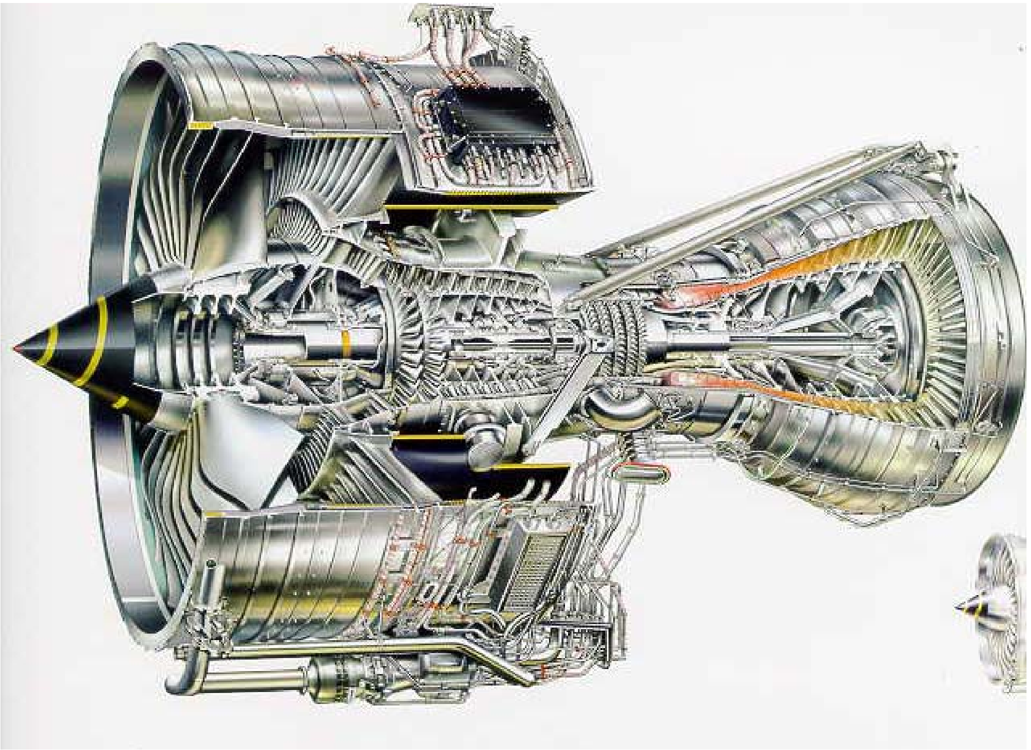
Vibration related examples of uncertainty issues in the design and validation of gas turbine components and systems.

Presenter: Hilmi Kurt-Elli **Company Specialist - Vibration,**
Rolls-Royce plc, United Kingdom

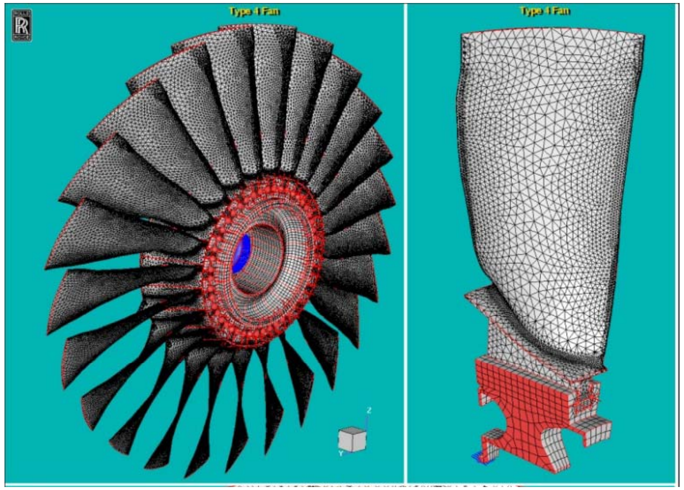
Examples of design and validation vibration issues for gas turbine components and systems

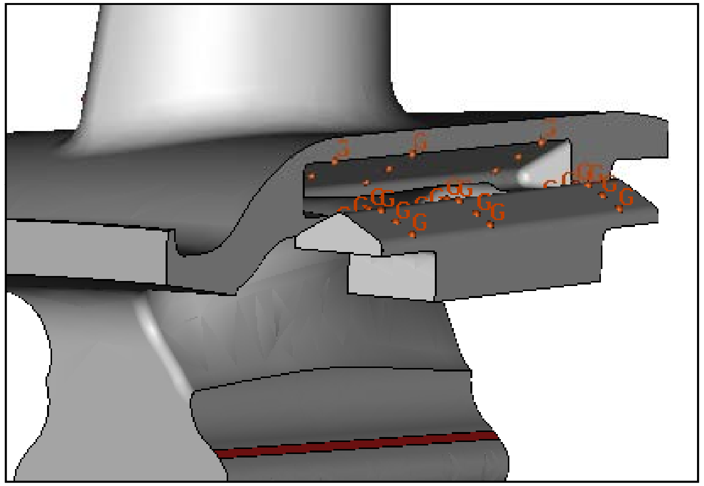
- Rotors – blades
  - Upstream wake excitation
- Vanes
  - Up/down– stream rotor excitation
- Mechanical, fluid, combustion, acoustic excitation & complex built up structure behaviour
- Accessories
- Mechanical damping
- Measurement technology limits coverage (e.g. Only 6 of 96 turbine blades gauged) AND experiments very expensive



Design-analysis approach

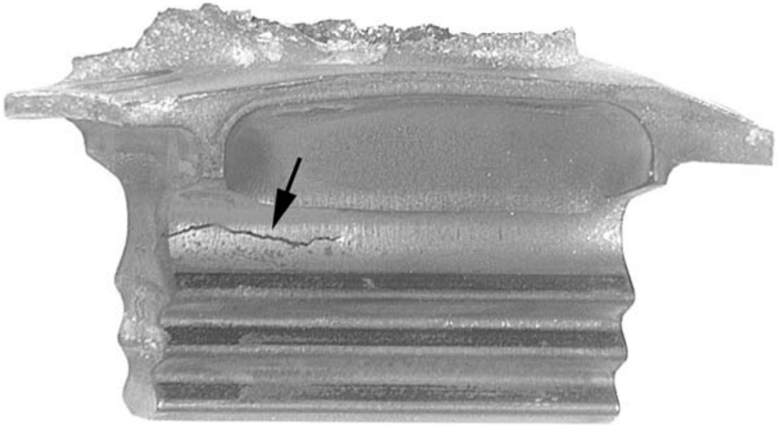
- Traditional, deterministic design approach (with state of art analysis tools).
  - Periodic forcing (engine flow conditions)
  - System nominal and represented by varying fidelity level of finite element models. Models inevitably large because of need to represent physics.
  - Linear fan rotor model; non-linear turbine model with u/p damper – CFD => forcing
- Nominal f.e. Model analysed; safety factor applied
- Safety factor – dependent on experience and our view of uncertainties in analysis, variability across nominally identical components and engines, etc.
- Design validation by testing & measurement





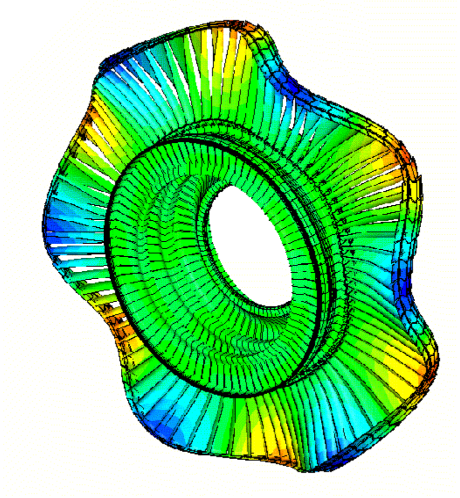
Motivation & Interest in Uncertainty Quantification

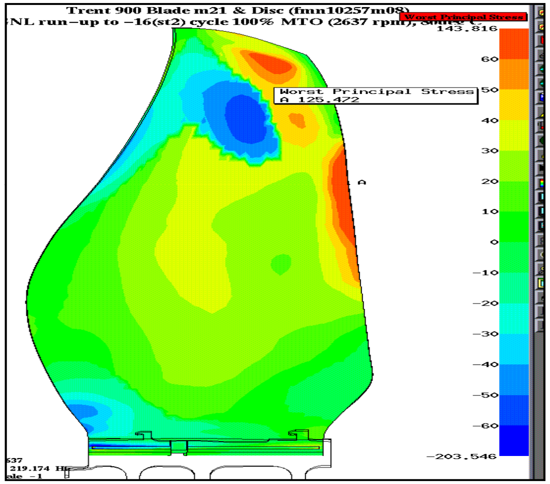
- **Strong desire to reduce weight and ensure safety improved via understanding / significance of margins (factors of safety do not give us this)**
- **High effort and yet still have costly failures.**
- **Analysis methods development requires better UQ**
  - **mistuning issue & rotor identification**



Prediction and validation of rotor vibration

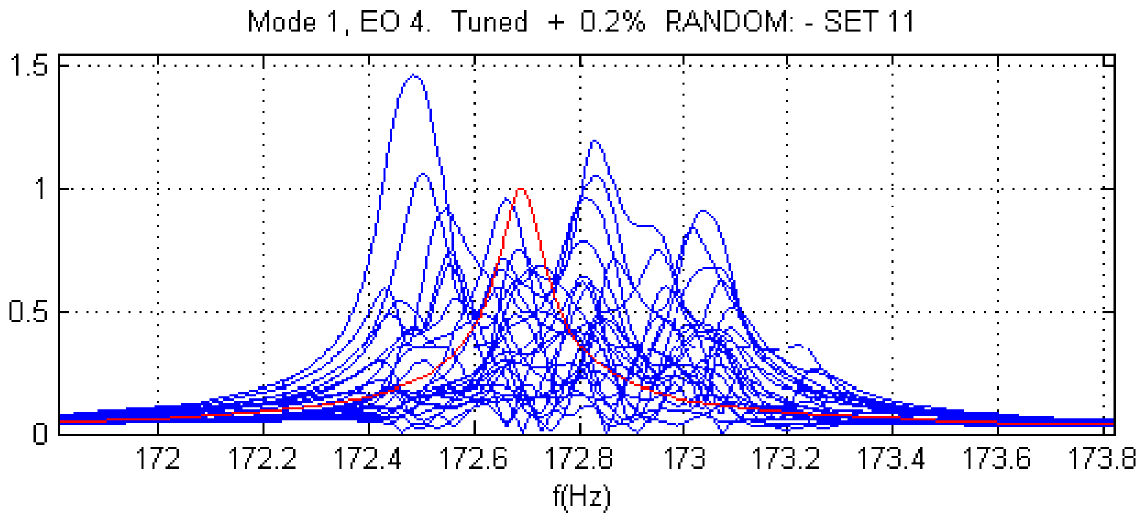
- **Engine Order Excitation - Typically forcing arising from circumferential disturbance in upstream flow; e.g. Wakes from vanes**
  - **Example Case: N = 24 blades, M = 4th Engine Order =>**
  - **Interblade Phase Angle, IBPA , $\Phi = ((2 * \text{pi}) * M) / N$**
  - **Blade 1 : $f = a \sin(wt)$ Blade 2 : $f = a \sin(wt + \Phi)$ Blade 3 : $f = a \sin(wt + 2\Phi)$ etc.**
- **Tuned – each blade identical**
- **Mistuned – very small differences across <u>individual</u> blades**
  - **+/- 2 % variation on cantilevered blades**

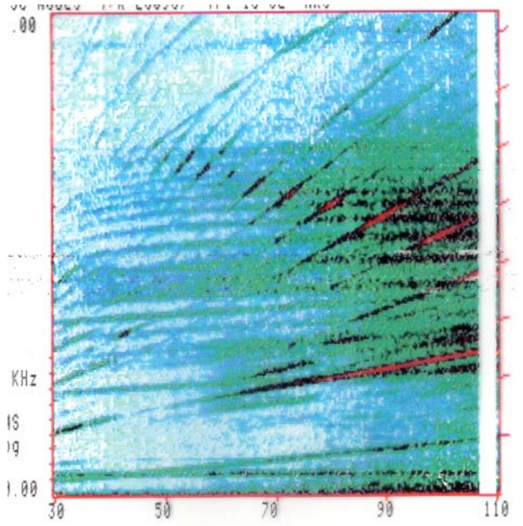




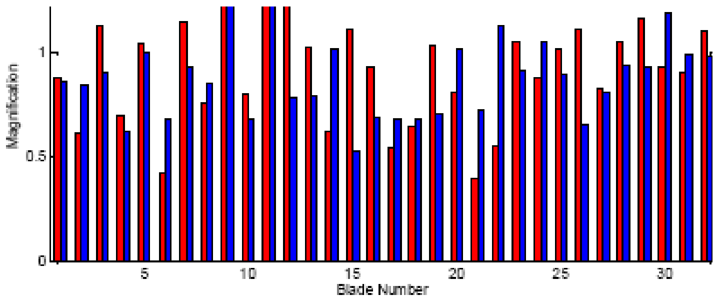
Engine Order forced response

- Engine Order responses of each blade (tuned vs mistuned). 1st family of modes. Large variation across blades for mistuned rotor – high localisation





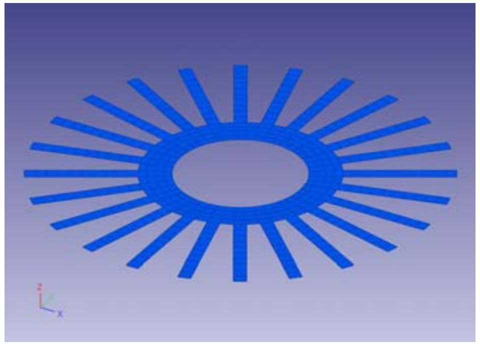
- Engine Order response plotting max amplitude of each blade



Mistuned rotor

- **Response distributions can be estimated but not validated**
  - **computationally expensive although confident for linear cases**
  - **for non-linear cases (friction contact dampers, etc) –computationally expensive, and significant uncertainty on accuracy**
- **Validation problematic – distributions not measured; need to validate against limited data => current drive to model 'the' tested item ; i.e. Synthesis of bespoke fe model describing behaviour of specific test rotor given measured assembly responses**
  - **measured response proportional to damping, aero excitation, modal behaviour – therefore unclear what in system simulation is in error when differences in response observed**
  - **integrating actual mistuned modal behaviour rather than the nominal model removes one unknown from the assessment of the 'goodness of overall prediction tools'**
- **Model updating approach, changing tuned (nominal) model, to describe mistuned behaviour**
  - **Artificial nature of updates at individual blade level & very high sensitivities is what makes this an unusual process**

Model updating approach: mistuned rotor



- A nominal fe rotor model is perturbed by adding one (or more) small masses attached to each blade model. These masses become the updating parameters in a ‘model updating strategy’
  - This is an artificial parameter update but acceptable because it has the desired effect on each blade-alone frequency; and the respective blade to blade frequencies dictate the assembly behaviour
- Under forcing simulating laboratory or engine running conditions, simulate the forced response of the perturbed rotor model and compare with the reference ‘test’ responses
  - For the purposes of this work, the reference ‘test responses’ themselves are simulated using a finite model of the mistuned model
- The damping also may be assessed by updating global modal damping parameters (and potentially individual blade damping parameters)
- Error between reference Frequency Response Functions and those predicted for given perturbed model minimised in an iterative manner

WORK STATUS

- Study in very early stages, initial indications that updating process can converge to give accurate results but process itself can be ‘sensitive’
- Workshop gives opportunity to highlight mistuning rotor identification work and receive feedback

Concluding remarks

- Localisation presents problems – system parameter variations may be small but combined system response can be very large
- In general and for new designs, not clear how to define the potential variability in component geometry.
- General structural dynamic problems such that variablities (real and or perceived due to ignorance) are too large for probabilistic design methods to be adopted
- However, for validation purposes, rotor identification via an updating approach may offer a way forward

BACKUP foil - Tuned assembly & behaviour

- **Natural frequency versus nodal diameter (or cyclic / Fourier index; modes orthogonal pairs – circumferential spatial orientation different)**
- **Forcing & tuned modes act such that 2EO only excites 2nd ND mode, etc. i.e. Other combinations orthogonal functions leading to zero modal force**
- **Extent of 'veering' behaviour function of blade to disc stiffness**