine blades has led to a considerable amount of technical literature in the last three decades.

At **AERMEC** we believe that investing in the direct experimental investigation is useful to:

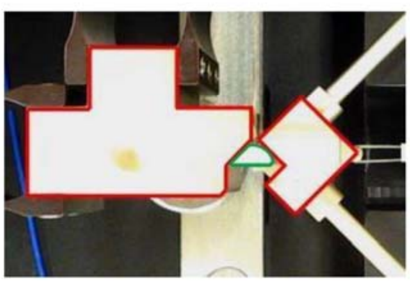
- provide **direct contact information** (contact stiffness and friction coefficient) to develop **reliable and exportable numerical models;**
- avoid a "black box" approach.

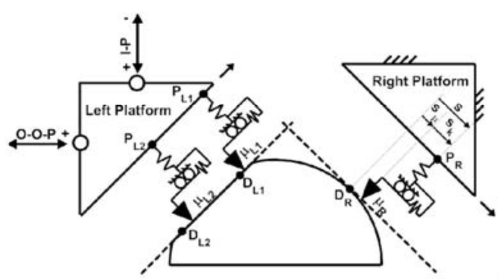


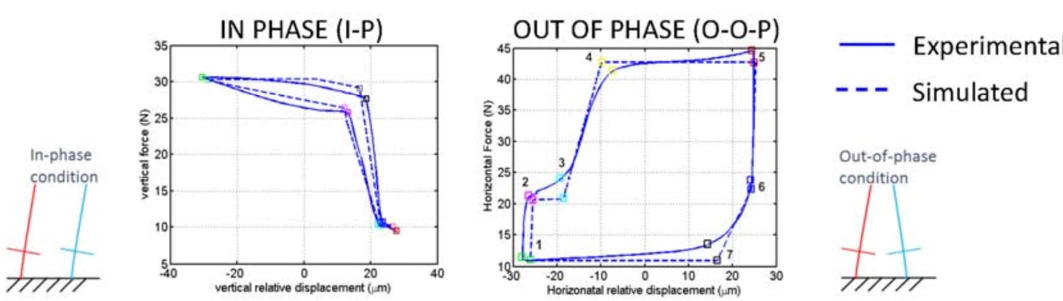


1 - Purpose and contents

A routine capable of computing the damper contact forces as a function of the platforms input displacements was successful in capturing the damper behaviour under different regimes.







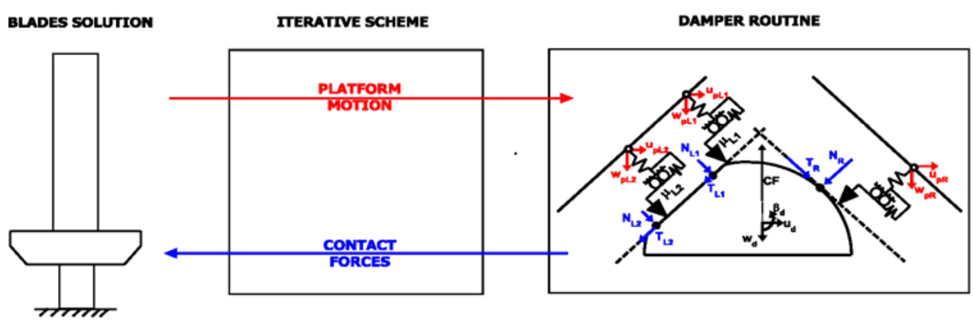
1 - Purpose and contents

PROS:

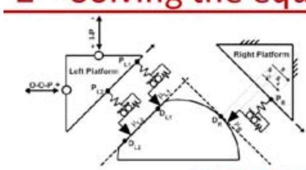
- Direct confrontation with the experimental setup -> contact parameter estimation
- The routine can easily be integrated into a complete FE model of a bladed disk

CHALLENGES:

- Solving the equilibrium equations: methods, convergence and time effectiveness
- Representing the contact and estimating contact parameters



2 – Solving the equilibrium: DTI vs MHBM



$$[M]\{\ddot{u}\}+[K_D]\{u\}=\{F_e\}+[K_P]\{u_P\}+[K_S]\{s\}$$

DIRECT TIME INTEGRATION

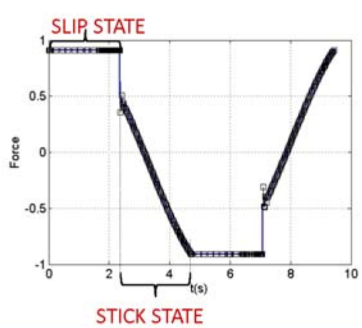
+ Exact solution
- **Time consuming**

PIECEWISE ADAPTIVE DTI SCHEME:
the size of the integration step Δt is a function of the contact state.

MULTI-HARMONIC BALANCE METHOD

+Time efficient
- Approximate solution

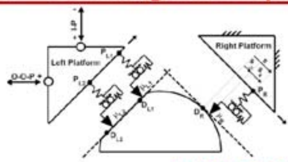
FRICTION-INDUCED NON-LINEARITY CAN BE SEEN AS A TIME SEQUENCE OF LINEAR SYSTEMS.



In the case of a curved-flat damper over a 2 period 100 Hz simulation

	Standard 4^{th} o. RK	Piecewise adaptive 4^{th} o. RK
Integration steps	≈1350	≈1050

2 – Solving the equilibrium: DTI vs MHBM



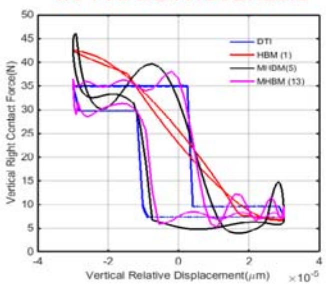
$$[M]\{\ddot{u}\}+[K_D]\{u\}=\{F_e\}+[K_P]\{u_p\}+[K_S]\{s\}$$

DIRECT TIME INTEGRATION
+ Exact solution
- **Time consuming**

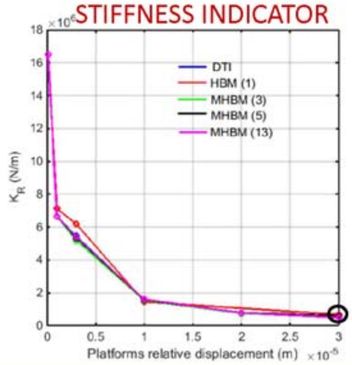
MULTI-HARMONIC BALANCE METHOD
+Time efficient
- **Approximate solution**

A CONVERGENCE STUDY is performed using DTI results as a benchmark

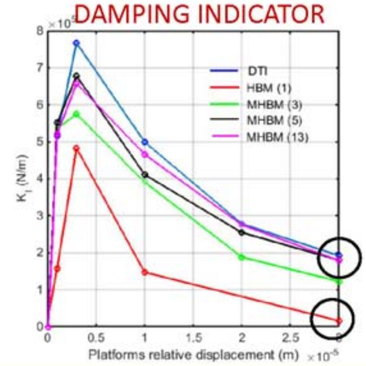
IN-PHASE HYSTERESIS



STIFFNESS INDICATOR



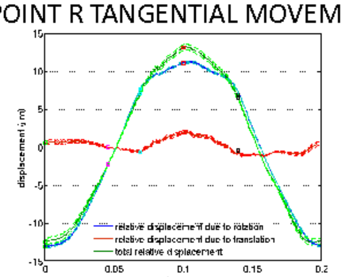
DAMPING INDICATOR



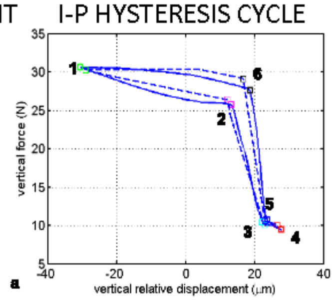
3 – Experimental evidence available…how do we use it?

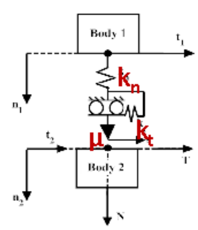
A **crucial** part in reproducing the damper behaviour is **modelling the contact.**
All contact models require tuning of contact parameters…
The accuracy of force and kinematic measurements allows a trustworthy comparison between numerical and experimental results.

POINT R TANGENTIAL MOVEMENT



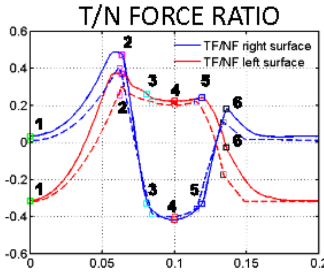
I-P HYSTERESIS CYCLE



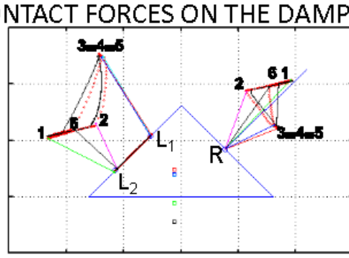


Quantity	σ
Right C. Force	3%
Left C. Force	3-5%
Left C. Force application point	0.5 mm
Platforms relative movement	0.08 μm
Damper rotation	5%
Right contact point disp.	5%

T/N FORCE RATIO

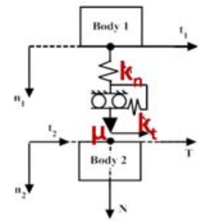


CONTACT FORCES ON THE DAMPER



4 – Tuning the contact: a search based manual tuning protocol

This technique is based on the **understanding** of the damper contact states and requires a **skilled operator**.
Remark: it may not be applicable to all hysteresis cycles since it requires the presence of particular contact states



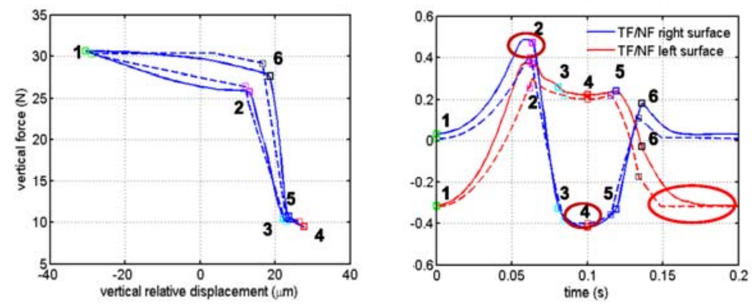
The friction coefficients can be estimated by looking at the measured ratio of tangential and normal component of the contact forces in the experimental diagrams:

$\mu_R \cong 0.4$ $\mu_{L2} \cong 0.32$

— Experimental
- - Simulated

I-P HYSTERESIS CYCLE

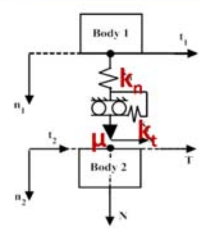
TANGENTIAL/NORMAL FORCE RATIO



4 – Tuning the contact: a search based manual tuning protocol

A **crucial** part in reproducing the damper behaviour is **modelling the contact.**
All contact models require tuning of contact parameters…

1. SEARCH BASED MANUAL TUNING PROTOCOL: understanding the damper

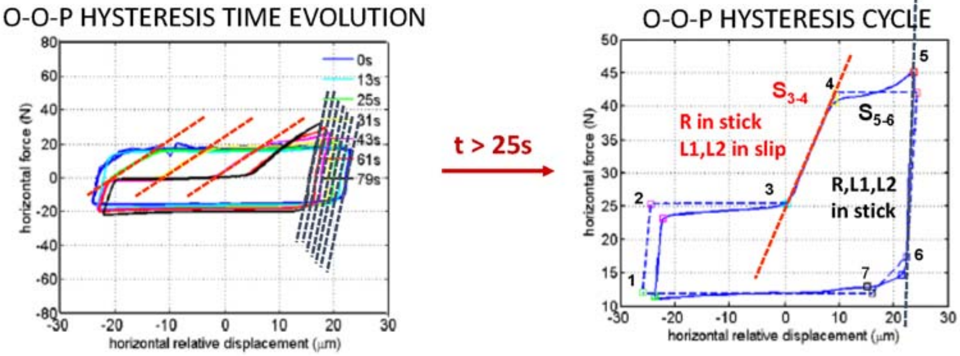


The contact stiffness values can be estimated by at the slope of particular portions of the hysteresis cycle. Knowledge of the contact state during that portion is required.

— Experimental
- - Simulated

O-O-P HYSTERESIS TIME EVOLUTION

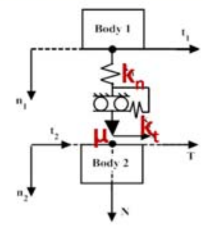
O-O-P HYSTERESIS CYCLE

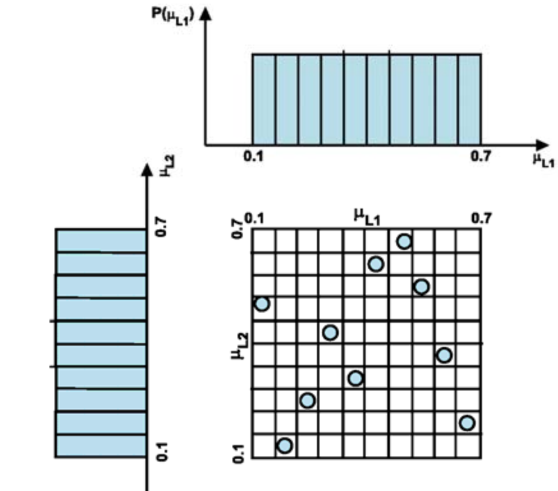


4 – Tuning the contact: an automatic random sampling alternative

A **crucial** part in reproducing the damper behaviour is **modelling the contact.**
All contact models require tuning of contact parameters…

2. RANDOM SAMPLING BASED TUNING PROTOCOL





EXAMPLE WITH A 2 PARAMETER SPACE AND N_S=10

1. **Each contact parameter is assigned a (uniform) distribution**
2. **A large number of samples (e.g. 5000) is drawn from the distributions through the Latin hypercube Sampling method.**
3. A series of indicators quantifying the similarity between simulated and measured cycle are computed for each sample (based on similarity of forces, dissipated energy, kinematical parameters).
4. Only the samples whose indicators below a prescribed tolerance are selected

4 – Tuning the contact: an automatic random sampling alternative

A **crucial** part in reproducing the damper behaviour is **modelling the contact.**
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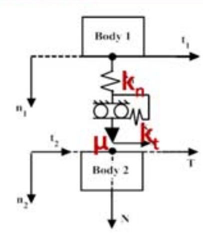
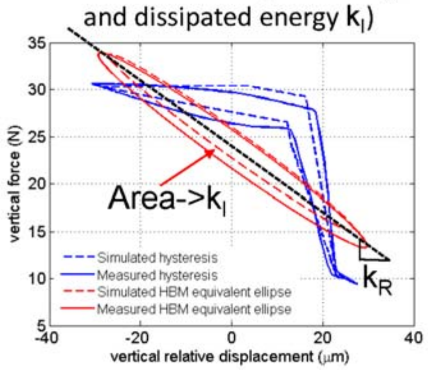


SIMILARITY INDICATORS

Similarity between 1st o. Harmonic of contact forces (stiffness k_R and dissipated energy k_I)



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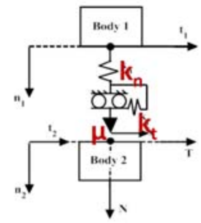
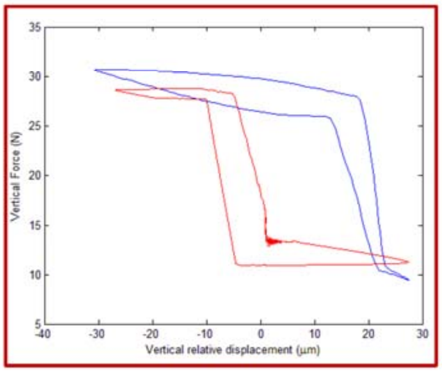


2. RANDOM SAMPLING BASED TUNING PROTOCOL

SIMILARITY INDICATORS

Similarity between hysteresis shapes



SIMILAR STIFFNESS (k_R)
SIMILAR DAMPING (k_I)
DIFFERENT BEHAVIOUR…
NOT EXPORTABLE OUTSIDE
LOCAL TUNING!

1. Each contact parameter is assigned a (uniform) distribution
2. … samples (e.g. 5000) is … tributions through the … mpling method.
3. … **ators quantifying the** … **n simulated and** … are computed for each sample (based on similarity of forces, dissipated energy, kinematical parameters).
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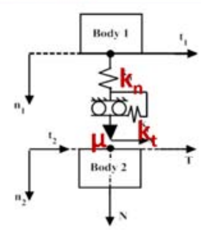
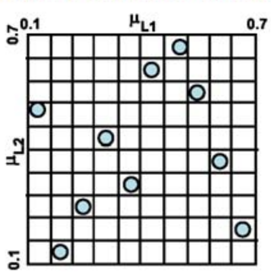
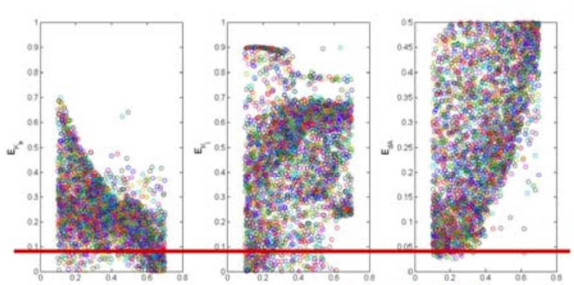


2. RANDOM SAMPLING BASED TUNING PROTOCOL

EXAMPLE WITH
A 2 PARAMETER
SPACE





TOLERANCE

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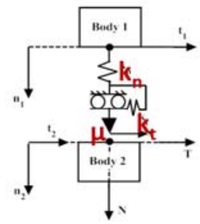
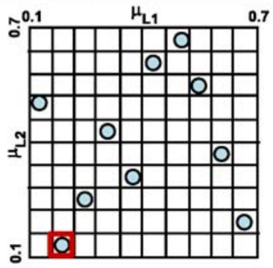
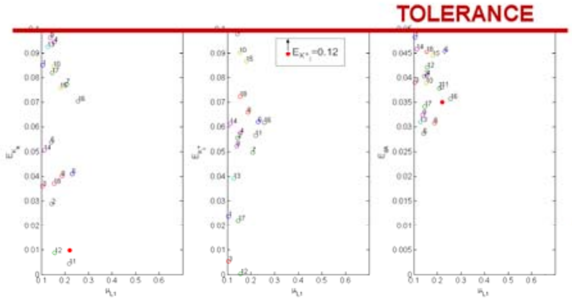


2. RANDOM SAMPLING BASED TUNING PROTOCOL

EXAMPLE WITH
A 2 PARAMETER
SPACE



TOLERANCE



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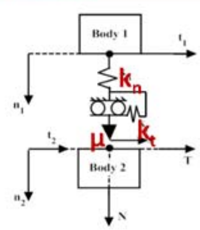
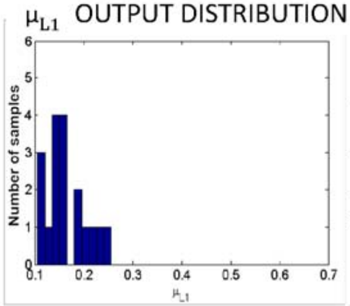


2. RANDOM SAMPLING BASED TUNING PROTOCOL

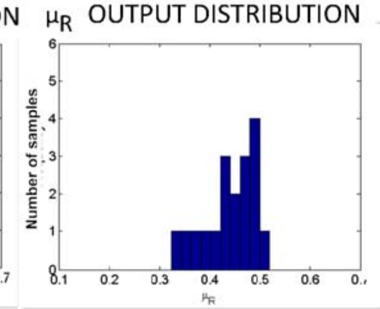
The chosen sample's parameters are in good agreement with the set found through the operator based results.

ID	μ_{L1}	μ_{L2}	μ_R	k_n	k_t
Random Sampling -8	0.19	0.28	0.43	2.14	1.61
Operator (Ph.D. Liu T.) 	0.22	0.32	0.40	2.4	1.6

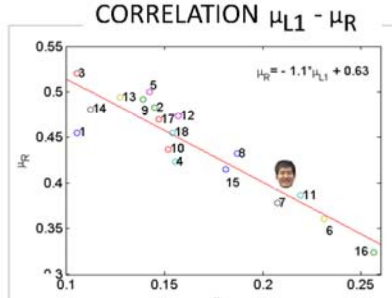
μ_{L1} OUTPUT DISTRIBUTION



μ_R OUTPUT DISTRIBUTION



CORRELATION μ_{L1} - μ_R



4 – Tuning the contact: an automatic random sampling alternative

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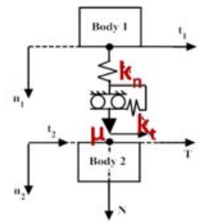

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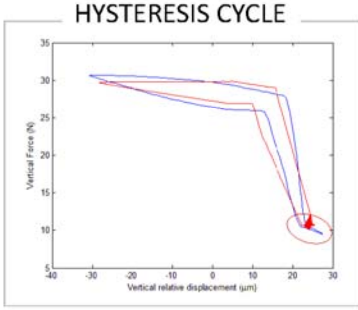
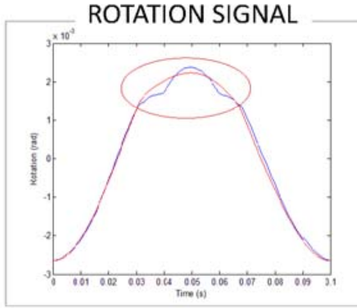
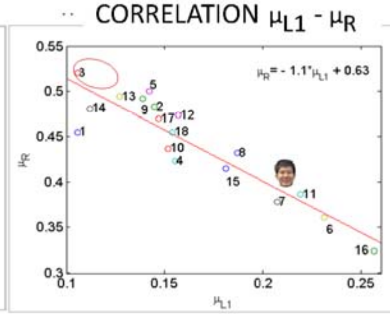
HYSTERESIS CYCLE



ROTATION SIGNAL



CORRELATION μ_{L1} - μ_R



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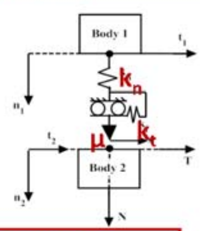

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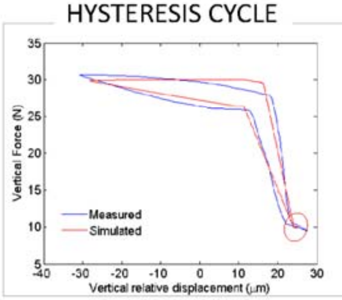
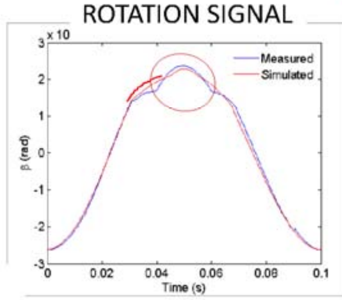
PROPERLY CHOSEN SIMILARITY INDICATORS ARE THE KEY TO THE SELECTION OF THE CORRECT FRICTION PARAMETERS.. OTHERWISE INDETERMINACY!!!!

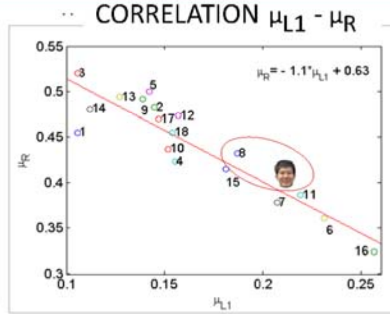
HYSTERESIS CYCLE



ROTATION SIGNAL

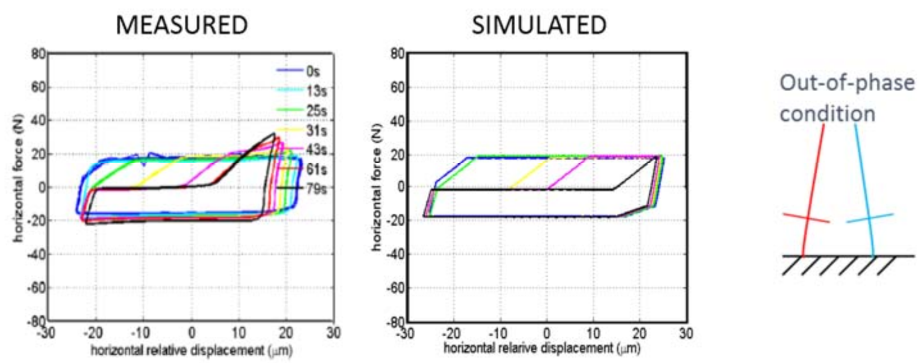


CORRELATION μ_{L1} - μ_R



5 – Conclusions

- Direct experimental investigation shows that the damper behaviour **evolves** significantly and in a **repeatable manner** with time due to changing friction parameters.
- The numerical model, properly tuned is capable of correctly reproducing this evolution.
- The correct estimation of the friction parameters ensures a predictive and exportable model.
- Regardless of the chosen tuning technique, the key to a predictive and exportable model is a direct experimental investigation.



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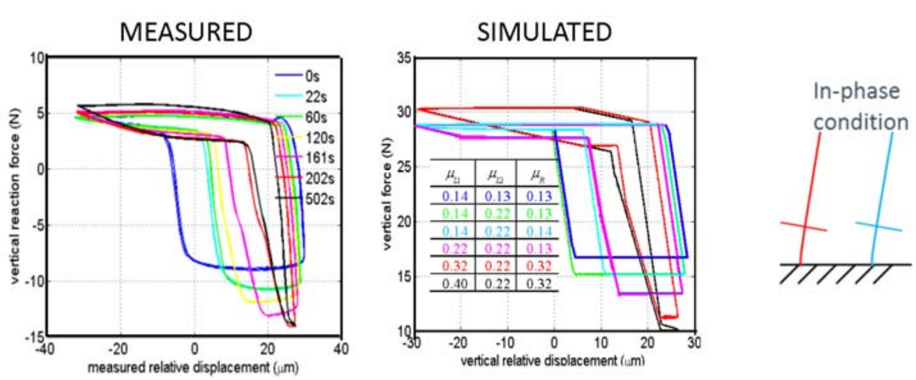


Closure

Thank You for your kind attention!

References

[1] Gola, M. M., Bragas Dos Santos, M., Liu, T., 2012. “Measurement of the scatter of underplatform damper hysteresis cycle: experimental approach”. In: Proceedings of ASME IDETC 2012

[2] Gola, M. M., Liu, T., M. Bragas Dos Santos M., 2013. “Investigation of under-platform damper kinematics and its interaction with contact parameters (nominal friction coefficient)”. In Proceedings of World Tribology Congress 2013

[3] Gola, M. M., Liu, T., 2014. “A direct experimental numerical method for investigations of a laboratory underplatform damper behavior”. International Journal of Solids and Structures,51(25-26), pp. 4245-4259.

[4] Gola, M. M., Gastaldi, C., 2014. “Understanding complexities in underplatform damper mechanics”. Proceeding of ASME Turbo Expo (2014)

[5] Gola, M. M., Gastaldi, C.,2015 “A random sampling strategy for tuning contact parameters of under-platform dampers”. In Proceedings of ASME Turbo Expo (2015)

Advanced friction models and fretting experiments of polymeric coatings

Andreas A. Polycarpou
Department Head and Kotzebue Professor
Department of Mechanical Engineering
Texas A&M University

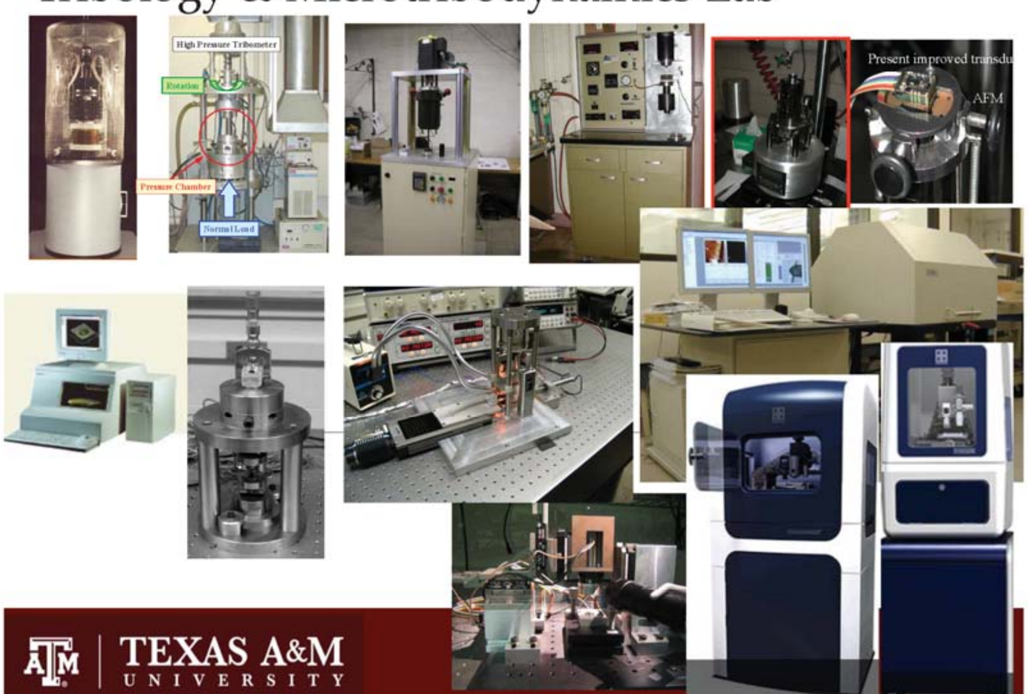
Presented at the
Fourth Workshop on Joints Modeling, Dartington, UK
October 18-21, 2015

TEXAS A&M UNIVERSITY

Polycarpou Research Interests

- Tribology, Microtribodynamics, Thin Films
  - Basic interfacial phenomena at small scales
    - NSF, DARPA
  - Microsystems (magnetic storage, MEMS)
    - **INSIC, Seagate, Samsung**
  -
  - Engineering surfaces/tribology
    - Driven by industry needs
    - Air-conditioning and refrigeration companies
    - Automotive compnies
    - Rail association
    - Oil & gas companies

Tribology & Microtribodynamics Lab



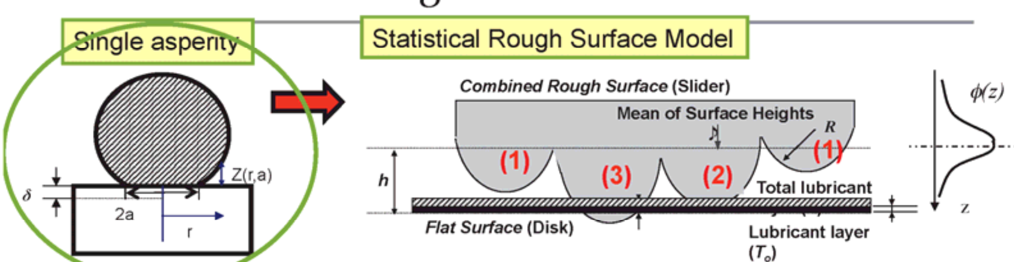
TEXAS A&M UNIVERSITY

Microtribodynamics

- Eliminate phenomenological models
- Physics-based analytical coupled models
- Develop nanomechanics models applicable to small-scale systems
- Successfully investigated the effect of adhesion and roughness in nanocontact mechanics and nm-thick layer effects

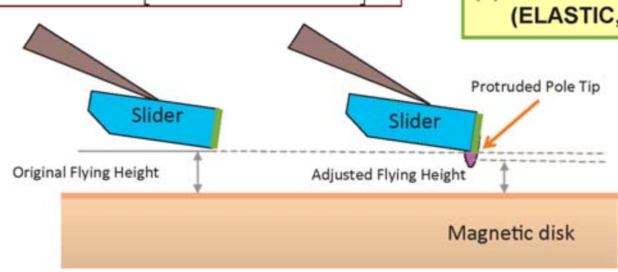
III. Design Optimization via DOE-Assisted Parametric Study

Adhesive friction rough surface statistical model



$$F_S(a) = 2\pi\int_a^\infty \frac{8}{3}\frac{\Delta\gamma}{\varepsilon}\left[\left(\frac{\varepsilon}{Z(r,a)}\right)^3 - \left(\frac{\varepsilon}{Z(r,a)}\right)^9\right] rdr$$

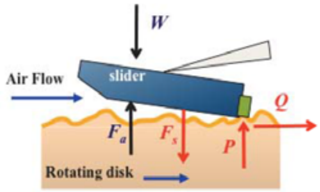
(1) NON-CONTACTING ASPERITIES
(2) LUBRICANT CONTACTING ASPERITIES
(3) SOLID CONTACTING ASPERITIES (ELASTIC, ELASTIC-PLASTIC)

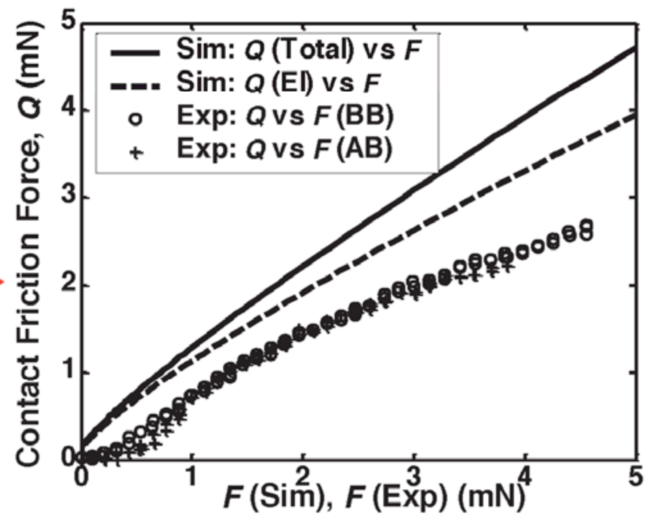


Lee and Polycarpou, JoT (2004), Suh and Polycarpou, JAP (2005), Shi Polycarpou, JCIS (2004), Xue and Polycarpou, JAP, JCIS, JAST (2006-08), Yu and Polycarpou, JCIS (2004), Vakis and Polycarpou, Micr Tech, Trib Let (2009-2013)

Friction Model Validation

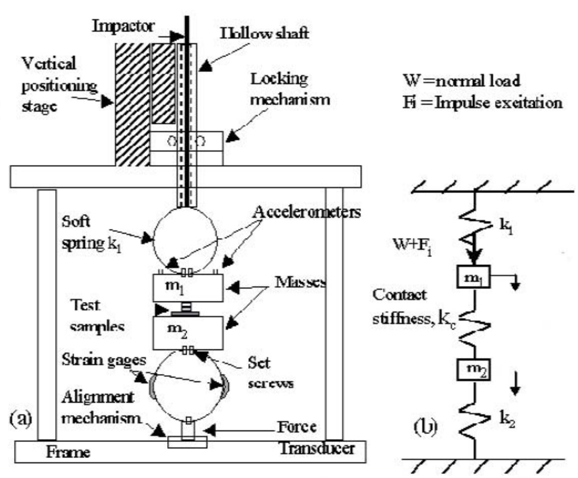
Experiments performed independently by IBM Almaden collaborators





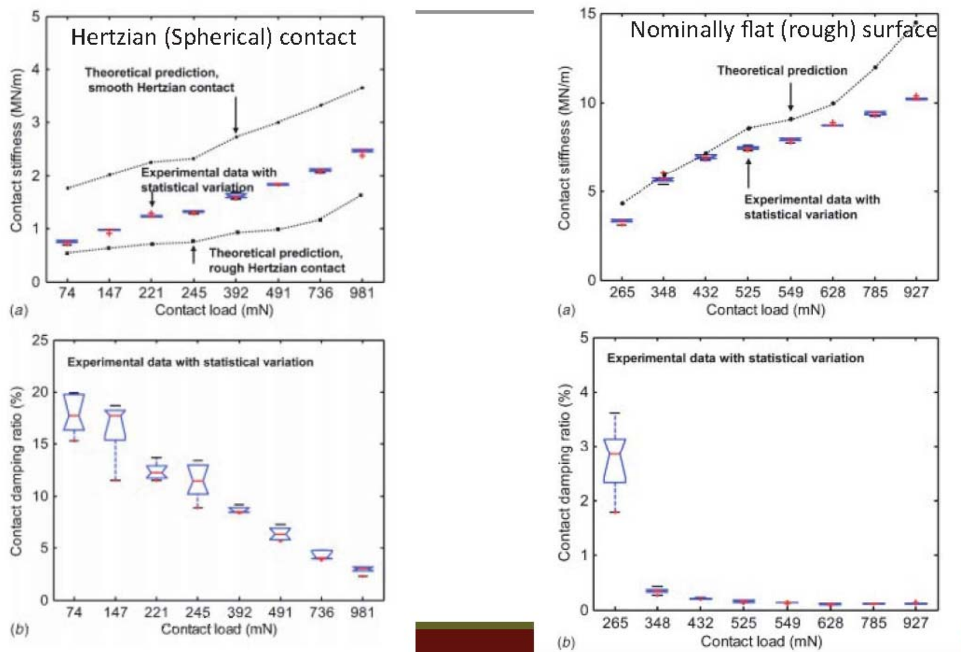
Suh et al Trib Let (2007)

Contact Stiffness and Damping

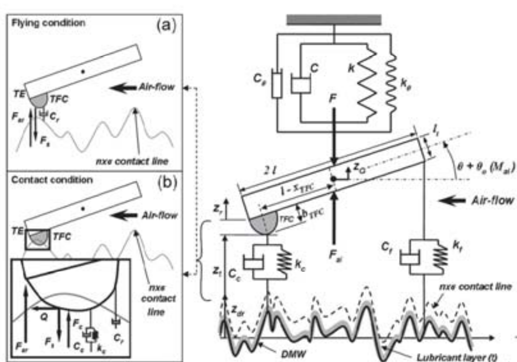


Shi and Polycarpou, JSV, 2004

Contact Stiffness and Damping

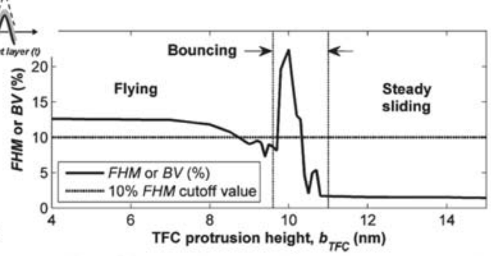


Couple contact and dynamic models



Schematic of vibration and dynamic model for HDI

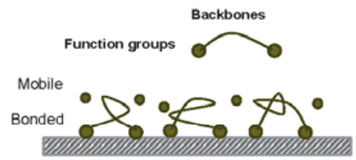
Lee and Polycarpou, Microsys Tech, 2004
Vakis, A.I., Lee, S.C., Polycarpou, A.A. : Dynamic head-disk interface instabilities with friction for light contact (surfing) recording. IEEE Trans. Magn. 45, 4966- 4971 (2009).



Possibility of bouncing vibration at different thermal protrusion height

Molecularly thin lubricant films under shear

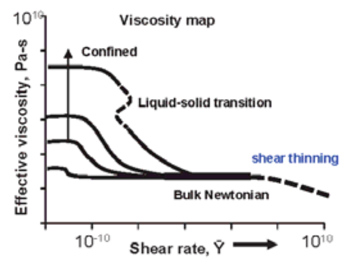
1. Bonded and mobile layers



2. Shear thinning

Much higher viscosity under confinement but decreases with shear rate (shear thinning). Effective viscosity varies with the shear rate:

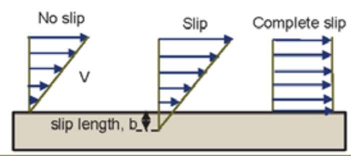
$$\mu \propto \dot{\gamma}^{-0.9}$$



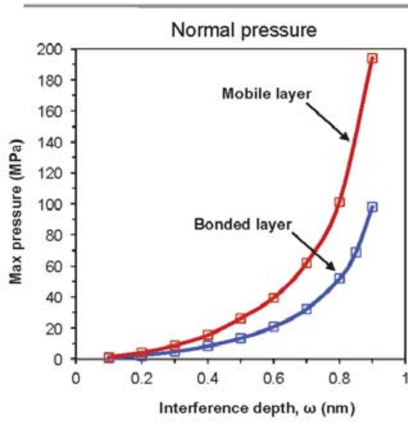
3. Slippage at the wall

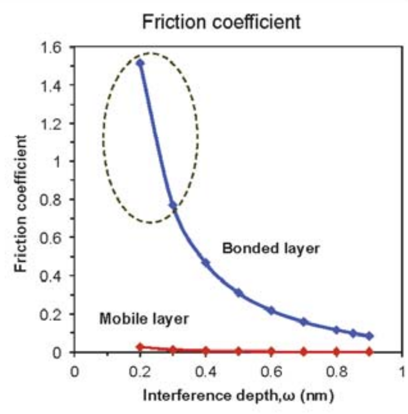
Slip length of lubricants (PFPE) in HDD is unknown. A slippage factor f^* is defined to take into account the slippage at the boundary.

$$f^* = (U - U_W)/U = h/(b+h).$$



Pressure and friction coefficient of bonded and mobile layers

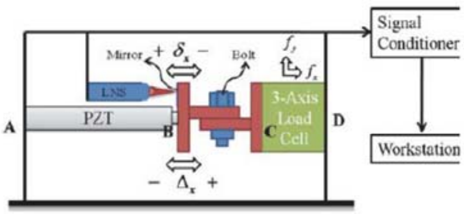




Slippage in the mobile layer leads to:
Higher pressure (bearing capacity)
Lower friction

Zhang, Polycarpou, Trib Let (in review), including temperature effects

Basic Fretting/ microslip



Experimental setup that isolates, decouples system dynamics and interfacial slip Eriten, Polycarpou, Bergman (E.g., Exp Mech (2011)

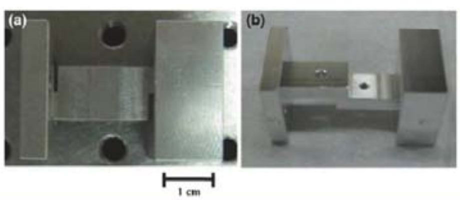


Fig.
ove

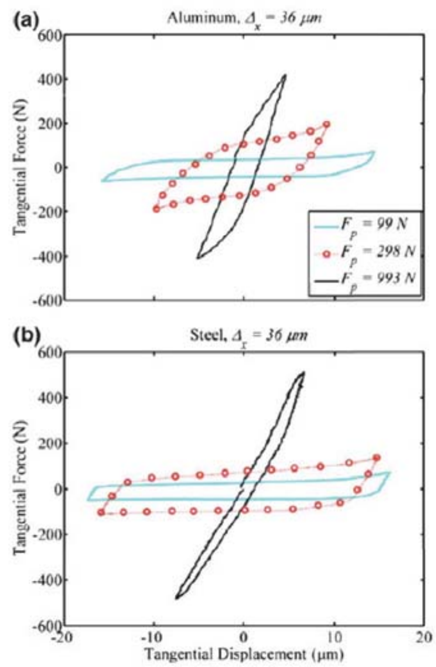
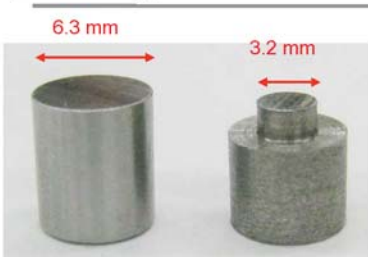


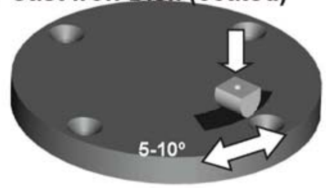
Fig. 10 Average of the last two fretting loops for (a) aluminum and (b) steel joints under constant imposed displacement of 36 μm and varying preload values

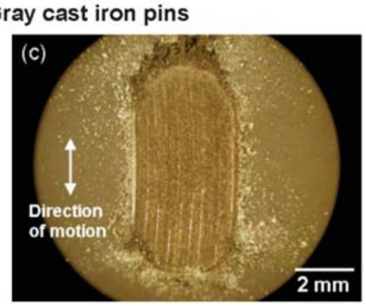
Sponsor driven :Tribotesting Simulating Fretting Conditions (dithering)



Gray cast iron pins

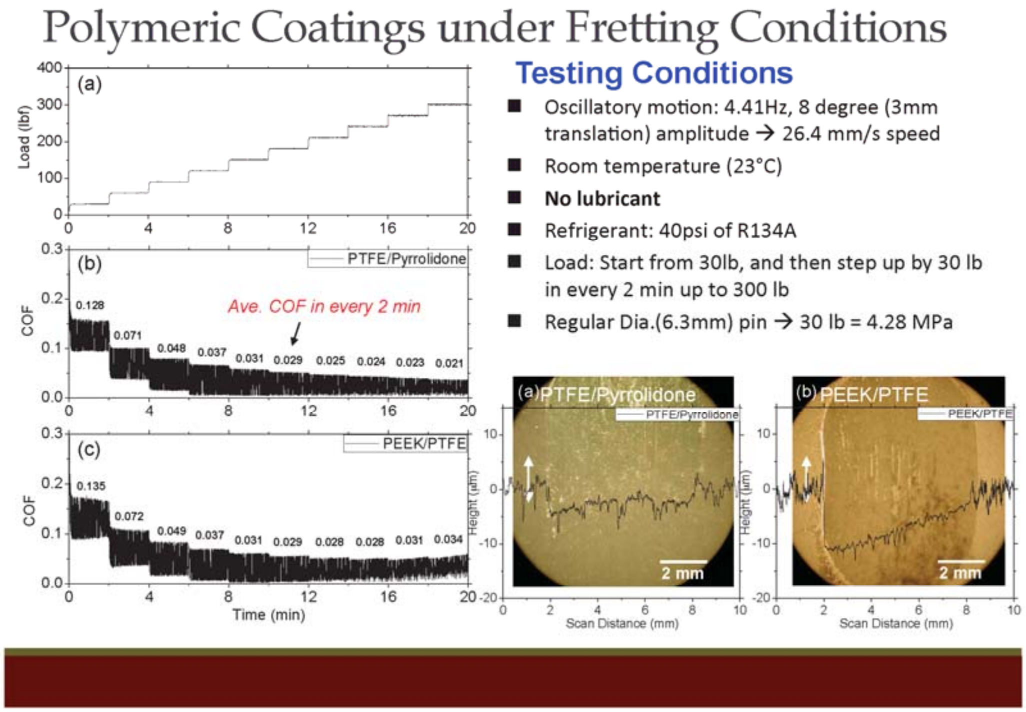
Gray Cast Iron Disk (coated)

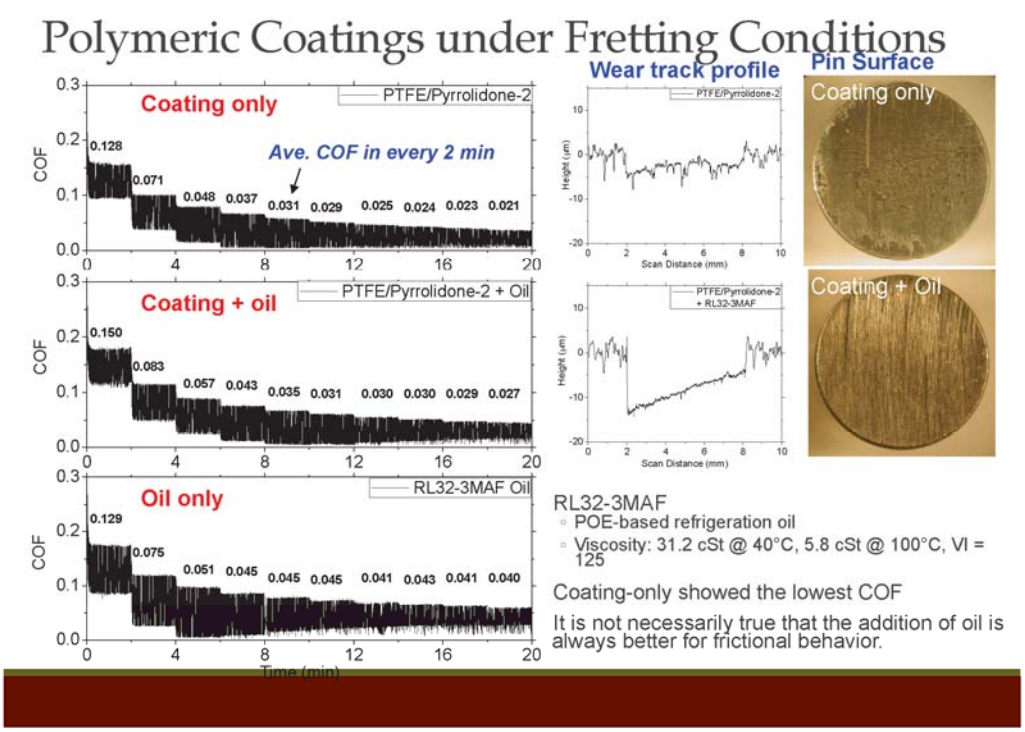




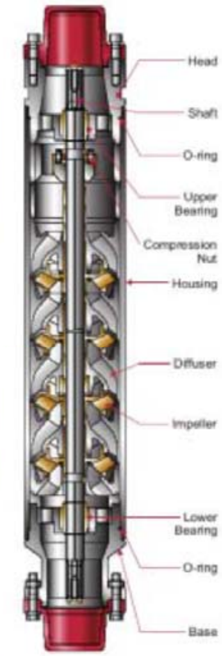
High bearing advanced polymeric coatings (spray-coated) [20 μm thick]
DuPont® 958-303 (PTFE/Pyrrolidone-1) DuPont® 958-414 (PTFE/Pyrrolidone-2) Whitford Xylan® 1052 (Resin/PTFE/MoS_2) Fluorolon® 325 (PTFE/MoS_2) Impreglon® 218 (Fluorocarbon) 1704 PEEK/PTFE® (PEEK/PTFE) 1707 PEEK/Ceramic/PTFE® (PEEK/Ceramic) ATSP-based Coatings
Gray Cast Iron (Dura-Bar® G2) Substrate Shoe and Pin

Yeo, Polycarpou, Trib Intern (2014)


Polymeric Coatings under Fretting Conditions
Testing Conditions
Oscillatory motion: 4.41Hz, 8 degree (3mm translation) amplitude → 26.4 mm/s speed
Room temperature (23°C)
No lubricant
Refrigerant: 40psi of R134A
Load: Start from 30lb, and then step up by 30 lb in every 2 min up to 300 lb
Regular Dia.(6.3mm) pin → 30 lb = 4.28 MPa
(a)
Load (lbf)
(b)
PTFE/Pyrrolidone
Ave. COF in every 2 min
COF
0.128
0.071
0.048
0.037
0.031
0.029
0.025
0.024
0.023
0.021
(c)
PEEK/PTFE
0.135
0.072
0.049
0.037
0.031
0.029
0.028
0.028
0.031
0.034
Time (min)
(a) PTFE/Pyrrolidone
(b) PEEK/PTFE
Height (µm)
2 mm
Scan Distance (mm)

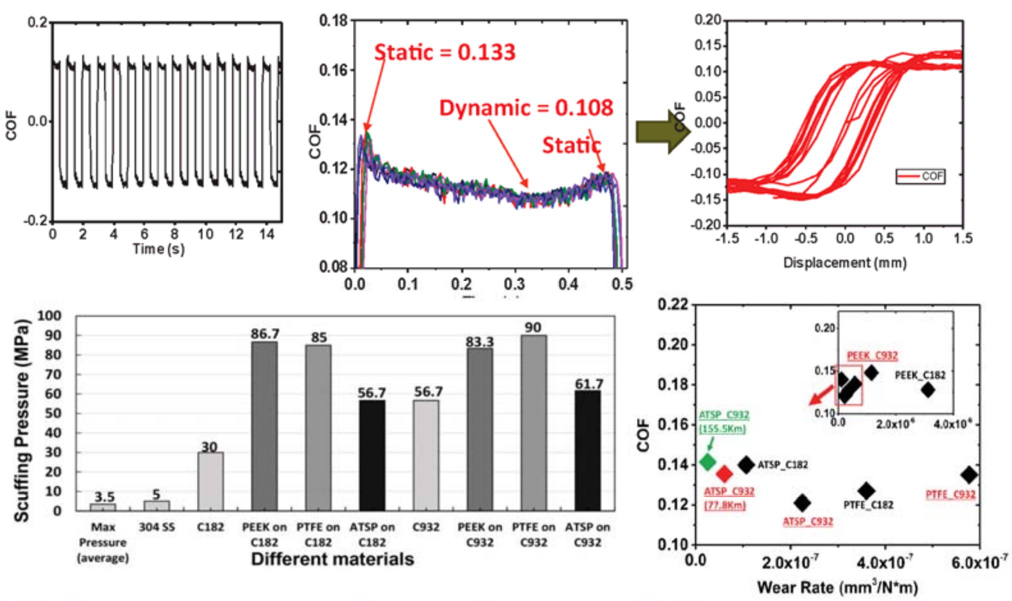

Polymeric Coatings under Fretting Conditions
Coating only
PTFE/Pyrrolidone-2
Ave. COF in every 2 min
COF
0.128
0.071
0.048
0.037
0.031
0.029
0.025
0.024
0.023
0.021
Coating + oil
PTFE/Pyrrolidone-2 + Oil
0.150
0.083
0.057
0.043
0.035
0.031
0.030
0.030
0.029
0.027
Oil only
RL32-3MAF Oil
0.129
0.075
0.051
0.045
0.045
0.045
0.041
0.043
0.041
0.040
Time (min)
Wear track profile
Pin Surface
Coating only
Coating + Oil
RL32-3MAF
POE-based refrigeration oil
Viscosity: 31.2 cSt @ 40°C, 5.8 cSt @ 100°C, VI = 125
Coating-only showed the lowest COF
It is not necessarily true that the addition of oil is always better for frictional behavior.

Fretting and extreme conditions tribolog



High environmental pressures
Temperatures: -200 oC to 600 oC
Corrosive and abrasive environmnets

Fretting and extreme conditions tribology: oil&gas

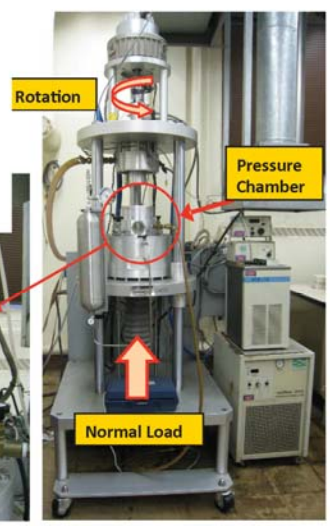


Summary

- Significant advances in continuum-based friction modeling
- Fretting/energy dissipation in other than gas turbine applications
  - New material systems/combinations needed for extreme conditions
  - Low friction
  - Low wear
  - Joint damping may or may not be important (because of the presence of fluids)
-

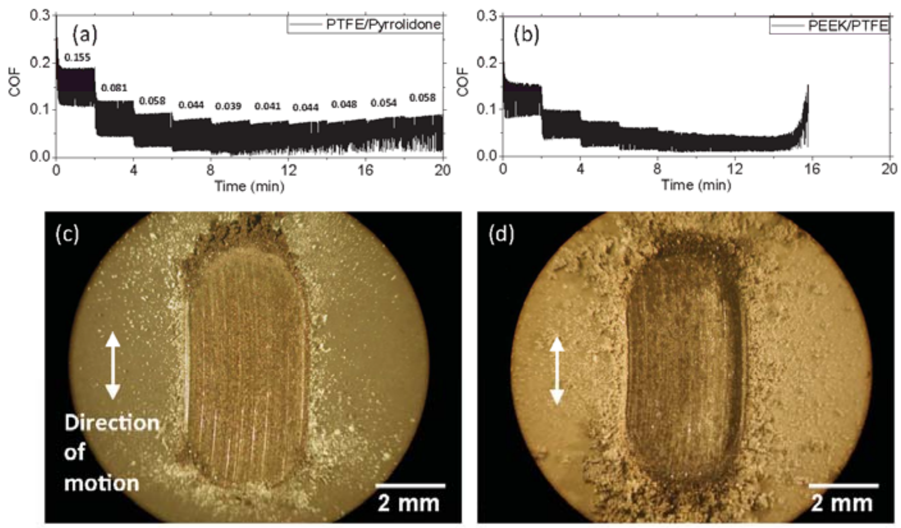
Specialized Tribometer Testing Simulating Compressor Conditions

	Testing conditions
Temperature (°C)	23 (room temperature)
Environment (refrigerant)	R-134A at 40 psi
Reciprocating frequency (Hz)	4.4
Reciprocating amplitude (mm)	3
Average linear sliding speed (mm/s)	26.4
Contact geometry	Nominally flat surface contact
Normal load (N)	133 - 1334
Nominal contact pressure (MPa)	4.28 – 42.8 (6.3mm pin), 16.58 – 165.8 (3.2 mm pin)
Test duration (min)	20 (up to failure)



Environmental Chamber

In-situ friction coefficient of(a)PTFE/Pyrrolidone and (b)PEEK/PTFE coating under unlubricated scuffing experiments. Optical images of worn surfaces of (c) PTFE/Pyrrolidone and (d)PEEK/PTFE. Higher contact pressure

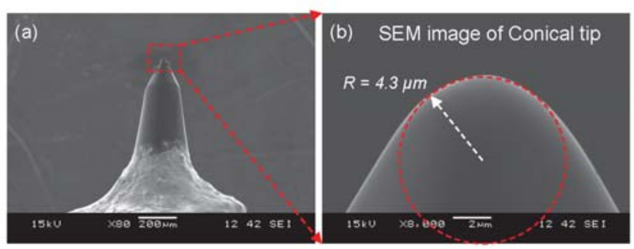


ATM

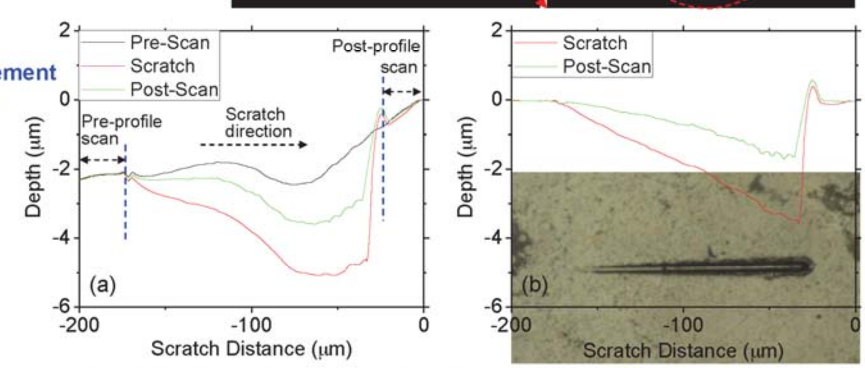
Elastic Recovery Property - Scratch Tests

3-D OmniProbe™

- Longer scratch length (15cm), and normal displacement (100μm)
- Larger lateral (5N) and normal (2.7N) force

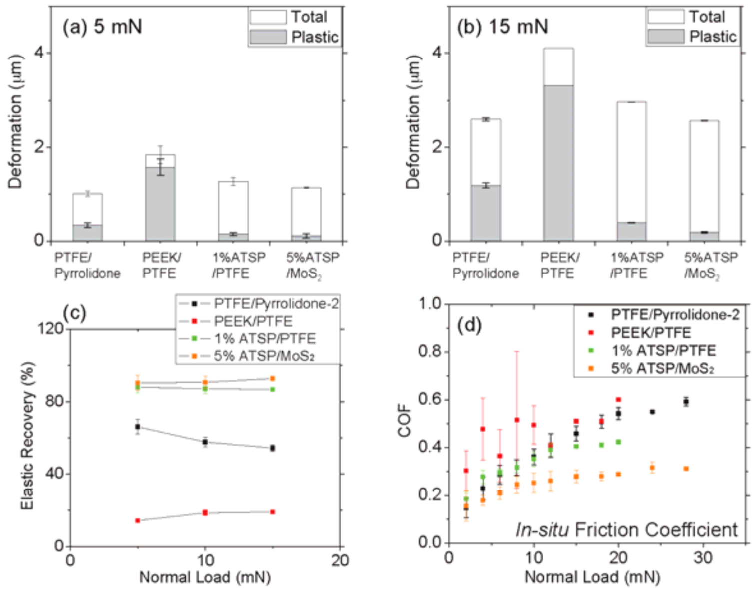


3-Step Measurement Process



Total deformation = Permanent (Plastic) deformation + Elastic deformation

Elastic Recovery Property - Scratch Tests

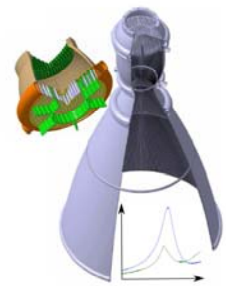


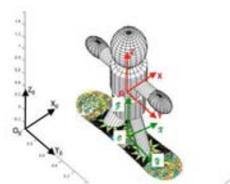
Polymers with higher elastic recovery showed better frictional behavior → Due to smaller real contact area


Imperial College
London
Frequency methods for contact mechanics:
personal experience and challenges for the future
L. Salles
VUTC - Imperial College London, UK
Dartington
20th October 2015

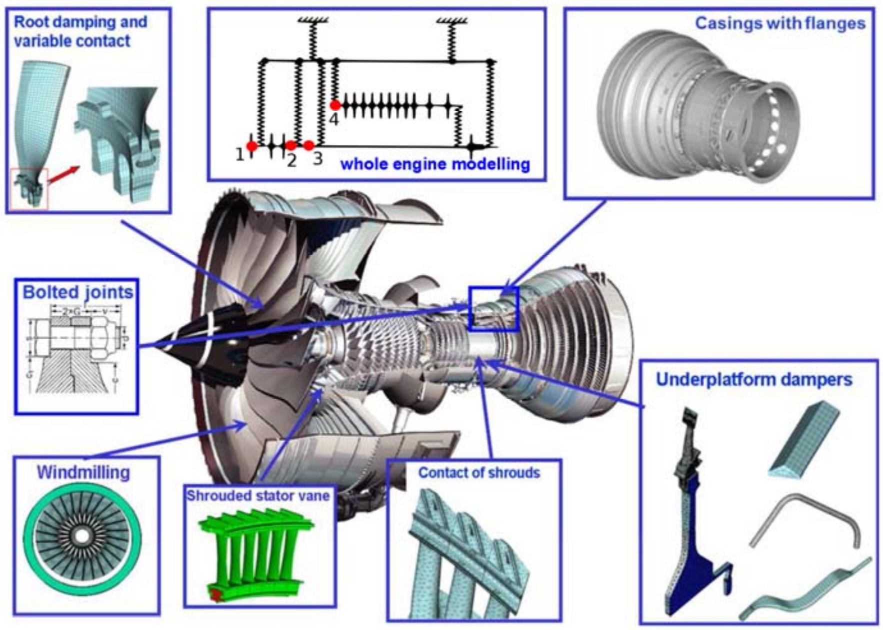
Biography

- 2006 Msc at BMSTU and Keldysh Research Center for Rocket engines: *High Frequency instability in combustion chamber of a liquid rocket engine*
- 2010 PhD at Ecole Centrale de Lyon and Bauman Moscow State Technical University: *Fretting-wear in contact joints under dynamical loading*
- 2012-now: Research Associate at Rolls-Royce VUTC Imperial College





Joints in aeroengine



Biography Harmonic Balance Method Contact algorithms Challenges

Outline

Biography

Harmonic Balance Method

Contact algorithms

Challenges for the future

Salles Loic Dartington 20/10/2015 Frequency methods for contact mechanics

Equation of motion

Equation of motion

$$M\ddot{U} + C\dot{U} + KU = F_c(U, \dot{U}, W) + F_{ex}(t)$$

Frequency equation of motion

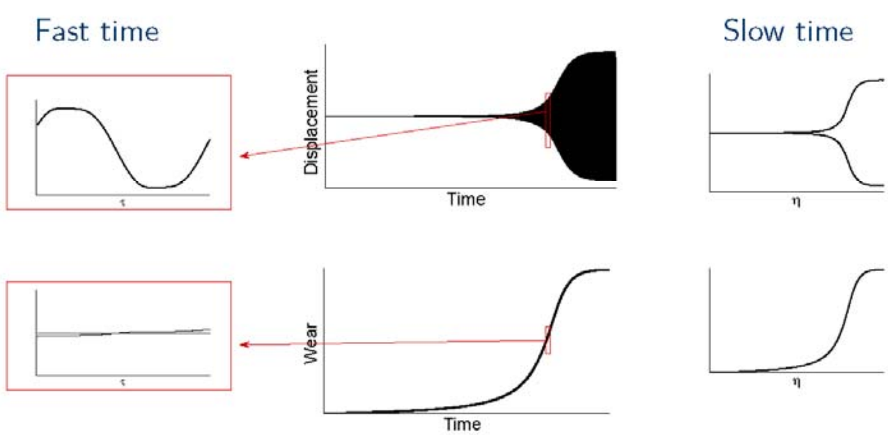
$$Z\tilde{Q} = \tilde{F}_{ex} - \tilde{F}_{NL}(\tilde{Q}, \tilde{Q})$$

Non-linear system with Hybrid reduced model

$$\tilde{Q}_r = H(\omega)\left(\tilde{F}_{ex} - \tilde{F}_{NL}(\tilde{Q}_r)\right)$$

Multiscale methods for dynamic fretting-wear

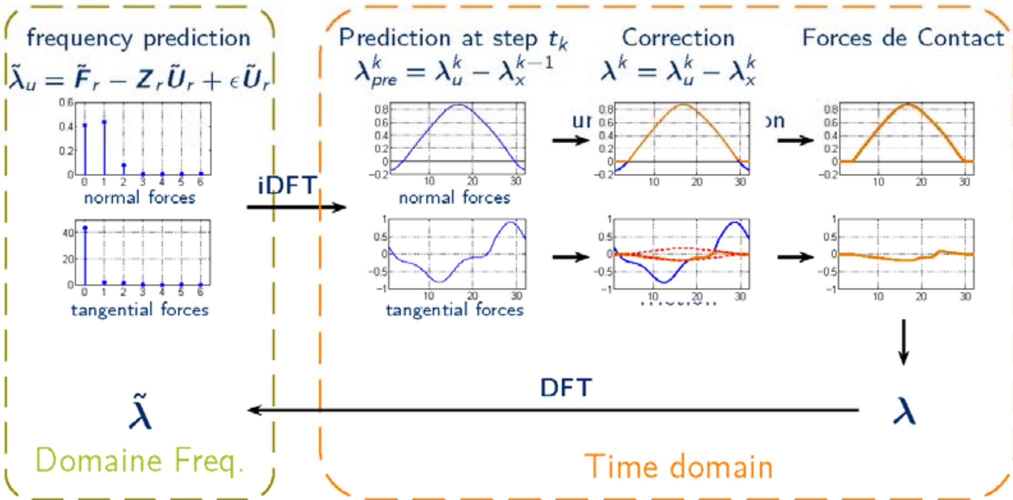
Separation in two time scales



- Hypothesis of periodicity on fast time scale
- Wear depths don't change on one fretting-cycle

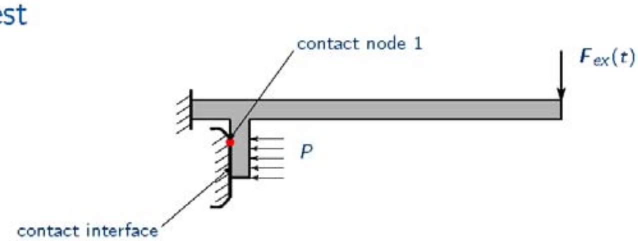
AFT procedure

Alternate frequency time procedure - calculation of contact forces

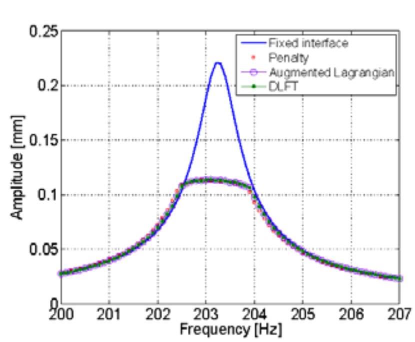


Comparison of 3 Contact Algorithms

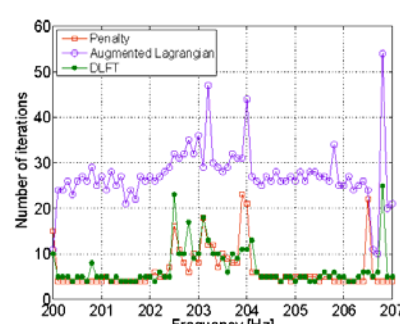
Model for test



FRF, first mode

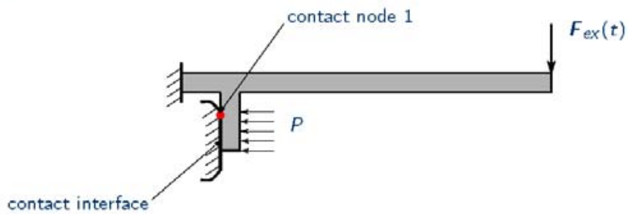


Number of iterations

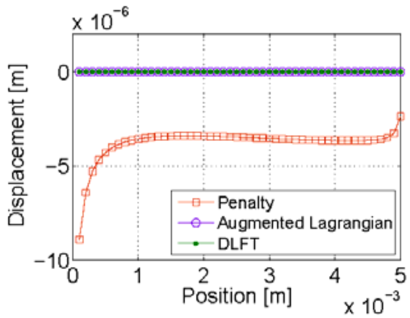


Comparison of 3 Contact Algorithms

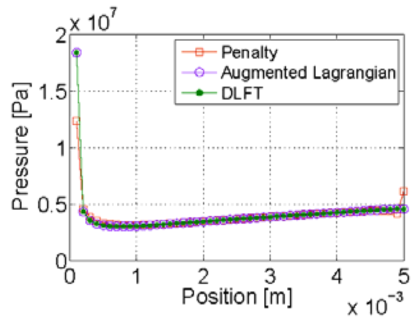
Model for test



Normal displacement

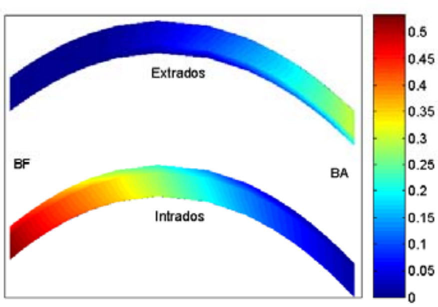


Contact pressure

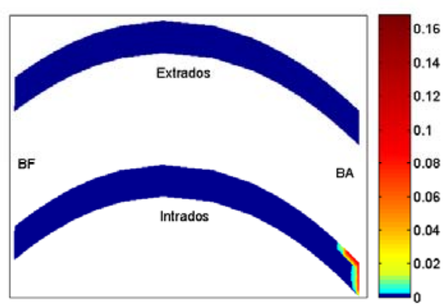


Fretting-wear of a dovetail root

Worn profile for LCF



Worn profile for mode 3F



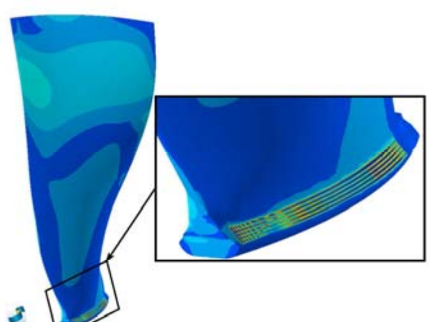
Comments

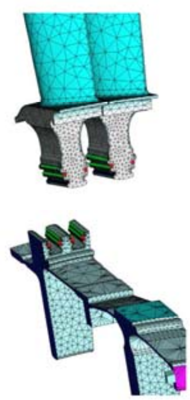
- Wear linked with take-offs and landings (quasi-static) is obtained in ABAQUS
- Worn area for mode 3F is different of worn areas obtained for LCF
- Worn area for mode 3F is much less important than for LCF

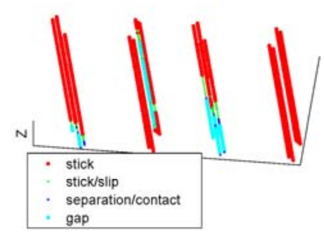
Challenge for the future

- Multiscale methods
- Multiphysics: fluid, smart material...
- High Performance Computing
- Reduced Order Modelling (Benchmark????)
- Microslip models

ROM: issue



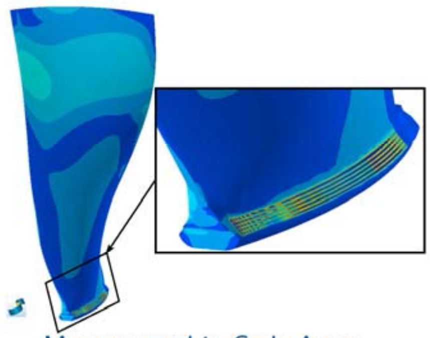


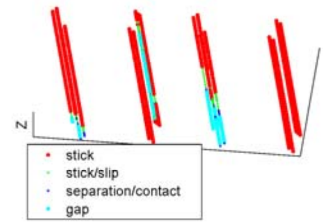


Memory used in Code-Aster

Elastic stiffness matrix, sparse matrix	130 MB
Nonlinear static simulation	$4.5GB$
Modal analysis with fixed interfaces	$5.1GB$
Calculation of static modes	$50GB$
Projection of stiffness and mass matrices	$24GB$
Matrices of ROM, full matrix	$250MB$

ROM: issue





Memory used in Code-Aster

Elastic stiffness matrix, sparse matrix	130 MB
Nonlinear static simulation	4.5*GB*
Modal analysis with fixed interfaces	5.1*GB*
Calculation of static modes	50*GB*
Projection of stiffness and mass matrices	24*GB*
Matrices of ROM, full matrix	250*MB*

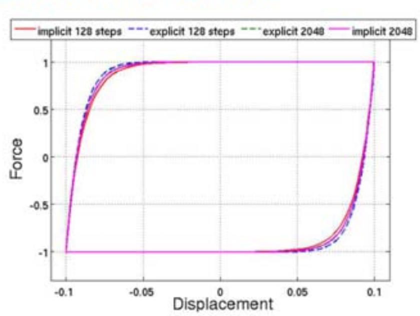
Microslip model

Valanis model

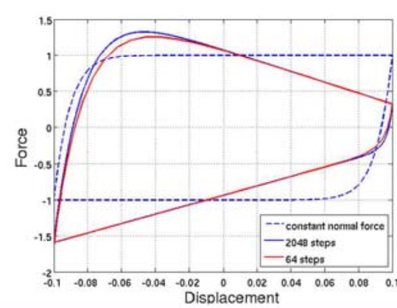
$$\dot{\sigma}_t = \frac{k_0 \dot{u}_t \left(1 + sgn(\dot{u}_t)\frac{\lambda}{k_0}(k_t u_t - \sigma_t)\right)}{1 + \kappa sgn(\dot{u}_t)\frac{\lambda}{k_0}(k_t u_t - \sigma_t)} \quad \text{with} \quad \lambda = \frac{k_0}{\sigma_0(1 - \kappa \frac{k_t}{k_0})}$$

Hysteresis loop

Constant normal load



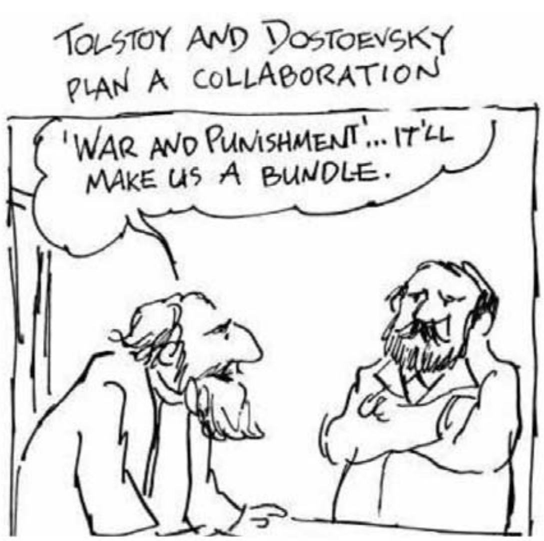
Variable normal load



Key factor

COLLABORATION

Key factor



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London

Explicit micro-slip modelling

T. Sanders, L.Salles, N. Hoffmann, C. Schwingshackl

Question

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- **How to Model microslip?**
  - **Implicitly with special model and dedicated input parameter**
  - **Explicitly with standard Coulomb friction elements**
- **Can I capture it with an explicit model?**

The challenge

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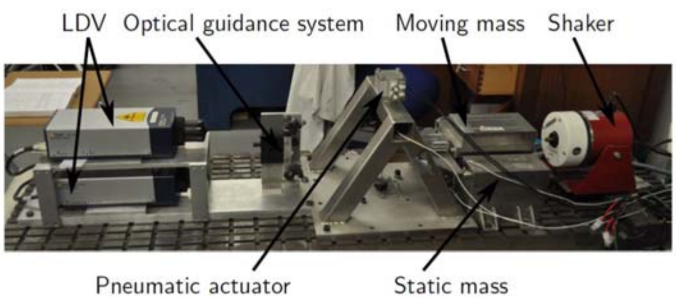

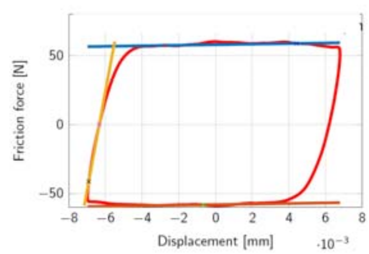

- **Predict the Hysteresis loop from 1D friction rig**
  - **Special focus is on the microslip zone**
- **Use measured input parameters for it**
  - **Friction coefficient, μ**
  - **Tangential contact stiffness, k_t**
- **Use FORSE for nonlinear analysis**
  - **MHB solver**

Linear FE models

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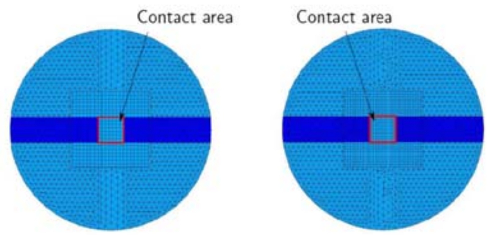
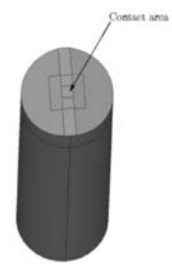


(a) 49 contact elements (b) 100 contact elements

- **Linear FE model of rig**
  - **Part of the arms included to capture boundary conditions**
  - **Rest of rig represented by stiffness**
- **Detailed mesh at contact interface**
  - **49-100-225-400 elements**
  - **Same number of nonlinear elements**



Liner analysis

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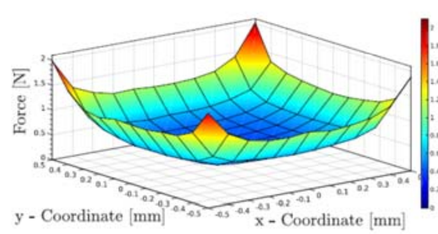
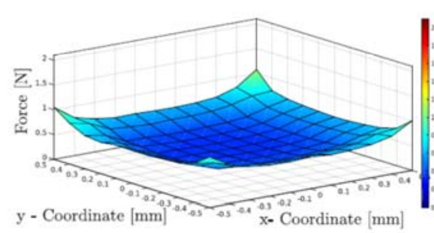
- **Static stress distribution at interface**
  - **Stress concentration at corners**
    - **Singularities**
  - **Modification to stress distribution to minimise effect**
- **Modal analysis**
  - **30 modes**



Normal load distribution of FE calculation



Adjusted normal load distribution

Initial simulation

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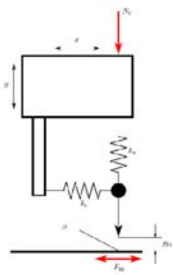
- Use of nonlinear MHB solver
  - 30 modes included
- Contact input values
  - μ=0.64
  - k_t=46.2kN/mm
- Parameter study
  - Harmonics
  - Normal load
  - Contact parameters
  - Measurement location



Adjusting the contact stiffness

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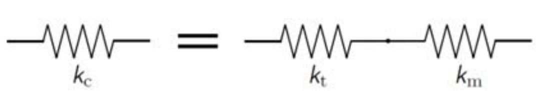
- Compliance in the specimen materials
  - Is being measured as well
  - Leading to k_c=46.2kN/mm
- Model includes material stiffness
  - Extract from simulation k_m=111kN/mm
  - Constant parameter
- Need modified contact stiffness k_t=79kN/mm



Influence of nonlinear model

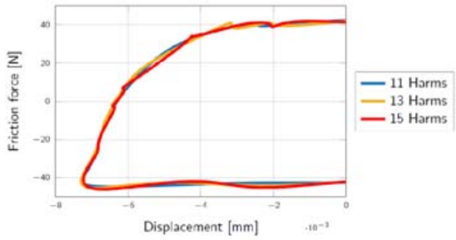


Abb. : Hysteresis loops for 100 contact elements with different numbers of harmonics

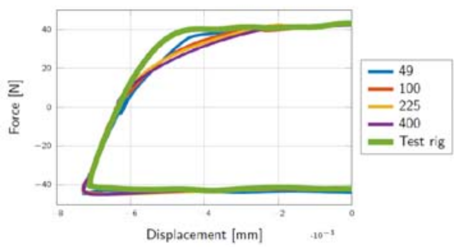


Abb. : Hysteresis loops - $N_0 = 65$ N

- **Microslip is occurring!**
- **Required number of harmonics?**
  - Approx. 15
- **Low element numbers**
  - Some microslip
  - Behaviour similar to bilinear spring
- **More elements**
  - Longer and smoother transition
- **Modified stiffness**
  - Good results

Influence of normal load

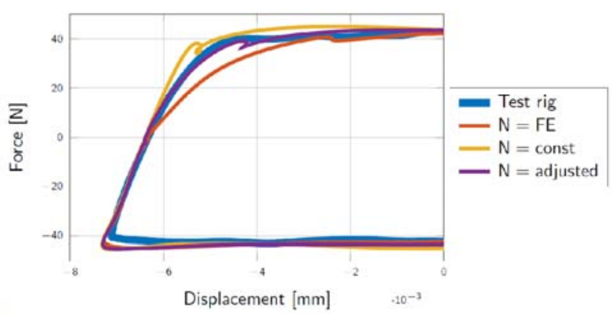
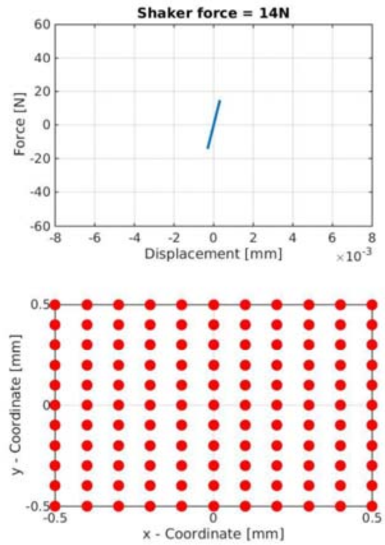


Abb. : Hysteresis loops for different normal load distributions - $N_0 = 65$ N

- **Constant pressure**
  - Slip occurs late
  - Pressure to high in centre
- **FE pressure**
  - Early slipping
  - Long transition due to high pressures in corner
- **Adjusted pressure**
  - Good agreement
- **By modifying the surface pressure**
  - Impact on the onset and duration of microslip

Micro slip transition

Imperial College London



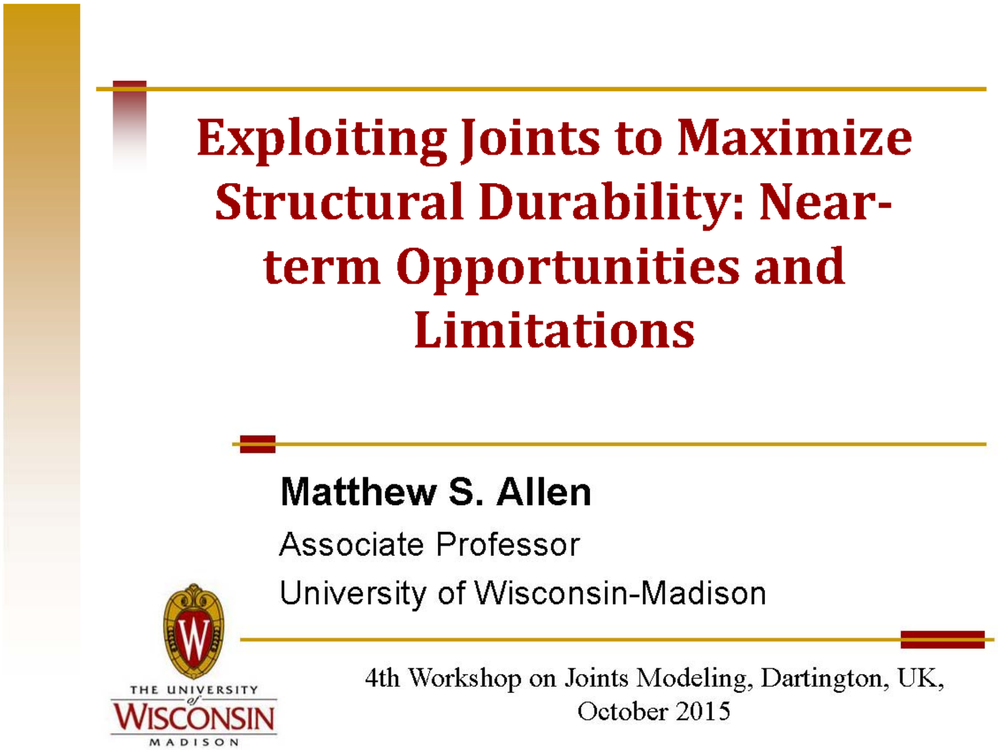
Summary

Imperial College London

- Explicit modelling is able to reproduce micro-slip
- A reasonably dense mesh is required to capture the behaviour
- It is mainly driven by the normal load distribution
- The measured contact stiffness must be adjusted to take compliance into account
- Slip develops from the interior, eventually moving outwards

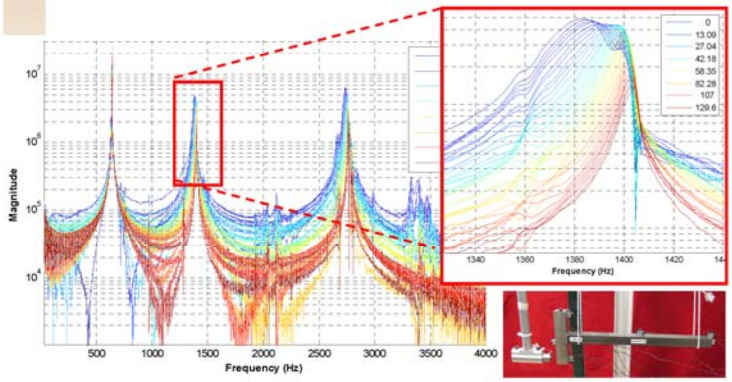
5.7. Session 4: Applications and Way Forward

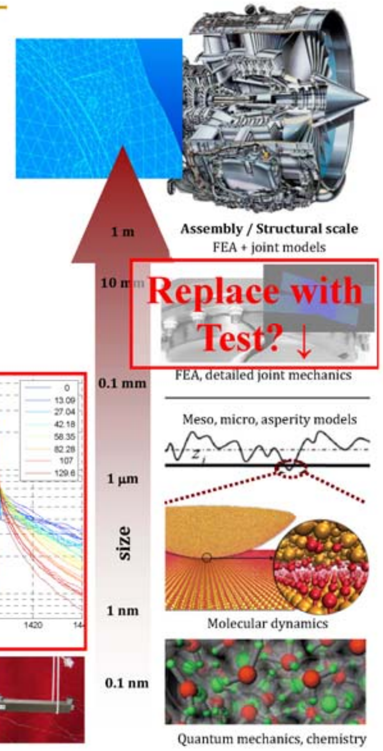
Matt Allen: Exploiting Joints to Maximize Structural Durability: Near-Term Opportunities and Limitations



Motivation

- Joints are nonlinear and influenced by physics at multiple length scales.
  - Goal: replace some lower scale models with test-based models.
- Resulting response is quasi-linear





Applications?

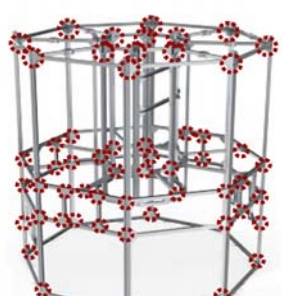
- The entire field of structural dynamics is significantly limited by our inability to predict the response of structures due to joints.
  - Test required $ increased cost, design challenges
  - Nonlinearity $ Difficulties in testing, FEM updating, response prediction in extreme environments, etc…
- We now have an opportunity to exploit joints in new (or existing) applications to maximize life/minimize weight.

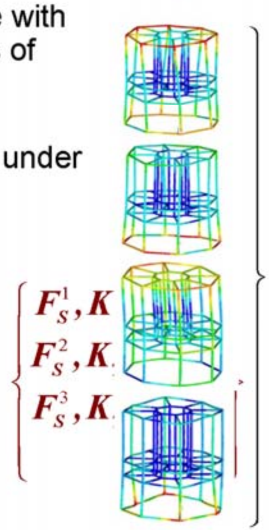
We may not yet know how to design joints to reduce vibration, but we have removed them and made it worse!

- **Examples:**
- F22 tail fin vibration problems
- Cowl in turbine engine
- **How many others?**

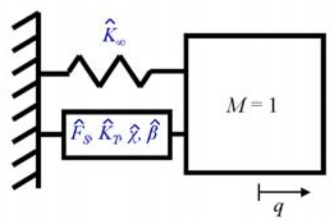
In many applications, uncoupled modal models can be used to simplify simulation, experiments, etc…

- Represent a structure with many modes in terms of uncoupled nonlinear oscillators.
- Theoretically justified under certain conditions [1]





Assumes that the linear modes are preserved, no coupling between modes!



$$\ddot{q}_r + \omega_{\infty,r}^2 q_r = \mathbf{\Phi}_r^{\mathrm{T}} \mathbf{F}^X + \hat{F}_r^J$$

[1] M. Eriten, M. Kurt, G. Luo, D. Michael McFarland, L. A. Bergman, and A. F. Vakakis, "Nonlinear system identification of frictional effects in a beam with a bolted joint connection," MSSP, vol. 39, pp. 245-264, 2013

Theoretical Foundation for Uncoupled Modal Modeling (Modal Iwan Modeling)

- Based on complexification and Averaging [1]: Consider the equation of motion for a structure with nonlinear joint forces $\mathbf{F}_J$.

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} + \mathbf{f}_J = \mathbf{f}_{ext}(t)$$

- If the motion of the structure is harmonic, the joint forces will be as well. Treat them as forcing terms:

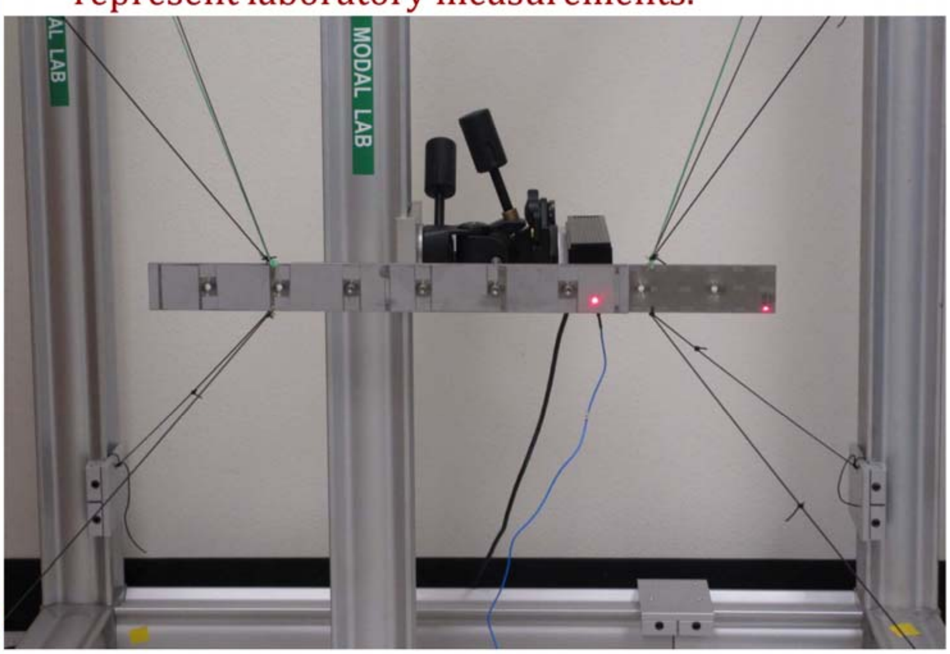
$$\mathbf{f}_J = \sum_{k=0}^{N} \mathrm{Re}\left(\mathbf{F}_{J,k} e^{ik\omega_1 t}\right)$$

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{f}_{ext}(t) - \sum_{k=0}^{N} \mathrm{Re}\left(\mathbf{F}_{J,k} e^{ik\omega_1 t}\right)$$

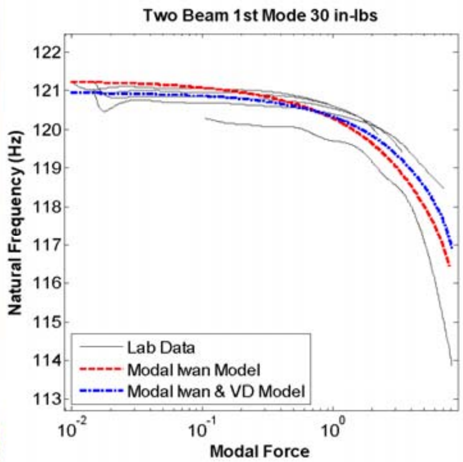
- Average over a vibration cycle $ the effect of the nonlinear forces is small (negligible) unless a significant forcing harmonic aligns with a natural frequency.
- **(Alternative: Undamped NNM based approach by Suess…)**
- Can readily be extended to cases where a few modes are coupled: [Festjens, Chevallier, Dion, JSV 2013]…

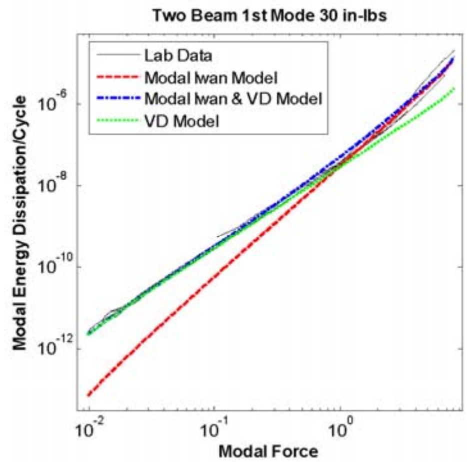
M. Eriten, M. Kurt, G. Luo, D. Michael McFarland, L. A. Bergman, and A. F. Vakakis, "Nonlinear system identification of frictional effects in a beam with a bolted joint connection," *MSSP*, vol. 39, pp. 245-264, 2013

A modal Iwan model was shown to accurately represent laboratory measurements.

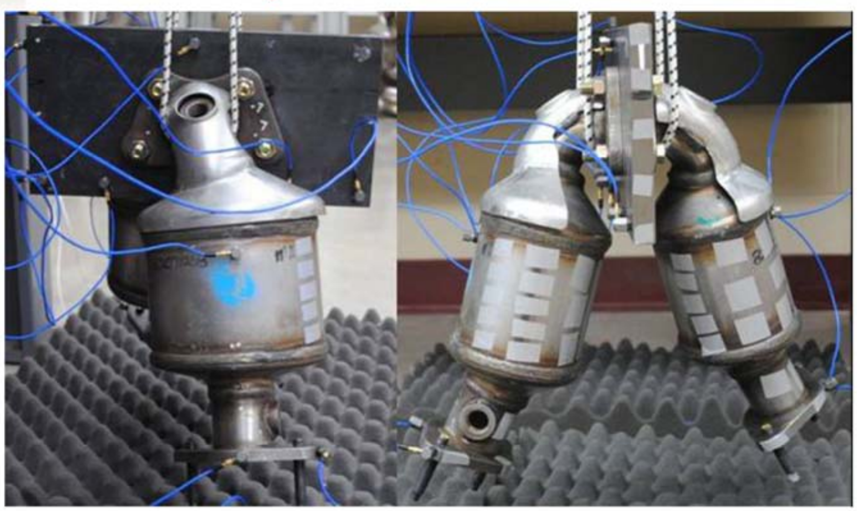


A modal Iwan model was shown to accurately represent laboratory measurements.



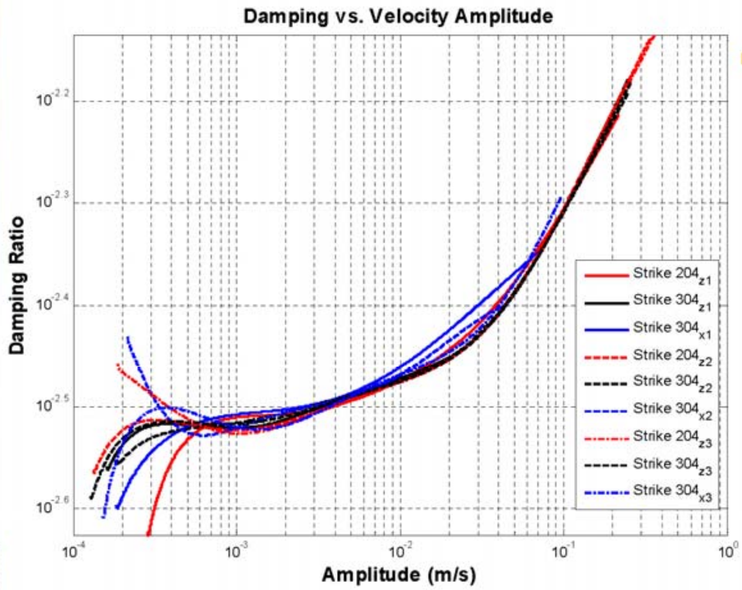


These concepts also hold for realistic hardware



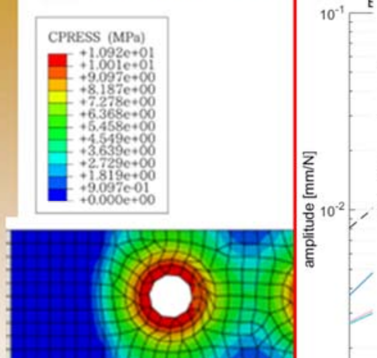
- Front and rear catalytic converters assembled together with required assembly torque and exhaust manifold gasket.
- System hung freely suspended by bungee cords to complete a roving hammer test.

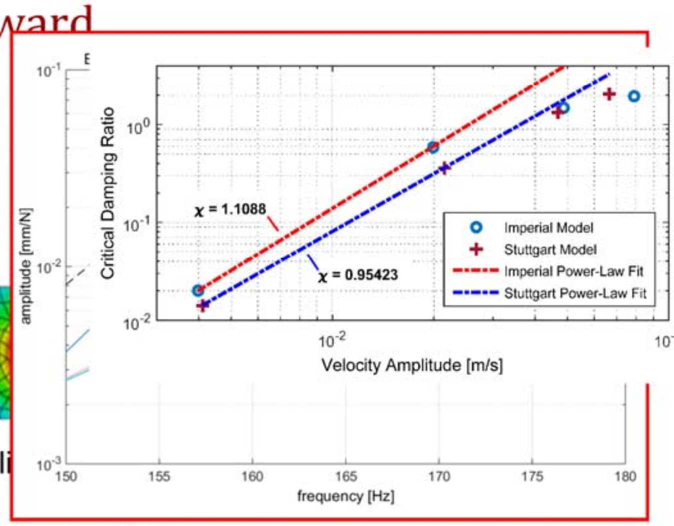
Modal Iwan model accurately captures the damping versus amplitude for various input points (various combinations of the different modal amplitudes)!



- Because the quasi-modal framework is valid, we can reduce the number of tests and test/model comparisons dramatically!

Path Forward

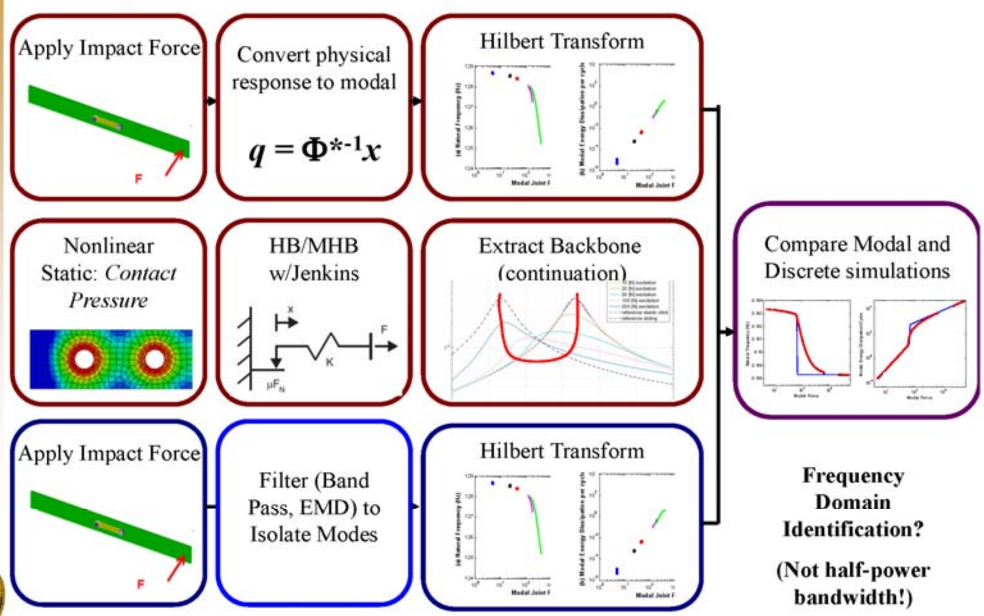




- Current modeli… themes!
  - Common performance?
  - Promising outlook for high fidelity modeling and prediction.
- To attack new applications (and some existing ones) we need methods that are **dramatically faster** and **accurate enough**.

Path Forward:

Improve identification procedures, experimental & analytical



Other near term objectives:

- Efficient tools already exist to simulate the uncoupled modal response.
  - Uncoupled nonlinear oscillators!

$$\ddot{q}_r + \omega_{\infty,r}^2 q_r = \mathbf{\Phi}_r^{\mathrm{T}} \mathbf{F}^X + \hat{F}_r^J$$

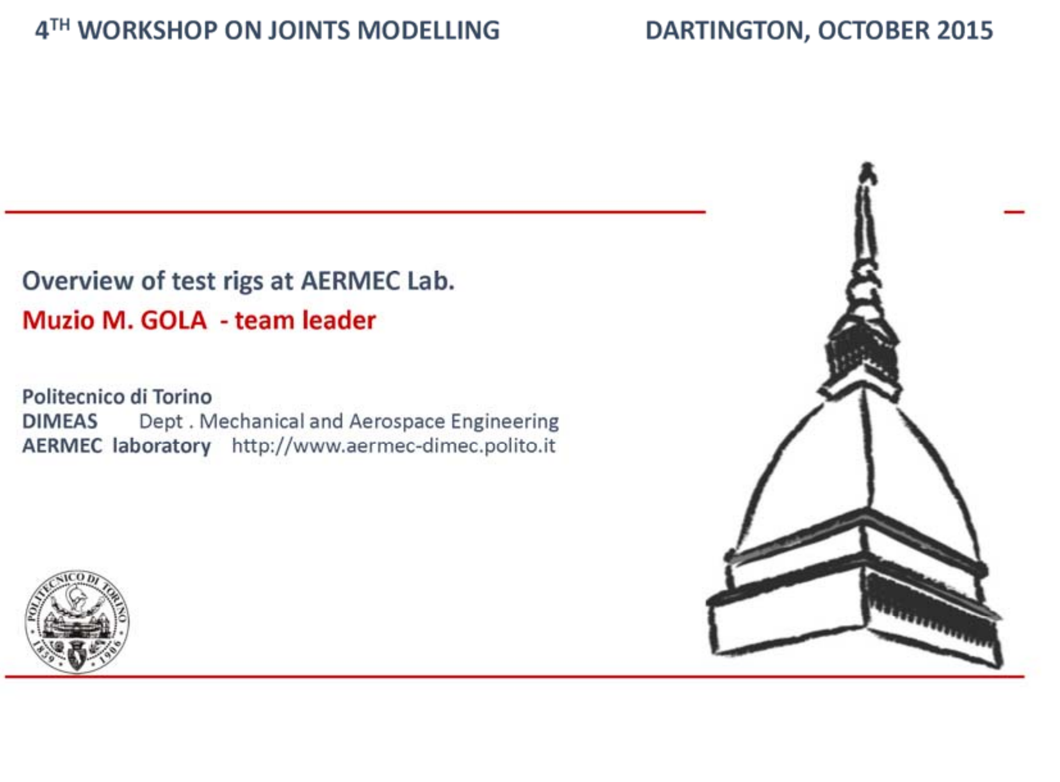
- Continue to improve prediction methods:
  - Friction laws, interface mechanics, etc
  - Simplified by recognizing that the resp linear – **Amplitude dependent DAMI STIFFNESS, SLIP FORCE**
- Continue to pursue fully coupled, nor simulations for macro-slip, shock, etc

Far From a Conclusion…

- **Many important challenges remain to be solved… but**
- We are nearing a crossroads where many more industrial applications will soon become feasible.

"The best symphony has not yet been written"

Muzio Gola: Overview of Test Rigs at AERMEC Laboratory



The AERMEC team

AERoMEChanics of turbomachinery

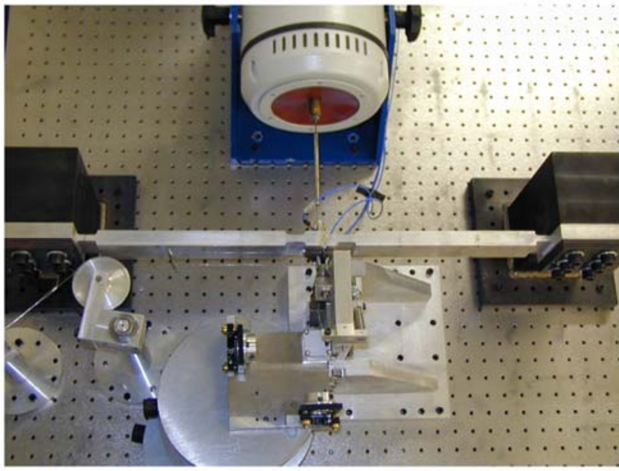
www.aermec-dimec.polito.it

Index

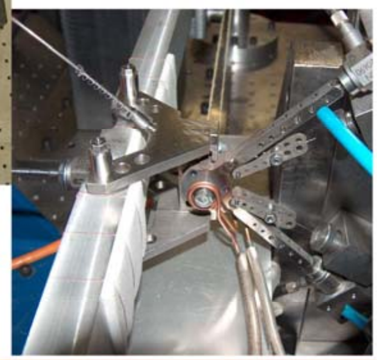
The test rigs designed by - and in use at - the AERMEC lab. :

1 - The first friction & wear test rig for "SPHERE" contact surfaces ~ 2002
2 - The second friction & wear test rig: "FLAT" contact surfaces ~ 2010
3 - The piezo test rig for dampers - version 1 ~ 2008
4 - The piezo test rig for dampers - version 2 ~ 2013
5 - The resonant test rig for dampers on blades ~ 2014
6 - Test rigs for a vane segment with interlocking - 1997, 2002
7 - Test rig with two blades and one damper - 2006
8 - Test rig for damping of blade root joints ~ 2004
9 - "OCTOPUS" test rig: bladed disk with underplatform dampers ~ 2010
10 - Spinning test rig ~ 2007
11 - Tip timing measurement system for the spinning rig ~ 2014
12 - Fatigue of materials for turbo-engine blades ~ 2010
13 - Fatigue at of dovetail type attachment ~ 2015

1 - The first friction & wear test rig for "SPHERE" contact surfaces ~ 2002



Max temperature: 1000°C
Temperature control: ± 1°C
Displacement: 2-200 μm
Displ. control: ± 0.5 μm
Normal force: max 250 N



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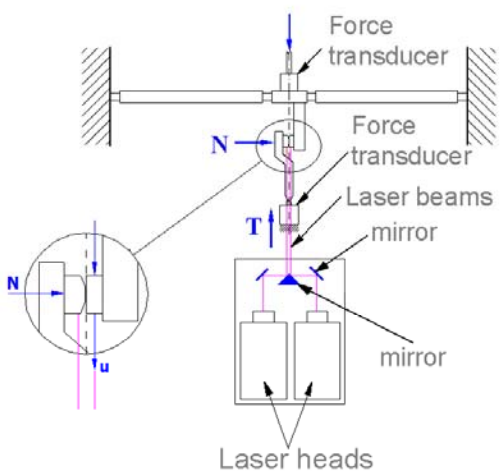
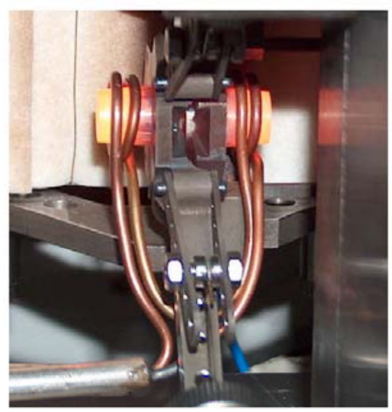
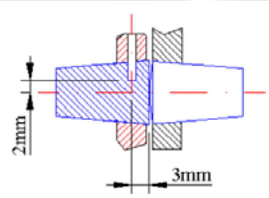
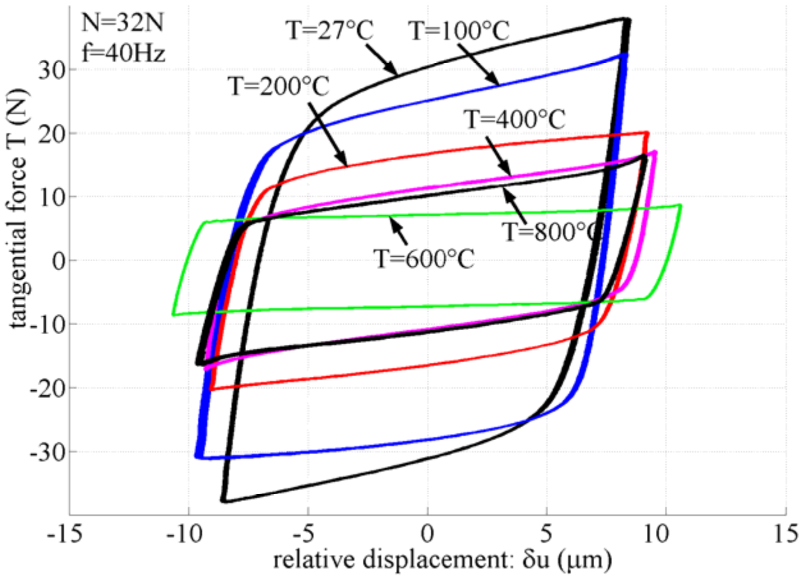


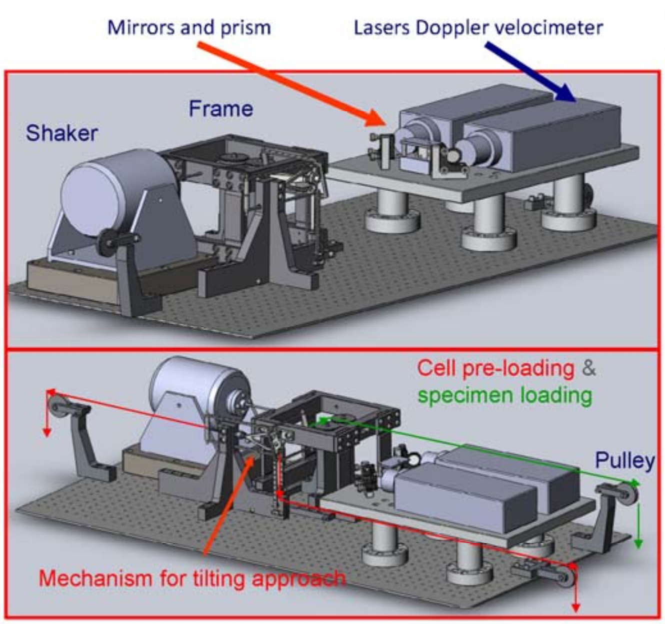




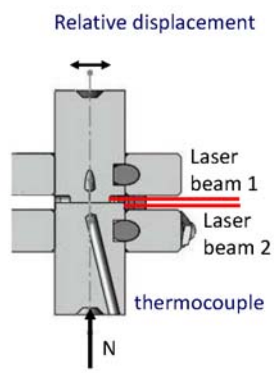
1 - The first friction & wear test rig for "SPHERE" contact surfaces ~ 2002



2 - The second friction & wear test rig: "FLAT" contact surfaces ~ 2010



Max temperature: 1000°C
Temperature control: ± 1°C
Displacement: 2-80 μm
Displ. control: ± 0.5 μm
Normal force: max 300 N



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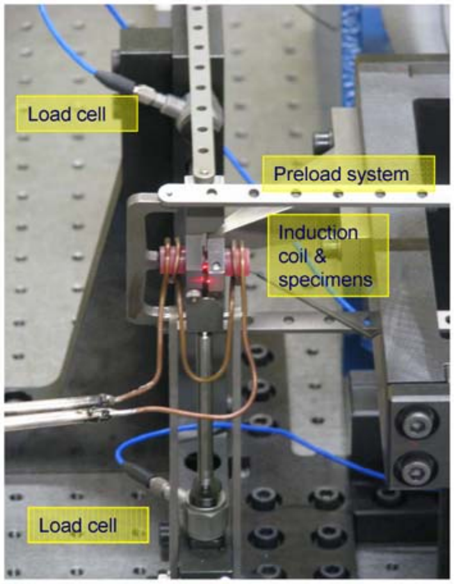

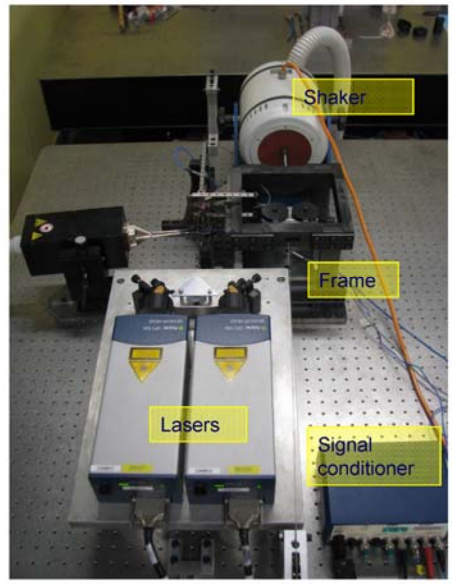

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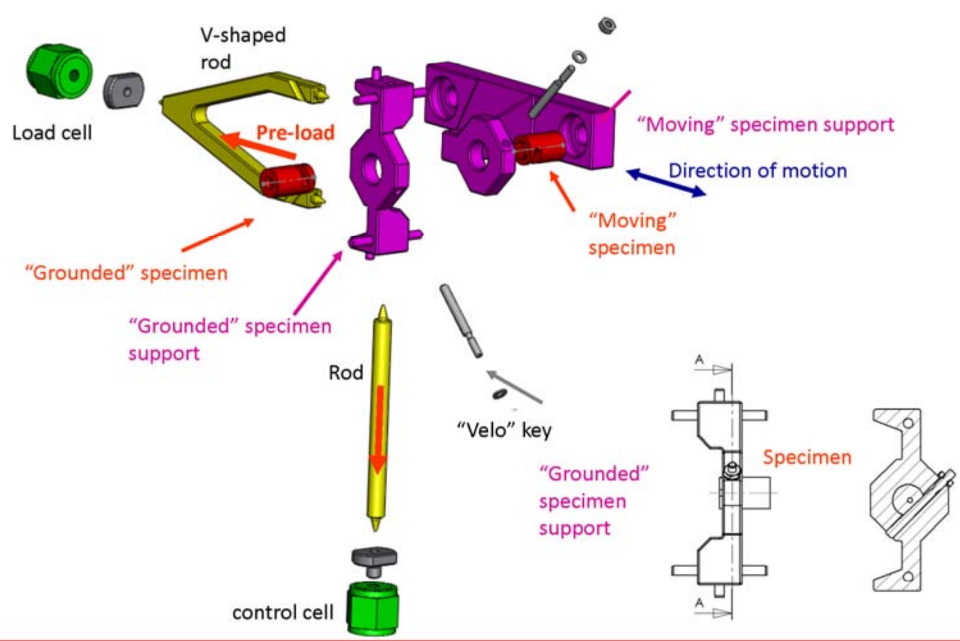
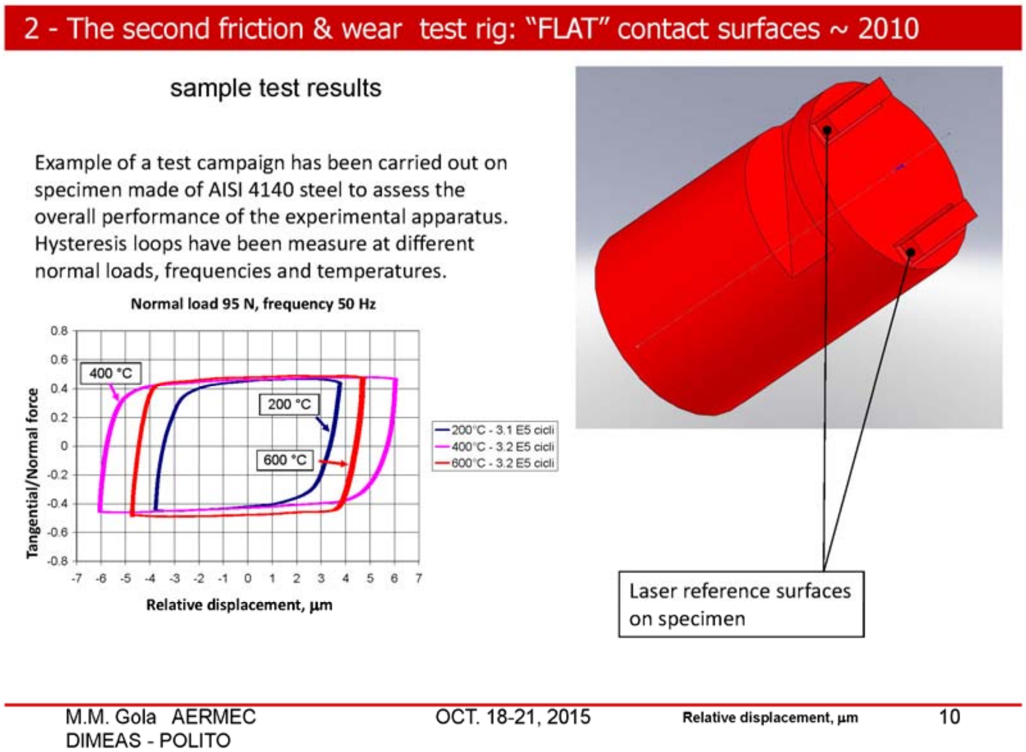
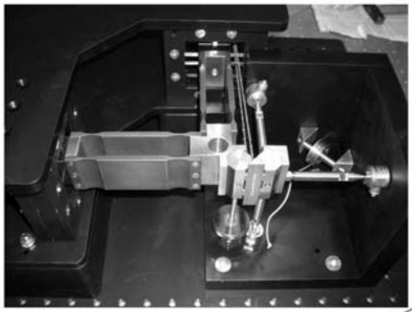




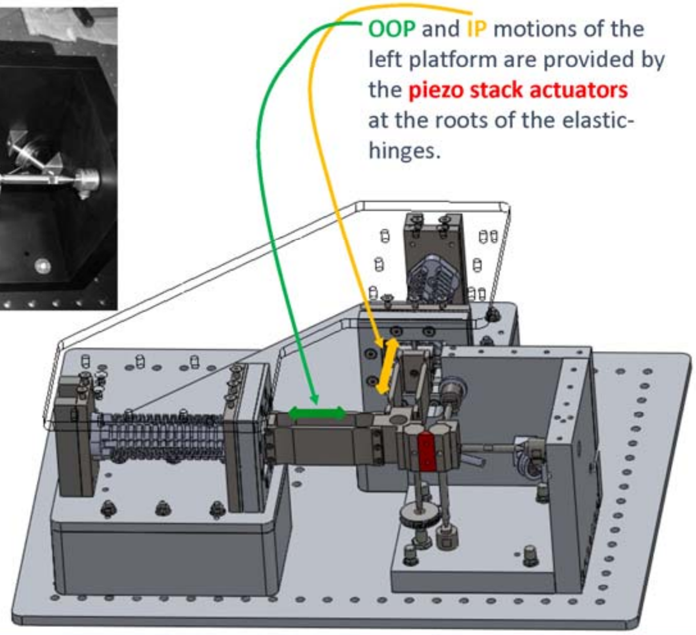
1. So far we concentrated on the measurement of contact parameters such as friction coefficient and tangential contact stiffness, that are the crucial parameters to simulate dynamic systems with joints. Little has been done to measure the normal contact stiffness that also play an important role, even if for a second order approximation, in the contact simulation of real systems. How could we measure it? We have to think of special rigs and sensors as the relative displacements along the normal to the contact plane are very small.

2. We are able to measure the contact kinematics, namely the relative displacements, at few points, usually located near the contact area but, and the reason seems obvious, on the external of the contact area. Can we infer information about the kinematic behavior of contact pairs lying within the contact area from this limited set of measurements? Are our contact models reliable enough to predict the stick/slip zones inside the contact area? This is a critical issue especially for conformal contact where the contact area extends on a broad region.

3 - The piezo test rig for dampers - version 1 ~ 2008



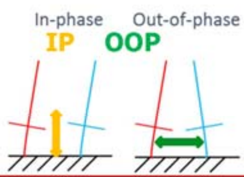
OOP and **IP** motions of the left platform are provided by the **piezo stack actuators** at the roots of the elastic-hinges.



Test rig version 1

Displ: 5÷75 μm

Pract. limit ≈ 80 Hz



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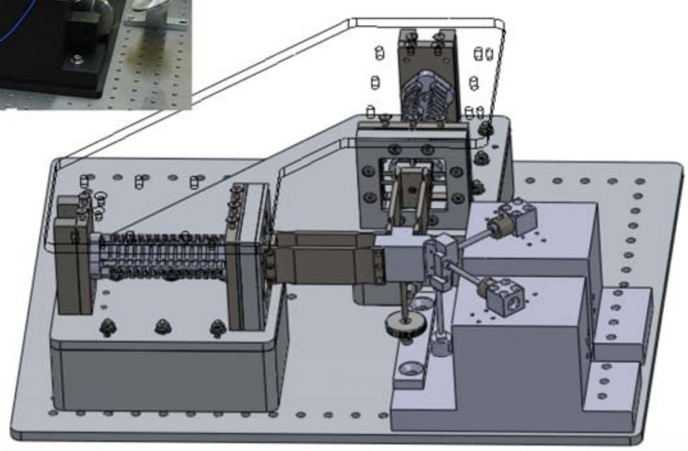
4 - The piezo test rig for dampers - version 2 ~ 2013



Test rig version 2

Displ: 5÷75 μm

Pract. limit ≈ 150 Hz

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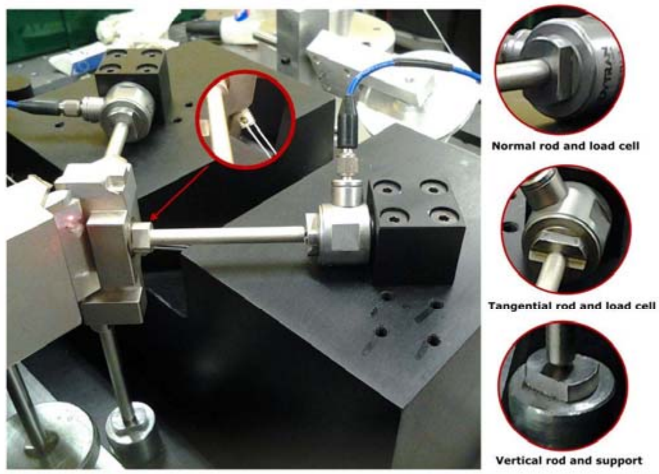
4 - The piezo test rig for dampers - version 2 ~ 2013

The test rig:
- measures the forces transferred between the two simulated **platforms** of neighbouring blades through the **under-platform damper**
- measures the motion of left platform and damper vs. the right platform.

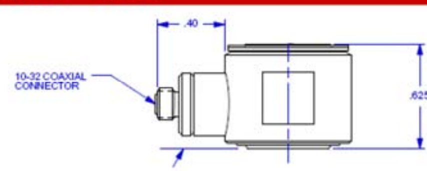
The damper is placed between the platforms and loaded by dead weights which reproduce the effects of the centrifugal force **Fc**.

Quasi statically determinate "tripod"



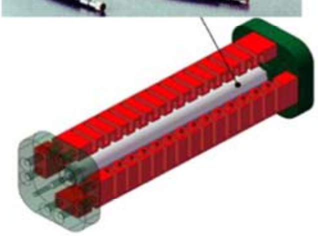
4 - The piezo test rig for dampers - version 2 ~ 2013

The two load cells: Dytran 1051V2, calibration factor 44.5 N/V (100 mV/lbf), force range ±50 lbf ; stiffness ≅ 2000 N/μm ; res. ≅ 0.003 N

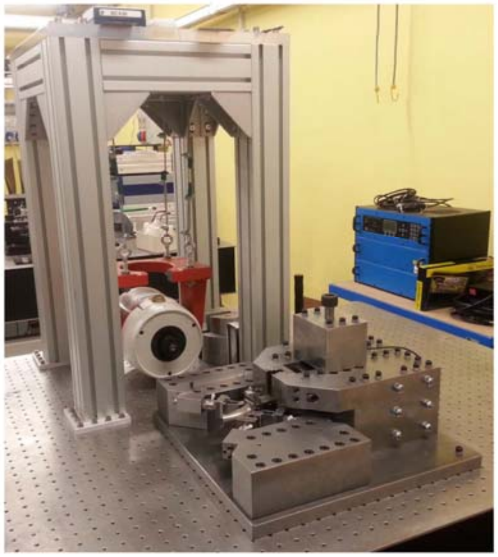


Displacements and the rotation of the damper during the tests measured by means of the differential laser doppler vibrometer Polytec - sensor heads OFV-512, vibrometer controller OFV-3001, resolution 20 nm.

Piezo-electric stacks: PI-Ceramic PICA™ P-010.80P. Each piezo stack is protected against traction and bending by means pre-loading four axial springs in parallel, based on Ω-bending: 4.28 ÷ 4.43 N/mm, ASTM 52SiCrNi steel



5 - The resonant test rig for dampers on blades ~ 2014



Measurements of forces and displacements on a couple of dampers acting on a single blade.

Four piezo-type load cells, two for each damper, for measuring both dynamic and quasistatic forces. Contact forces on damper are measured in tangential and normal direction. Overall output force resolution (load cell+ amplifier) 0.3N

Displacements are measured with two laser vibrometers. Overall output displacement resolution 0.1 μm.

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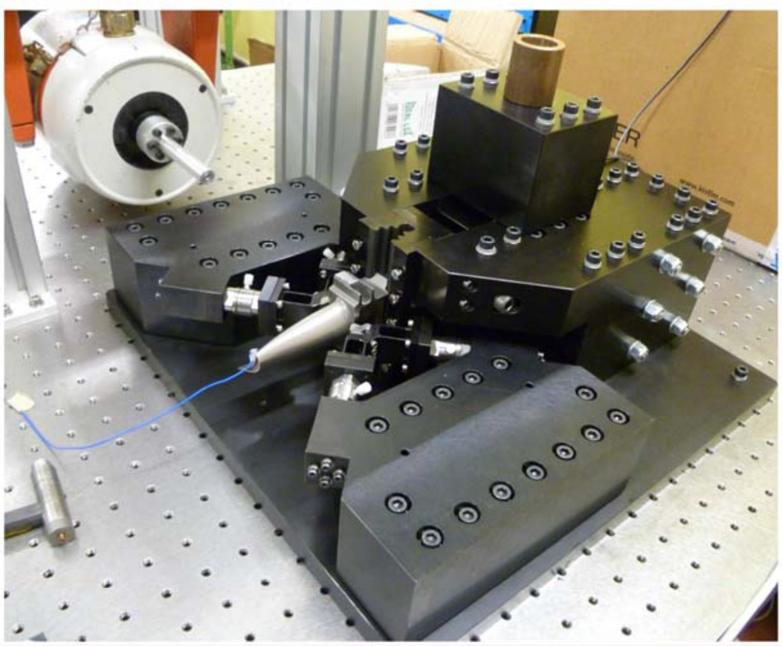
5 - The resonant test rig for dampers on blades ~ 2014



The rig has been designed for room temperature only.

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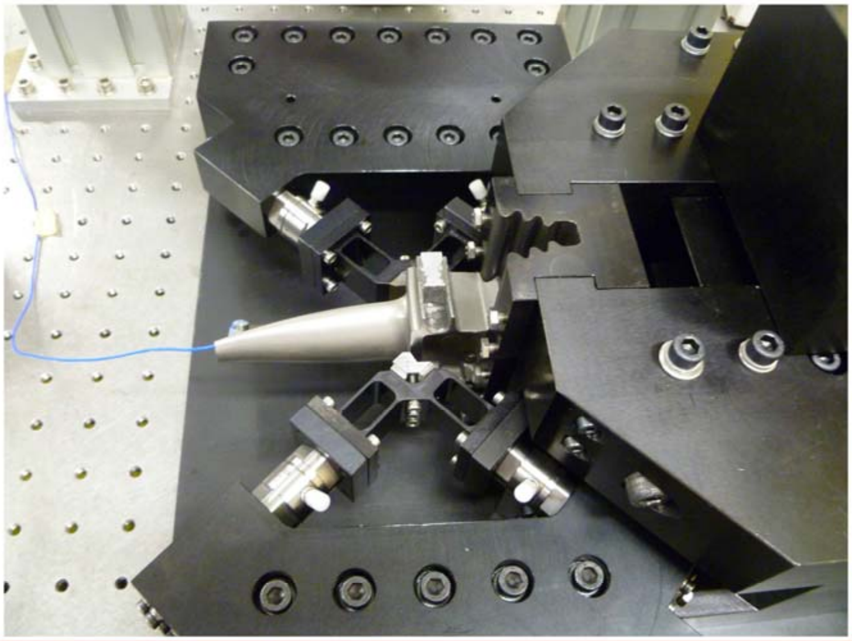
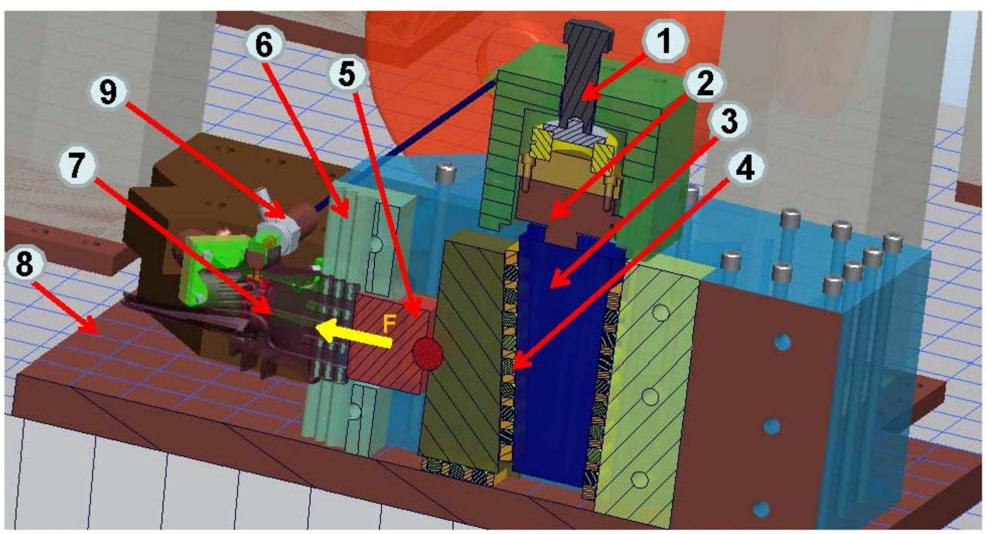
5 - The resonant test rig for dampers on blades ~ 2014



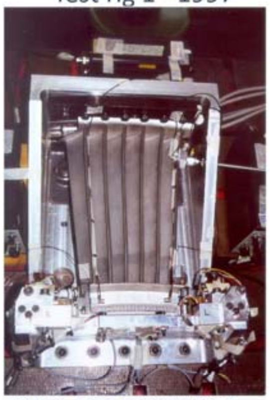
5 - The resonant test rig for dampers on blades ~ 2014



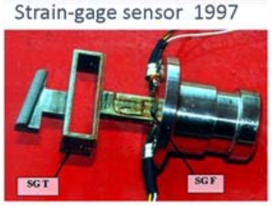
1: Bolt 2: Strain Gauge 3: Wedge Block 4: Linear Flat roller Bearing 5: Adjusting Pin
6: Blade Adapter 7: Turbine Blade 8: Base Plate 9: Stinger with strain gauge

6 - Test rigs for a vane segment with interlocking - 1997, 2002 on

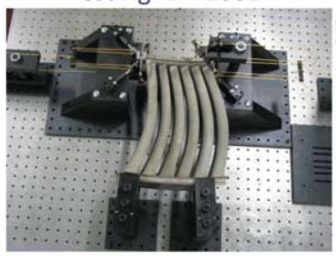
Test rig 1 - 1997

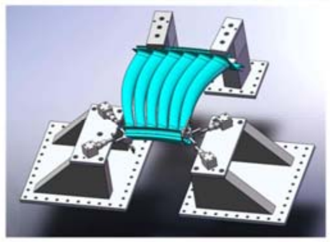


Strain-gage sensor 1997



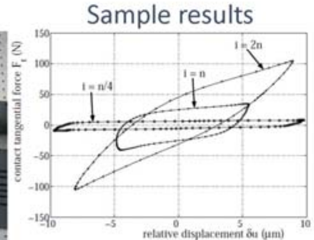
Test rig 2 – 2002

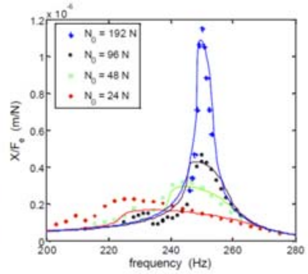




Piezo sensor 2002

Sample results





7 - Test rig with two blades and one damper - 2006 on

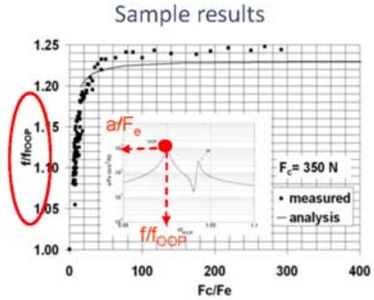
Test rig

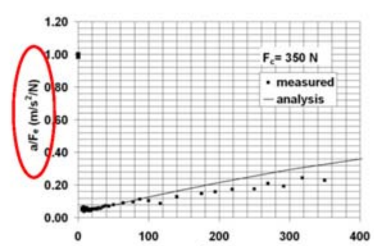
Real blades

Dummy blades

Dampers tested

Sample results



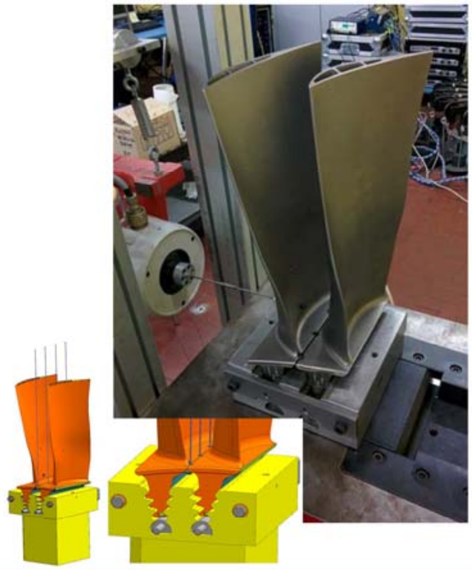


7 - Test rig with two blades and one damper - 2006 on

Steam turbine blades

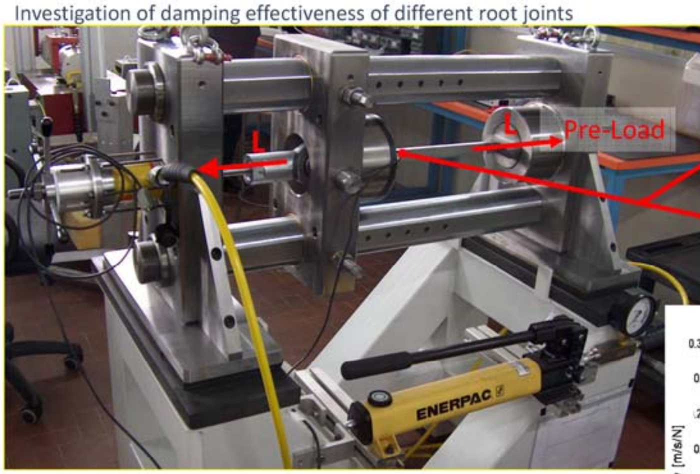
sample damper shapes

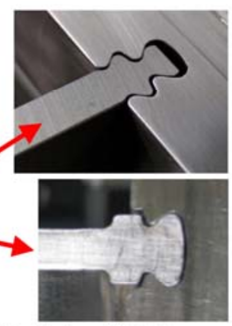
Gas turbine blades

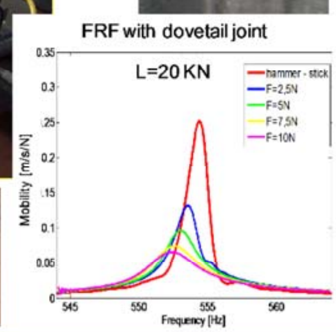


8 - Test rig for damping of blade root joints ~ 2004

Investigation of damping effectiveness of different root joints







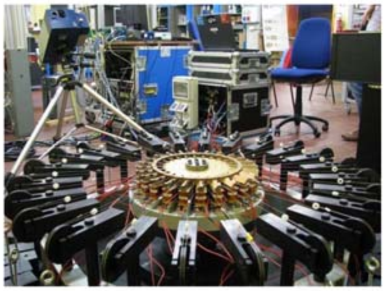
Non- contact (electromagnetic) excitation force

Non- contact (laser) measurement

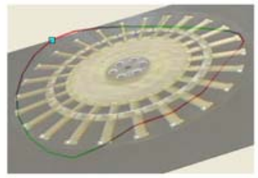
9 - "OCTOPUS" test rig: bladed disk with underplatform dampers ~ 2010

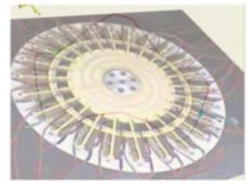
Main Features: underplatform dampers, non-contact (magnetic) rotating excitation force.

Test rig



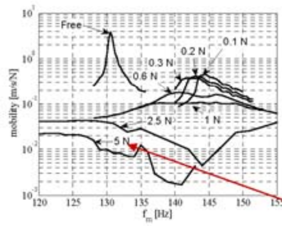
Sample results

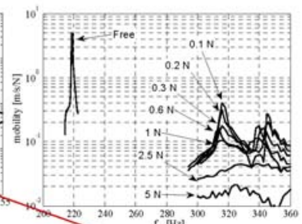




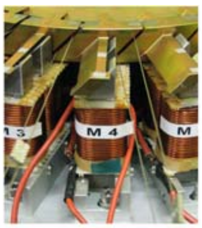
ND 2

ND 4





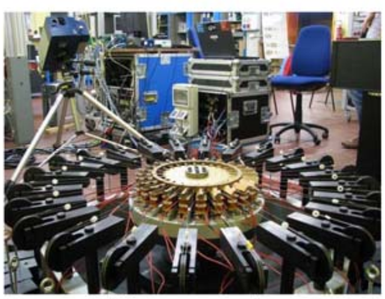
Excitation force



Electromagnets
Force amplitude
5 N @ 600 Hz,
15 N @ 300 Hz

9 - "OCTOPUS" test rig: bladed disk with underplatform dampers ~ 2010

Test rig

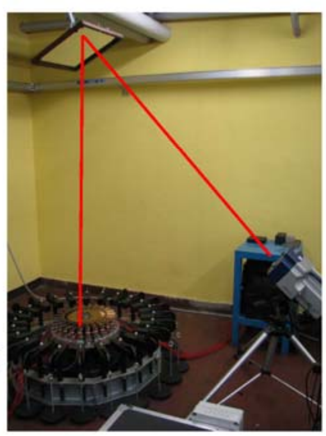


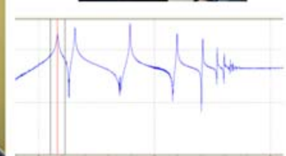
EMs power supply

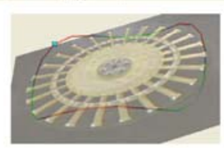
Signal generator
and controller NI cRIO

12 amplifiers
2 channel (800 W)

Laser scanning vibrometer



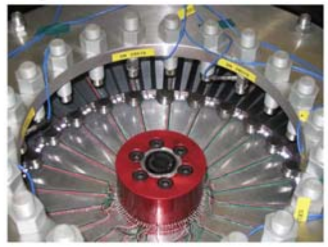




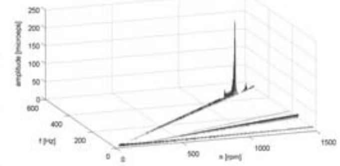
10 - Spinning test rig ~ 2007

Vertical axis spinning rig

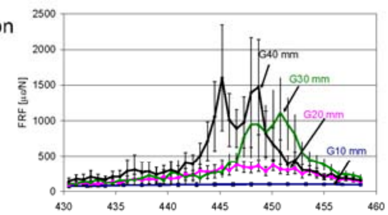
Excitation by means of permanent magnets



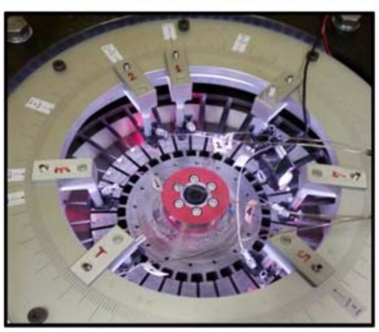
Results example
3D Campbell diagram



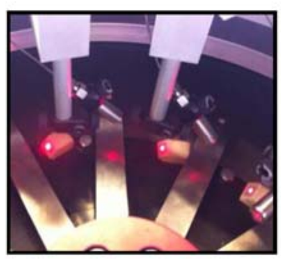
- Max rotation speed 4000 rpm
- Max disk outer diameter 630 mm
- Rotation in void
- Telemetry system 24 channels
- From 1 to 24 magnets for excitation
- 6 magnets with force trasducers (max amplitude 10 N)



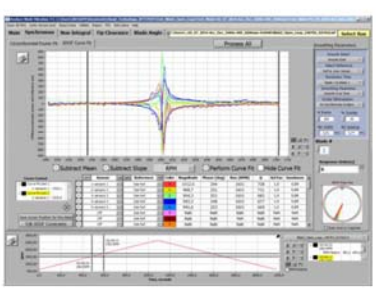
11 - Tip timing measurement system for the spinning rig ~ 2014



Hood Tip timing System: new acquisition (2014)



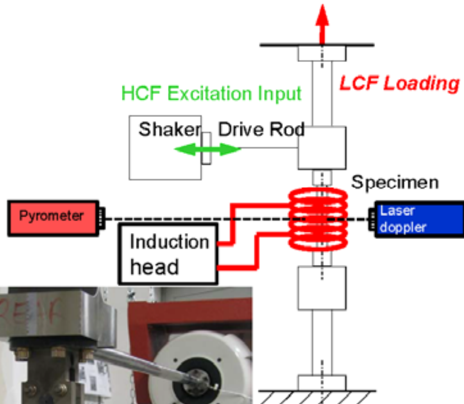
8 Optical probes



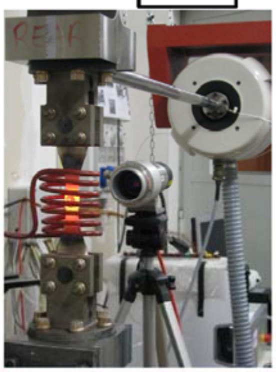
Data post processing from measurement on each blade:
Amplitude, frequency Q factor

Expected results: experimental data to validate numerical codes

12 - Fatigue of materials for turbo-engine blades ~ 2010



Single crystal nickel based super alloys fatigue testing machine
LCF max load 150 kN,
Test frequency ~ 1800 Hz
Displacement ~ 1 mm (at the center of specimen)
Test temperature 1100°C



M.M. Gola AERMEC
DIMEAS - POLITO
OCT. 18-21, 2015
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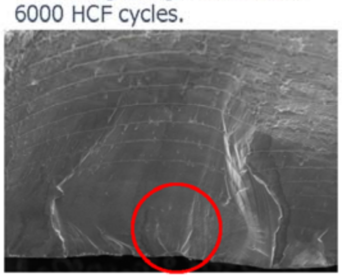
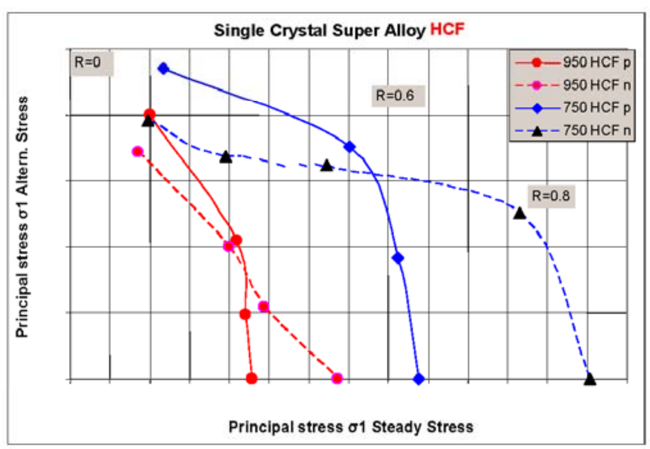
12 - Fatigue of materials for turbo-engine blades ~ 2010

Fracture surface of CCF. Arrest lines every LCF cycles, containing the growth due to 6000 HCF cycles.



HCF tests: comparison **p**lain vs **n**otched



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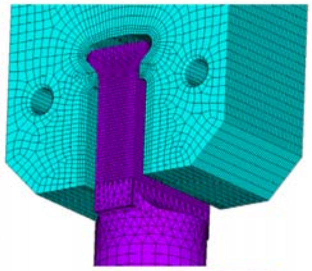
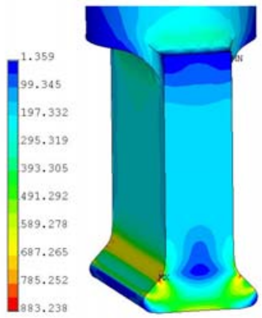
13 - Fatigue at of dovetail type attachment ~ 2015

Specimens and dummy disk

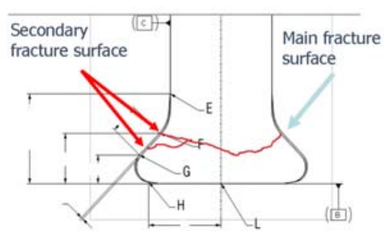


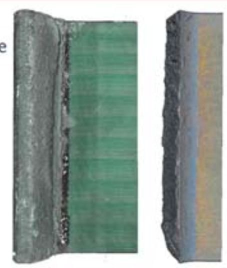


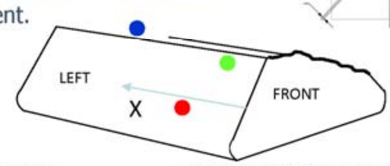
Normal load 20 kN
Test frequency ~ 0.5 Hz
Test temperature 650°C

13 - Fatigue at of dovetail type attachment ~ 2015

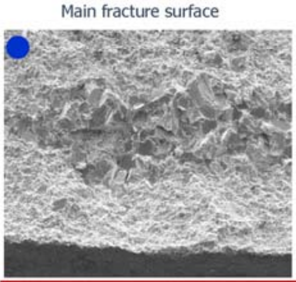
Cracks initiate at the end of contact area. On the main fracture surface, where the crack initiates and propagates till failure, a single fracture surface can be identified. On the secondary fracture surface two fracture surfaces close to the stress peaks are evident.







Main fracture surface



Fretting cracks inside the contact area

Fretting crack near the contact end

Structural Dynamics from a Complex Systems Perspective: Are we missing something out here?

Workshop on Mechanical Joints
Dartington, October 2015

Norbert Hoffmann
Imperial College London
Hamburg University of Technology

Where to start from? Some Previous Work.

Dynamics and Friction Interfaces

- Friction Self-Excitation
- Friction Models

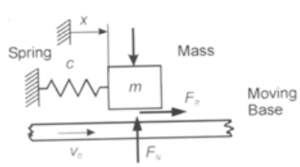
System Dynamics and Vibration Response

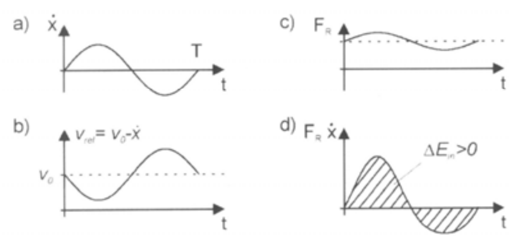
- Standard and Non-Standard Bifurcations
- Sliding in Detail and Break Away
- Random vs. Deterministic Dynamics

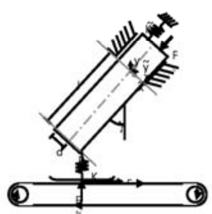
Modelling Intensity vs. Data Intensity:

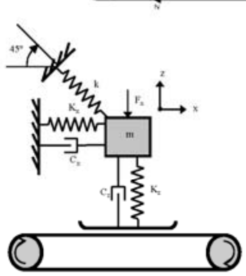
Where are we, where do we move, where should we go?

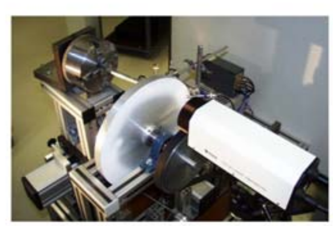
Friction Self-Excitation





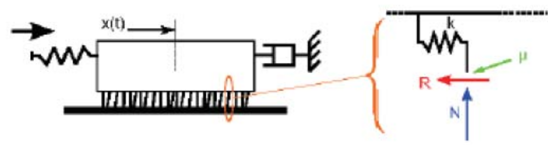






Friction Models

$$F = \frac{N}{N_0}\left(\sigma_0 z + \sigma_1 \dot{z} + \sigma_2 v_r\right)$$
$$\dot{z} = v_r - \frac{\sigma_0 \left|v_r\right|}{g\left(v_r\right)} z$$

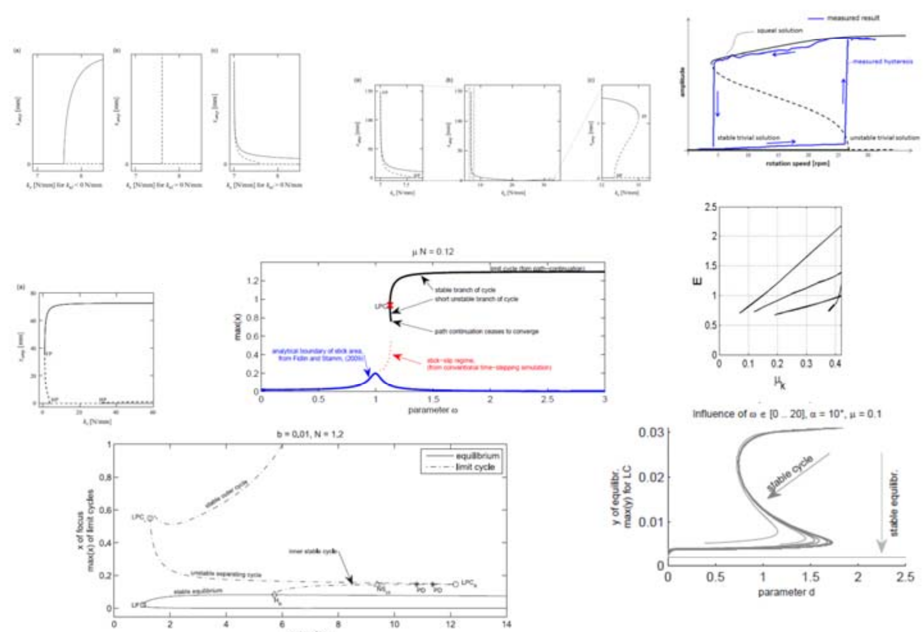


$$\underbrace{\sigma_{xy}(x,t)}_{\substack{\text{frictional}\\ \text{stress}}} = \underbrace{\bar{\sigma}(x,t)}_{\substack{\text{asperity}\\ \text{dynamics}\\ \text{stress}}} + \underbrace{\eta \dot{u}_x(x,t)}_{\substack{\text{velociy}\\ \text{strengthening}\\ \text{stress}}},$$

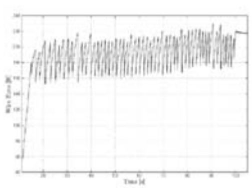
$$\dot{\bar{\sigma}}(x,t) = \underbrace{\mu_0 A \dot{u}_x(x,t)\,/h}_{\substack{\text{elastic interface}\\ \text{deformation}}} - \underbrace{\underbrace{\Theta\left(\sigma_{xy}(x,t)\,/A - \sigma_c\right)}_{\text{stress threshold}}\ \underbrace{\bar{\sigma}\dot{u}_x\,/D}_{\text{decrease rate}}}_{\text{asperity rupture}},$$

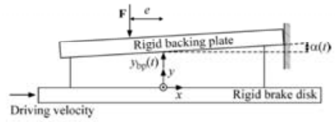
$$\dot{A}(x,t) = \underbrace{\left(A(x,t) - A_0\right)/\tau_0}_{\substack{\text{increasing}\\ \text{contact area}}} - \underbrace{\underbrace{\Theta\left(\sigma_{xy}(x,t)/A - \sigma_c\right)}_{\text{stress threshold}}\ \underbrace{\kappa A u_x\,/D}_{\text{decrease rate}}}_{\text{decreasing contact area}}.$$

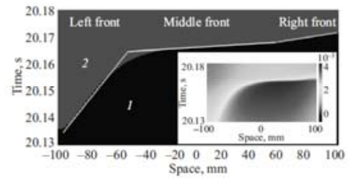
Standard and Non-Standard Bifurcations

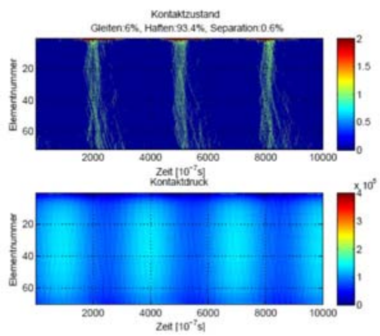


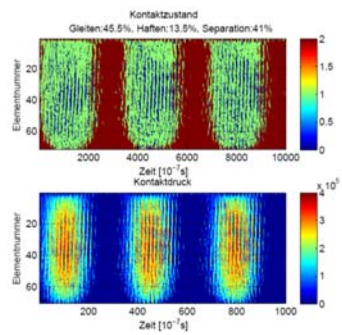
Details During Sliding or Break Away



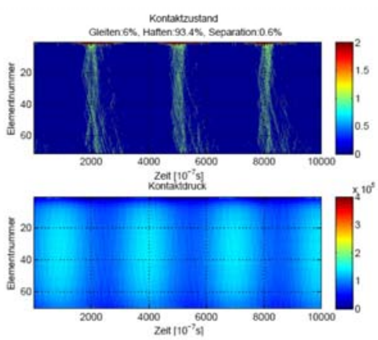


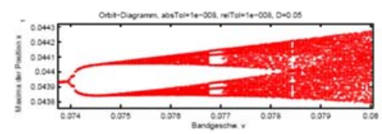


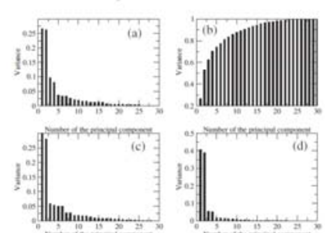


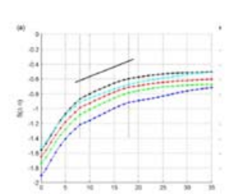


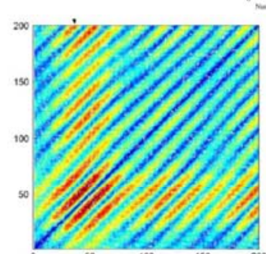
Deterministic Dynamics vs. Random Dynamics

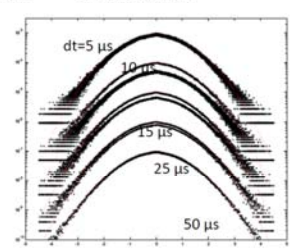




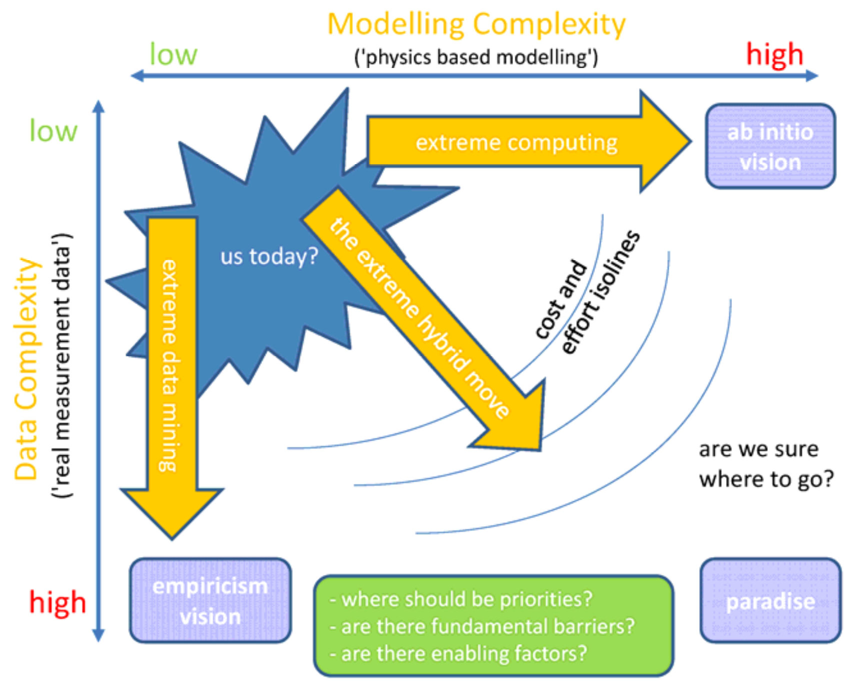








An Intensity Chart: Data vs. Modelling



David Nowell: Challenges in the Measurement and Modelling of Frictional Contact

Where do we need to make progress?

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- A 'top down' model is unlikely to be fully predictive
  - Phenomenologically-based
  - Will correlate only over a limited parameter space
  - May be computationally efficient
- A 'physics-based' model should be fully predictive
  - Based on understanding the joint behaviour in detail
    - Is this simply phenomenological at a lower length scale?
  - Should predict over a wide parameter space
  - Likely to be complex and difficult to use directly at the structural level

- Example – contact stiffness
  - What physics do we need to capture?

2

Behaviour of Smooth & Rough Contacts

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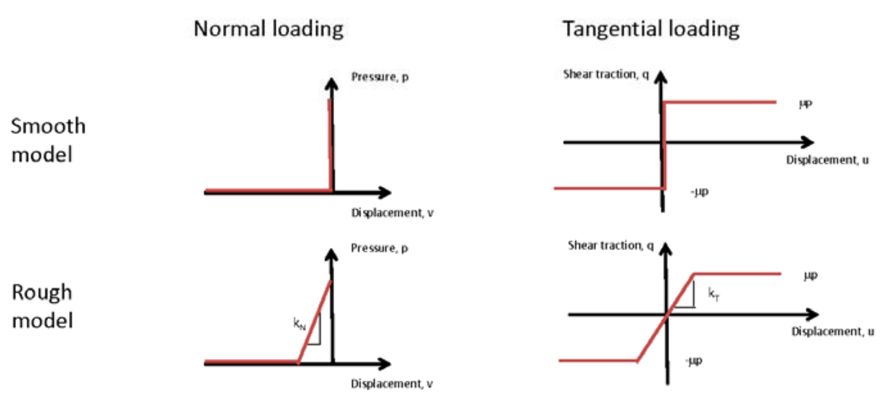


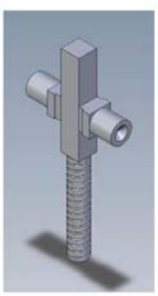
- For a smooth contact model we only need the friction coefficient, μ
  - Predicting μ from surface geometry, material properties, etc. is a difficult problem
- A rough contact model also needs k_N and k_T
  - In principle, these might be rather easier to predict
  - However we need reliable experimental data for model validation

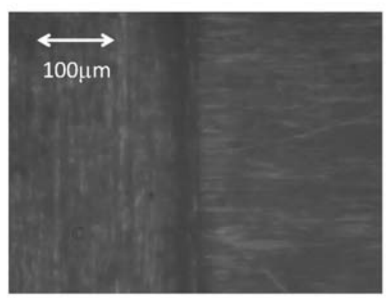
Experimental Techniques

- Two approaches to contact stiffness measurement are available:
  - Direct Measurement of Load and Displacement
    - Digital Image Correlation (Mulvihill, Kartal et al)
    - Laser Velocimetry (Schwingshackl, Gola, et al)
  - Indirect Measurements
    - Ultrasound (Dwyer-Joyce et al)

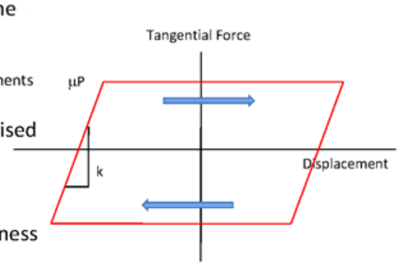
4

Measured and idealised hysteresis loops





- A bar is clamped between two pads in a test machine
  - Can be loaded in oscillatory sliding and the hysteresis loop measured
  - Digital image correlation can be used to obtain local measurements of displacement
- In tangential loading, the idealised loop is characterised by contact stiffness, k and friction coefficient, μ
  - These can be reasonably representative of real loops (at least initially)
- Similar measurements can be made for normal stiffness
- Ti 6/4 is used in the current work (ground surfaces)



Comparisons (PAMFJP, 2010)

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Normalised Tangential Contact Stiffness (N/m/mm²)

	Oxford	Imperial
Ti Smooth	(9.1)x10^6	(17.3-24.6)x10^6
Ti Rough	(10.0-10.6)x10^6	(17.6-25.9)x10^6
Ni Smooth	(10.3-11.9)x10^6	(24.1-47.6)x10^6

- Imperial contact stiffness (normalised by area) is higher by factors of 2-4

However both are likely to be incorrect:

- Imperial displacement measurement locations include some additional compliance
- Both measurements are normalised with respect to average pressure
  - Local pressure at Oxford measurement location will be lower
  - But we can't measure pressure (though we may be able to measure strain normal to the surfaces)

A closer look: typical DIC results

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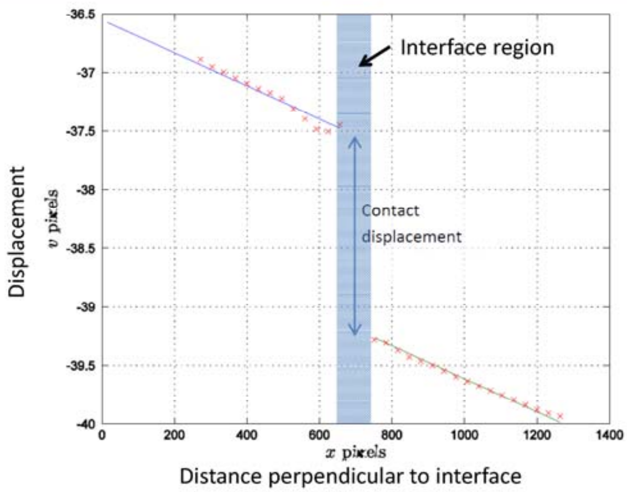
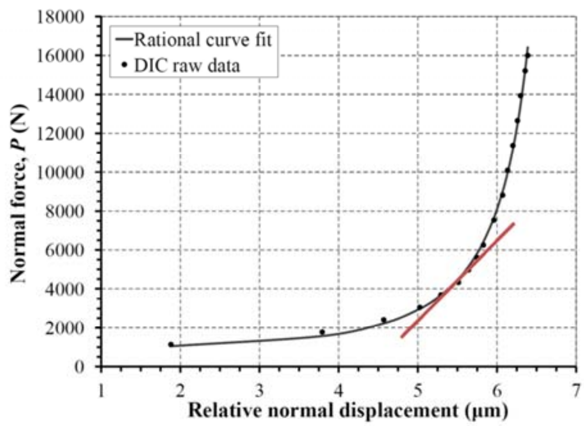


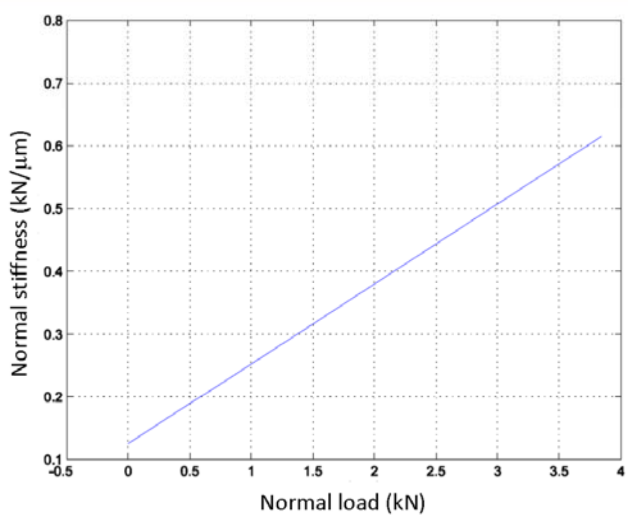
- Measured displacements across the interface are not continuous
- Note importance of measurement location
- To obtain tangent stiffness, one needs to measure at a number of loads

DIC Measurements



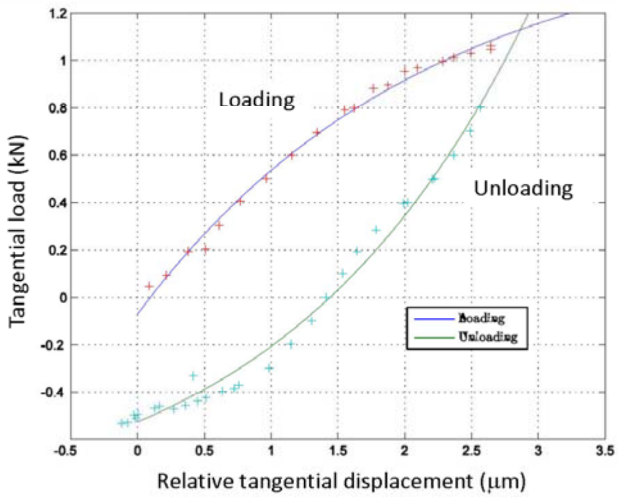
- DIC results give relative displacement as a function of load
- Curve fitting and differentiation gives tangent stiffness
- Note that stiffness is a function of normal load (as predicted by simple models)

Results: normal loading



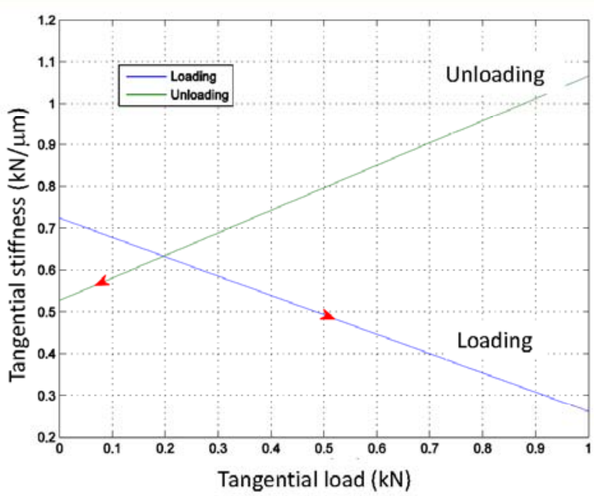
Calculated stiffness from exponential curve fit gives a normal stiffness that is proportional to normal load

Results: tangential loading



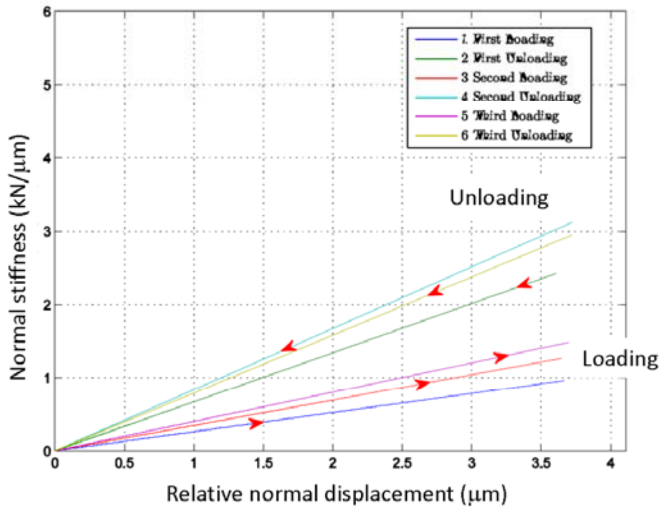
- Behaviour shows reducing tangential stiffness with tangential load
- There is considerable hysteresis between the loading and unloading curves

Results: tangential loading



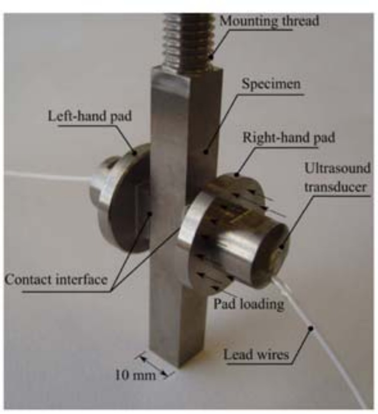
- Stiffness reduces with linearly with load (for chosen exponential curve fit)
  - Softening behaviour
- Note large increase in stiffness on load reversal
  - Similar to macroscopic behaviour in partial slip

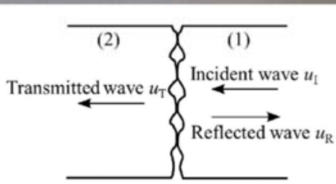
Results: repeated normal loading



- Unloading stiffness is higher than loading stiffness
- Stiffness increases with each successive loading (hardening behaviour)

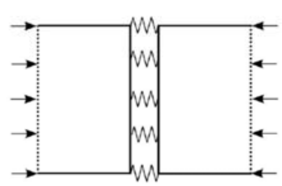
Ultrasound measurements



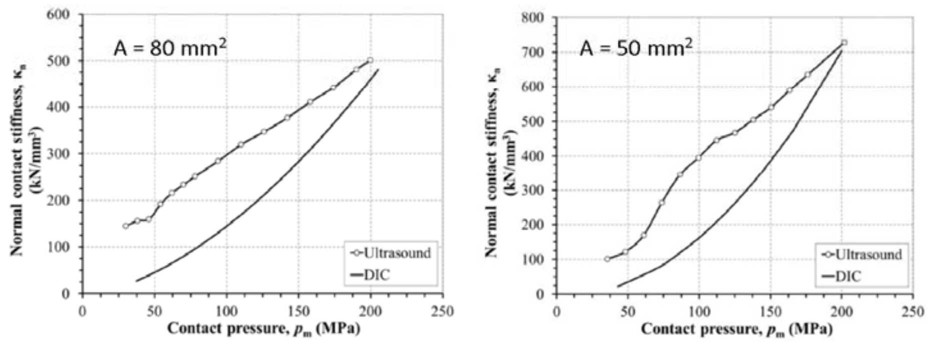


- Collaboration with University of Sheffield
- Normal and tangential stiffness can be derived from reflection coefficient, R, of normal and shear waves

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

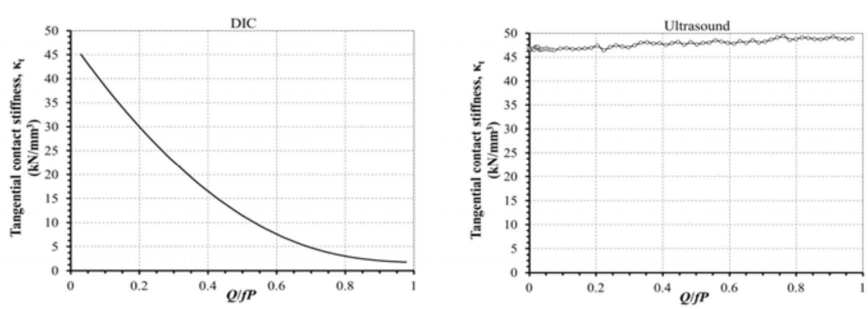


Comparison: Normal stiffness



- Results shown are normalised by contact area
- Ultrasound stiffness appears higher, particularly at low normal loads, but agreement is reasonable
- Both measured values increase with normal load

Comparison: Tangential Stiffness



- Graphs above show variation of tangential stiffness with tangential load, Q.
- Note that (in this case) initial value is very similar, but variation with tangential load Q is very different
- Ultrasound is measuring an unloading stiffness, whereas DIC measures a loading stiffness

Loading and unloading stiffness

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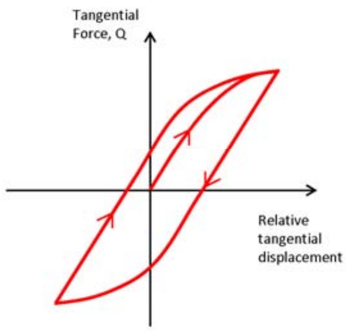

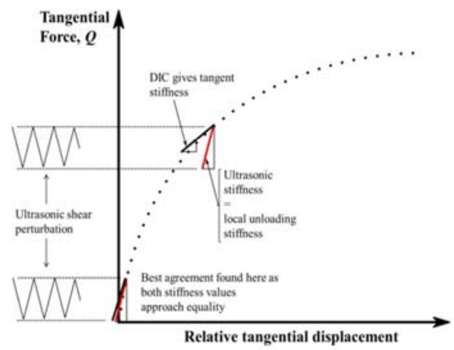

- For normal stiffness, loading and unloading values will be very similar
- In the case of tangential stiffness, the load is transmitted by friction, and unloading will produce 'stick'
- Hence loading and unloading stiffnesses will be very different
- Ultrasound measures an unloading stiffness
- Which do we want in dynamics problems?

Conclusions – the way forward?

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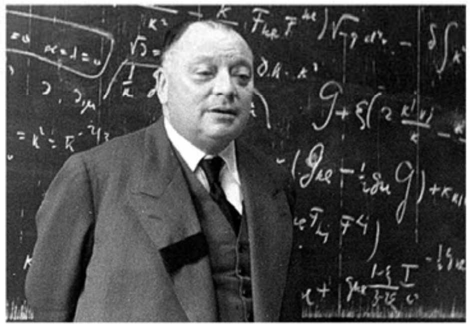
- Stiffness measurements are not straightforward
    - We can't even reliably measure what we want to model
- Stiffness is not a surface property
    - Normal stiffness will depend on normal pressure
    - Tangential stiffness will depend on normal pressure and on local shear traction
    - Stiffness may increase with repeated loading
- The example has looked at stiffness
    - Friction will be much more difficult
    - However, we do need physics-based models of these phenomena
- There remains the challenge of how to incorporate such complexity in our joint models – how much do we need?
- The contact mechanics community needs to understand more about vibrations and dynamicists need to understand more about contact mechanics.

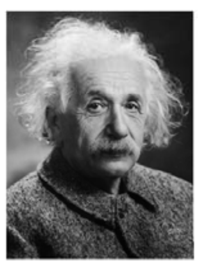
17

Wolfgang Pauli 1900-1968



- “God created solids, but surfaces are the work of the Devil”

Albert Einstein 1979 - 1955



- “If you can’t explain it simply, you don’t understand it well enough”

5.8. Concluding Perspectives

Dan Inman: Summary from an Outsider: "Divide and Conquer"

Summary from an Outsider
"Divide and Conquer"

D J Inman

Applicability:

- Defining the ideal joint (*iJoint*)
  - Redesign connections to be “model” friendly
  - Stiffness, damping, integrity
- Classification of types of joints
- Classification of application types
  - Context/application is key in discussing joints
  - Level of importance per application

Suggestions:

- A lot of focus was on damping
- Perhaps the problem should be “divide and concur”
- Defining an ideal connection has the potential for game changing importance
- Be clear about what is wanted “O”

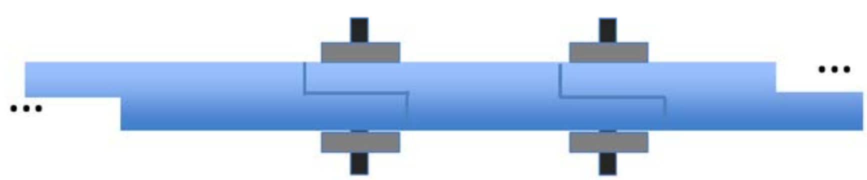
Repeatability:

- What and when does it matter?
- Simple joint experiment to validate codes against
- Define what is good enough
  - Statistical values vs. bounds vs. factor of safety
- Round Robin: specify protocol, fixture and what parameters should be measured

Suggestions: iJoint/eBolt

Significant Thought: fluids are random non linear collisions that can be modeled as a linear continuum*

An expanding series of nearly identical joints**



Etc.

At some point does it become linear? Repeatable?

*Norbert Hoffman, ** Pablo Tarazaga

Predictability

- Classify what one wants to predict
  - Wear, failure
- Classify the purpose
  - Design, physical understanding, performance
- Define accuracy
- Do simple experiments for physics
- Input-output models for systems

Suggestions

- Clarify accuracy required
- Clarify purpose
- Determine if Coulomb friction is correct
- Make sure basic physics and system researchers interact
- Know X and O, and have a plan B

Applications and Way Forward

- Defining the mix of physics and system level analysis
- Quantifying the cost of not solving these problems
- Building "Pathways to Impact"
- Refine and sharpen objectives (O)
  – Understanding length scale vs complexity

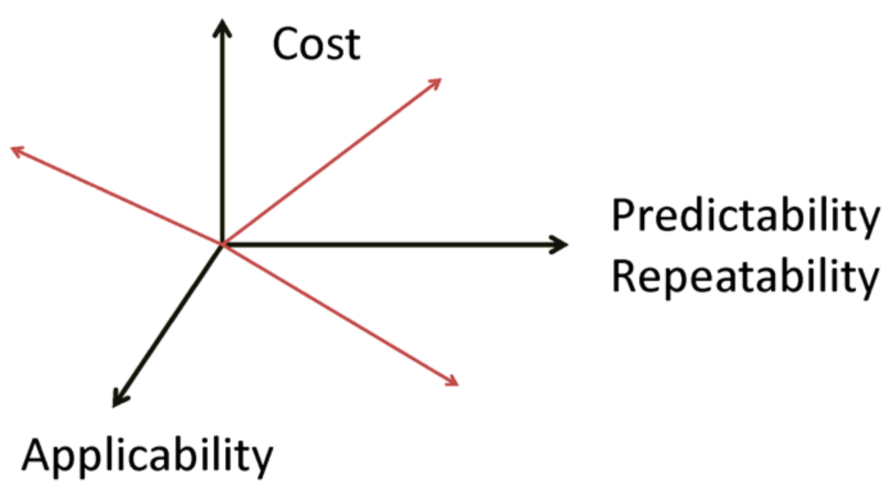
Suggestions

- Enable disparate researchers to work together
  – *Specifically* static and dynamic joint modelers
- Realize the need to have multiple models for the same system under different applications
- Start the middle ground
  – Something in between the single joint model and the entire structure

Know X and O, and have a plan B

Thanks for a great experience

The dilemma



Observations

Adnan Akay
Bilkent University

Comments on the 4th Workshop

- Lots of ideas.
- Bi-modal distribution of presentations – some on nominally the same problems and those on somewhat unrelated problems. All very helpful.
- Probably missing some earlier work that may have significance on the current problem
- Discussions focused almost entirely on joints at the risk of missing the ultimate objectives – vibration reduction and failure prevention in a jointed structure.
- Friction is assumed to be the sole source of damping
- Bottom line: The problem has been around for a while and will insist to be around for a while more … much like brake noise.

Joints

- Joints are essential elements of a structure and need to be considered in context.
- Joints have a *primary* Functional Requirement (FR) in the design of a structure.
  - This role is fairly well understood, although problems still arise as requirements change.

Joints

- Joints happen to play a significant role in the overall vibration response of a system.
- As a result, joints are viewed as having *secondary* functional requirements (FR)
  - How to regulate (maximize) system damping.
- Joints should also be viewed as "complex impedance nodes." They help dissipate energy but also modify, perhaps regulate,
  - Transmission of vibratory energy flow within the structure.

    <u>*We have not heard much about this.*</u>

Joints - uncertain, unrepeatable, and unpredictable

- All related to the secondary functional requirements
- Suggest existence of hidden parameters – that seemingly do not influence the primary FR.
- Also suggest a certain dependence on the dynamic response of the structure within which joints exist.
- Surface geometry, friction are not so hidden but their particulars seem to be elusive.

Focus on the Physics of Joints

- It is highly likely that the important, yet hidden, parameters are at micro scale, including flatness of the surfaces.
- One difficulty arises when trying to deduce the role of micro-scale parameters from macro behavior of the joints (using the conventional measurement and analysis tools).
- Consequently, we "integrate" micro parameters, under dynamic response, to compare with macro responses.
  - Consider friction in a joint
    - Micro stick-slip, unobserved at macro-scale may be one of the "hidden parameters." Tectonic plate example.
  - What exactly are the sources of nonlinearities?
    - Experiments suggest that time-dependent BCs due to slippage is a major cause.

A few reminders

- Quantities such as loss factor, coefficient of friction, viscosity represent with one number the behavior of millions of degrees of freedom.
  - Terms such as coefficient of friction have been very helpful in classical engineering but their usefulness in many current problems may be very limited and counter productive, since they hide important details.

- Since words define thoughts, we need to re-examine the traditional terminologies used to describe phenomena, for they may not be as adequate for new concepts.

Recommendations

- Need a road map to identify and quantify
  - Macro-scale impedance parameters and their sensitivity of micro-scale parameters and operational conditions (dynamics) [Joint as a dial]
- Need an anthology or hierarchy of joints according to geometric configurations, types, etc.
  - They can be mapped to or from industries and applications.
- Need to consider joints globally
  - Does the same type of joint have the same influence on the system irrespective of its location?

Recommendations

- A more useful understanding of concepts such as friction is essential to better understanding of joint behavior,
  - Which depends on better understanding of molecular-level dissipation processes, which depends on quantum dissipation.
  - It will take time…
  - → Develop methods to reduce and proceed with the uncertainties
- Collection of seemingly deterministic and well defined components can lead to a "complex system," defined as that having emergent behavior.
  - Joints may be such an enabler.
  - A look at complex system analysis methods may be helpful.

Next Steps

- Develop clear problem statement(s)
- Identify unknown but important knowledge gaps relevant to joint-related problems but may fall outside of this community's knowledge base
  - Have short tutorials on such topics
- Reduced order models & experiments
- Focus on the physics of joints

Congratulations and thank you

- A rich and vibrant research community is focusing on the problem, with wisdom and new ideas
- Great workshop with many new ideas.
- And a lot of opportunities for more progress.

6. SUMMARY

The Fourth International Workshop on Jointed Structures was held in October 2015. At this workshop, 45 researchers from across the world came together to discuss progress made since the previous workshop (in 2012) and to develop a roadmap for the research directions in the area of mechanics of jointed structures over the next five to ten years.

Previous workshops had focused on defining a set of actions and challenges. The progress towards those actions and challenges were summarized to start the workshop, and can be found in detail in [5]. The direct outcome of this fourth workshop was to define a new roadmap for the future of joints research. This roadmap is heavily focused on strategy, which is defined to consist of four parts. First, there is a clear understanding of the current state (see [5]). Second, a clearly defined objective is needed. For the joints community, it was put forth that this objective could be:

A validated method for the Design and Analysis of dynamically loaded structures with frictional joints.

Third, an inventory of the possible tools to lead to the objective is needed. This is the concept behind the roadmap, which is developed at three different levels. The highest level, the Atlas, describes the tools in terms of seven themes that are broad and encompassing of multiple disciplines outside of joint mechanics in order to attract new researchers to the challenges specific to joints. These seven themes are suggested to be:

- Building external consensus for support;
- Experimental investigation of repeatability and variability;
- Techniques to characterize/identify nonlinearities;
- Constitutive model development;
- Numerical methods for nonlinear dynamics;
- Multiscale investigation of interfacial physics; and
- Uncertainty-based strategies for modeling and experiments.

Lastly, the fourth component of a strategy is a plan for how to achieve the objective. This plan is currently under revision and will be eventually published as the Dartington Declaration.

7. REFERENCES

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3. D. J. Segalman, L. A. Bergman, and D. J. Ewins, “Report on the SNL/AWE/NSF International Workshop on Joint Mechanics, Dartington, United Kingdom, 27-29 April 2009,” 2009, *SAND2010-5458*, Sandia National Laboratories.

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5. M. R. W. Brake (Ed.), “The Mechanics of Jointed Structures,” 2007, *Springer*.

8. DISTRIBUTION

All distributions are electronic unless otherwise noted.

1	ASME Research Committee on the Mechanics of Jointed Structures	
1	SNL Fasteners Working Group	
1	MS 0815	Center 1500
1	MS 0899	Technical Library, 9536

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