Muzio M. Gola

AERMEC LAB
AEROMECHANICAL LABORATORY

Second Workshop on Joints Modelling
Dartington, April 27/29 2009
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Dept. of Mechanical Engineering – the team

Co-ordinator

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BRASIL

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(end 2008)
Current main contracts and research programs

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

**Italian government Research Grants**
- PRIN Design criteria for mistuned turbomachinery (2006-2009)

**Research Contracts with AVIO Group**
- High temperature tribology for turbine materials (2007-2009)
- Design of damper rings for aerospace application (2008-2010)
- Study of turbine disk vibrations with MISTUNING (2007-2008)
Research lines

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

Complementary activities
- Tribology, wear measurement
- Spin-test rig (work in progress)
- Dynamics of turbine disks with asymmetrical effects (MISTUNING)
- Real time evaluation of temperature and thermal stresses at critical locations of turbine disc (disc lifing)
- X Ray evaluation of residual stress in turbine components
Contact mechanics - 1

Working range:
- Displacement: 0.1 mm - 100mm
- Normal loading: 1kg a 10kg
- Operating frequency: 1 - 100Hz
- Induction heating 20 – 80 kHz
- Temperature : 20 - 800°C
Turbine blades vibration: friction damping

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

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

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

Hertzian theory extended to the case of tangential loading

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


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

Other models consider another factors, e. g. roughness (Bowden & Tabor, Greenwood & Williamson, Archard, O’Connor & Johnson), velocity, etc.
<table>
<thead>
<tr>
<th>The test rig at Imperial College 1999</th>
<th>Friction behaviour associated with fretting fatigue</th>
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<tr>
<td>Flat-on-flat contact type, measurements after wearing-off the surfaces</td>
<td>Reference to Murthy et al. (2002), Murthy and Farris (2003), Matlik and Farris (2003), who investigated fretting fatigue as a function of temperature in advanced materials utilized in turbine engine components.</td>
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Tests at temperature up to 610°C in the contact region demonstrated that the friction coefficient increased with the wear of contact surfaces, and that the friction coefficient is dependent on the contact history.

in co-operation with Prof. A. Akay
Carnegie Mellon University


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

Measurement system: Force transducers measure the tangential force. Relative displacements by two LDV beams.
Design Requirements at room temperature

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

For high temperatures, there are additional requirements

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

Mechanism

constant pre-load

material: Nimonic C263

Conical Specimens

the Pittsburgh-Polito 1D test rig – room temperature
2 Vibrometric laser Polytech single-point: controller OFV-5000, sensor head OFV-505

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

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

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

Induction heating machine: MTC-6, power 6 kW, operating frequency: 20 to 80 kHz.
Contact mechanics - 10

The heating system

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

  *MTC-6 Induction Machine:*
  - Nominal power: 6kW;
  - Working frequency: 20-80 kHz

- Temperature measurements: **k-type thermocouple**, placed near the contact

- Temperature control: **NI-card + Labview**

- **Error** on temperature measurements: estimated by FEM thermal analysis

\[ Temp_T = Temp_M \cdot (1 \pm 1.5\%) \pm 8^\circ C \]
Contact mechanics - 11

the Polito 1D test rig - high temperature
Contact mechanics - 12

the Polito 1D test rig
- high temperature
Determination of contact parameters from hysteresis cycles

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

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

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

\[
E = E_{\text{microslip}} + E_{\text{gross-slip}}
\]

\[
E = 4 \cdot (u \cdot \mu \cdot N) - \frac{24}{5} \cdot \frac{(\mu \cdot N)^2}{k_T}
\]
Behaviour of $K_{Re}$ and $K_{Im}$ of the cycles at high temperature

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

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

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

Dimensionless equiv. contact stiffness:

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

Hysteretic effective contact damping:

$$\tilde{c}_e(\tilde{X}) = K_{Im}(\tilde{X}) = \frac{1}{\tilde{X} \mu P} \int_0^{2\pi} q \sin(\theta) d\theta$$
First set of experiments: specimens of Inconel 100

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

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

Temperature Range: T=20°C up to 800°C

Hysteresis cycles measured for different temperatures (Normal load: 61N)
While the friction coefficient decreases rather sharply from room temperature up to 200°C and then becomes almost stable, yield and ultimate strength are practically constant up to almost 800°C and the Young modulus decreases almost linearly with temperature in the same temperature range. Therefore no simple relationship seems to exist between the friction coefficient and the basic material properties.
If the contact followed the Cattaneo-Mindlin model, the stiffness would be proportional to the Young modulus of the specimens raised to the power of 2/3. The behavior of the Young’s modulus with temperature is also reported in the diagrams. But it can be noted that the variation of Young’s modulus with temperature does not explain the variation of contact stiffness, and so far no explanation has been found for the behavior of the stiffness with temperature.
More problems found in the Test Procedure

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

20-800°C: d>1mm

Materials under test now (2008/2009 - AVIO restricted access to data): RENE 77, 80, 108, 125; CMSX-4; Inconel 718 - with and without T800 coating (high roughness and hardness)
Modelling damping components -1
underplatform dampers - test rigs

Non contact electromagnetic excitation
Modelling damping components -2
vane segments and shrouded blades – test rigs

Laser scanner measurement
Modelling damping components -3 blade root damping characterization

The Test Rig

Numerical Contact model based on contact mechanics principles

Experimental results

Numerical results
Dynamic response of turbine discs - 1

Non linear forced response calculation with friction damping

POLI.contact software

underplatform damper

Rigid damper

Elastic damper

Kinematic assumptions

No kinematic assumptions

contact Model

shrouded blades and vane segments

Direct blade-to-blade contact → simple contact kinematics

contact model
Dynamic response of turbine discs - 2

Flow-chart of the numerical code developed for the forced response calculation of bladed disks with shrouds.
Reduction technique to compute the forced response of frictionally damped bladed disks.

\[
\begin{align*}
\{X\} &= [\Psi] \cdot \{q\}; \\
N &= n; \\
N &>> n
\end{align*}
\]

\[\begin{bmatrix} K_R \\ M_R \end{bmatrix} = [\Psi]^T \cdot [K] \cdot [\Psi] \]
\[\{F_R\} = [\Psi]^T \cdot \{F\}\]

- Component Mode Synthesis (Craig-Bampton).

\[\{q\} = \begin{bmatrix} X_M \\ \eta_S \end{bmatrix}\]
Complementary activities

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

Measurement of wear on contact surfaces
Validation of theoretical wear models

Point contact test

$10 \cdot 10^6$ cycles
amplitude $30 \, \mu m$

$T = 900 \, ^\circ C$
Spin-test rig (work in progress)

- Test disk: diameter up to 650 mm
- Motor
- Telemetry system (12 channels)
- Vacuum chamber
- Containment ring
- Non-contact magnetic excitation
- Rotative seal
- Rotation speed up to 4000 rpm
Dynamics of turbine disks with asymmetrical effects (MISTUNING)

Identification model of mistunning

Comparison of different reduction techniques and improvement

Experimental validation by means of dummy disks

Magnification Factor: 1.42
Real time evaluation of temperature and thermal stresses at critical locations of turbine disc (disc lifing)

Original methodologies for temperature and thermal stress monitoring based on the modal reduction techniques and on the Green’s function theory for both linear and non-linear applications.
X Ray evaluation of residual stress in turbine components

Residual stress measurements by means of X-ray diffractometer (Siemens D5005)
VERDI EU Project (6th FWP 2005-2008)
Validation of numerical model for mechanical working (milling and turning) simulation

Residual stress versus depth

![Graph showing residual stress versus depth](image)