Muzio M. Gola

DIPARTIMENTO di MECCANICA

POLITECNICO di TORINO

AERMEC LAB

AEROMECHANICAL LABORATORY

Second Workshop on Joints Modelling

Dartington , April 27/29 2009

Politecnico di Torino

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

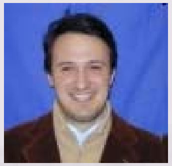
Co-ordinator

M. M. **Gola**

Assistant professors

T. **Berruti**

D. **Botto**



S. **Zucca**

Research Assistants

C. **Firrone**

M. **Lavella**

A. **Campagna**

D.**Zanello**

V. **Maschio**

PhD students

S. **Pavone**

P. **Vargiu**

L. **Tong (CHINA)**

Visiting researchers

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

C. **Siewert**
University of Hannover
GERMANY

(end 2008)

Current main contracts and research programs

European Union Projects

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

Italian government Research Grants

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

Research Contracts with AVIO Group

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

Research lines

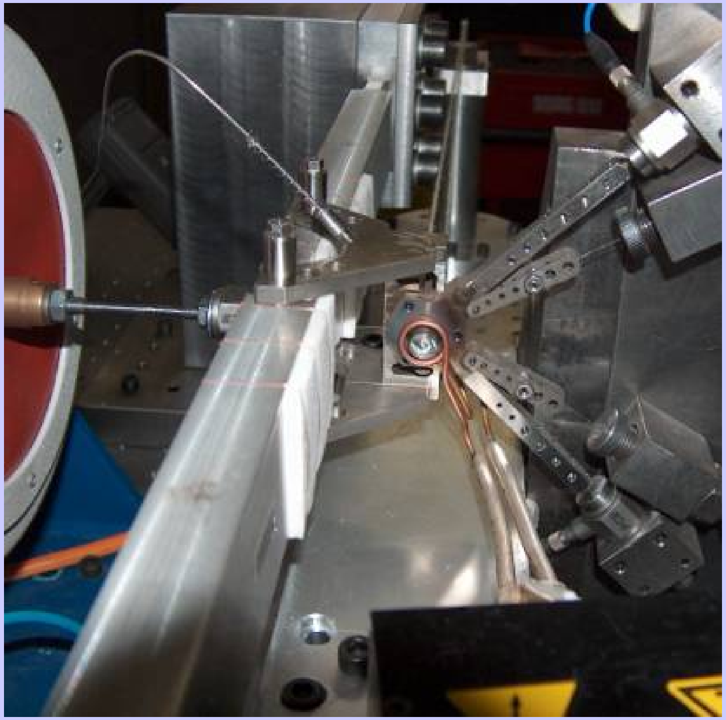
- Contact mechanics & contact modelling
- Modelling damping components (underplatform, shroud, blade root)
- Dynamic response of turbine discs

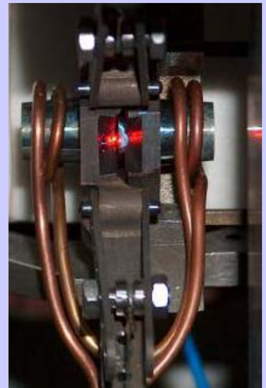
Complementary activities

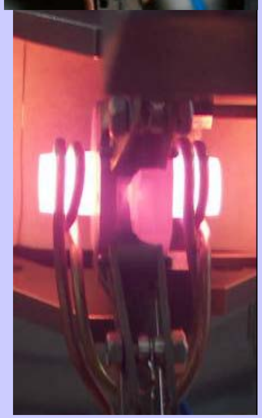
- Tribology, wear measurement
- Spin-test rig (work in progress)
- Dynamics of turbine disks with asymmetrical effects (MISTUNING)
- Real time evaluation of temperature and thermal stresses at critical locations of turbine disc (disc lifing)
- X Ray evaluation of residual stress in turbine components

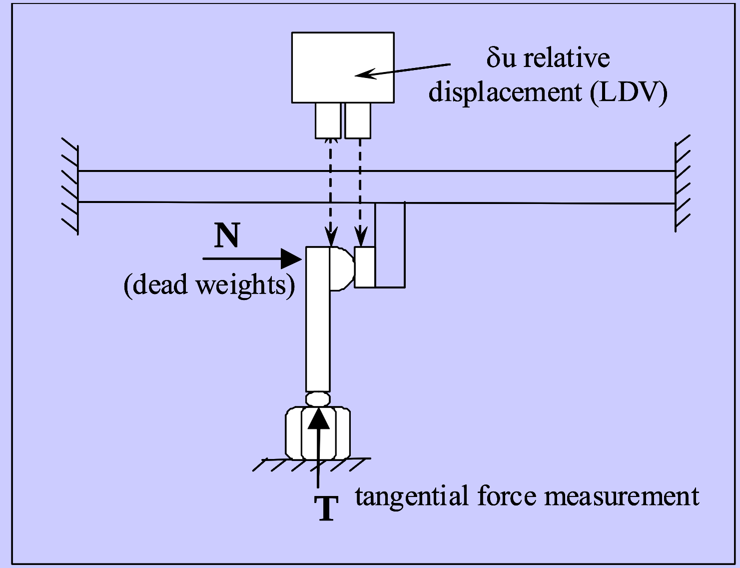
Contact mechanics - 1

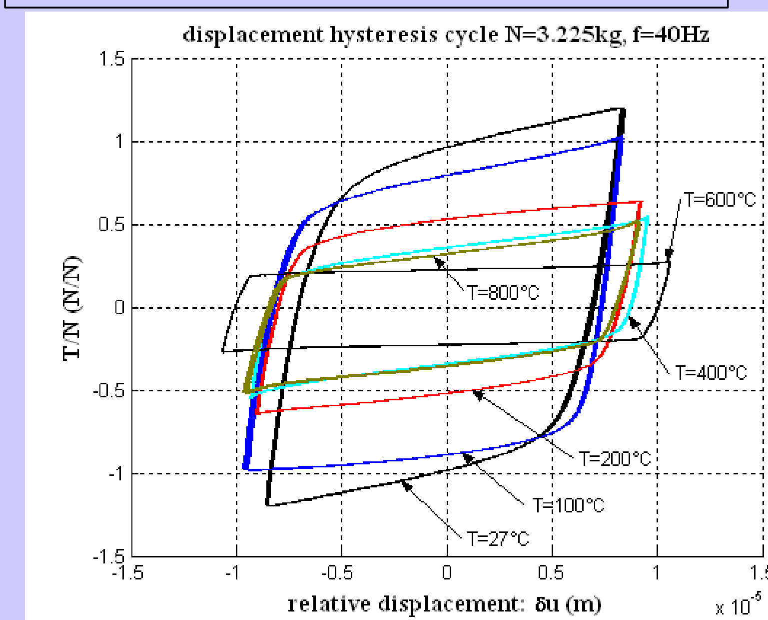
High Temperature Test Rig









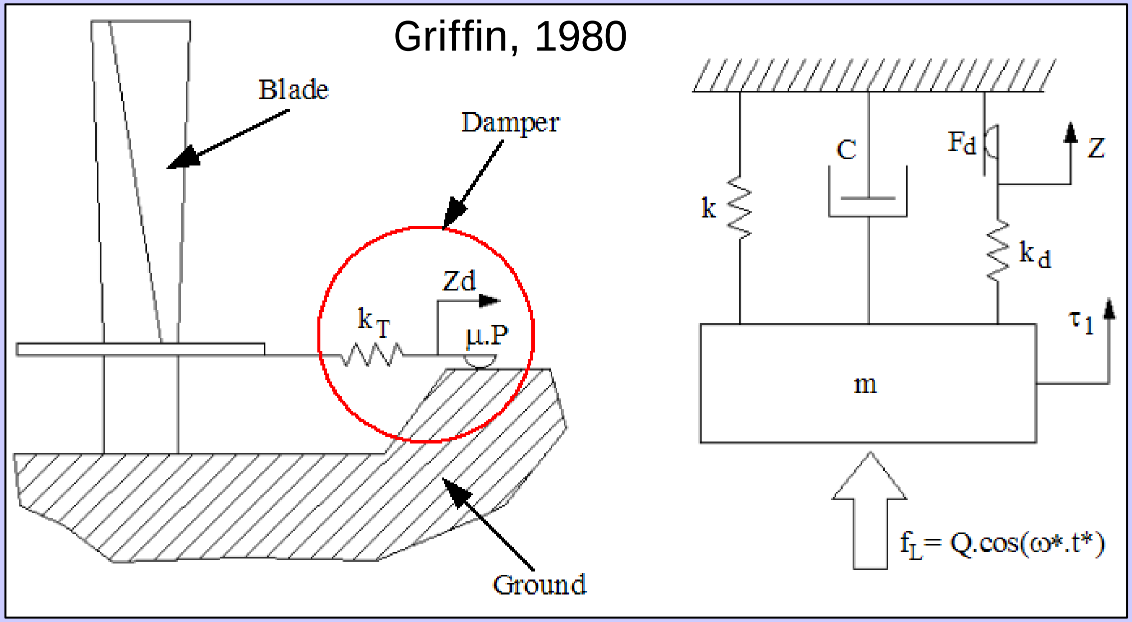


Working range:

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

Contact mechanics - 2

Turbine blades vibration: friction damping



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

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

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

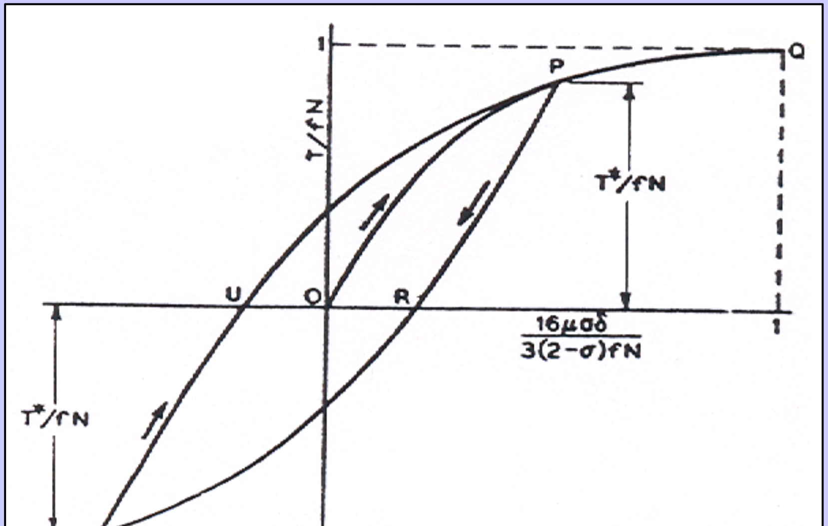
Contact mechanics - 3

Cattaneo-Mindlin contact model

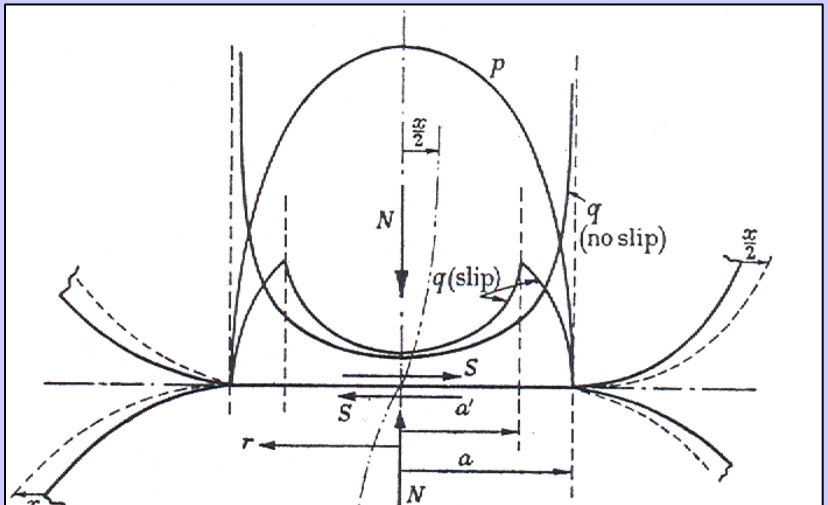
Hertzian theory extended to the case of tangential loading

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

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



Mindlin et al., 1952



Johnson, 1954

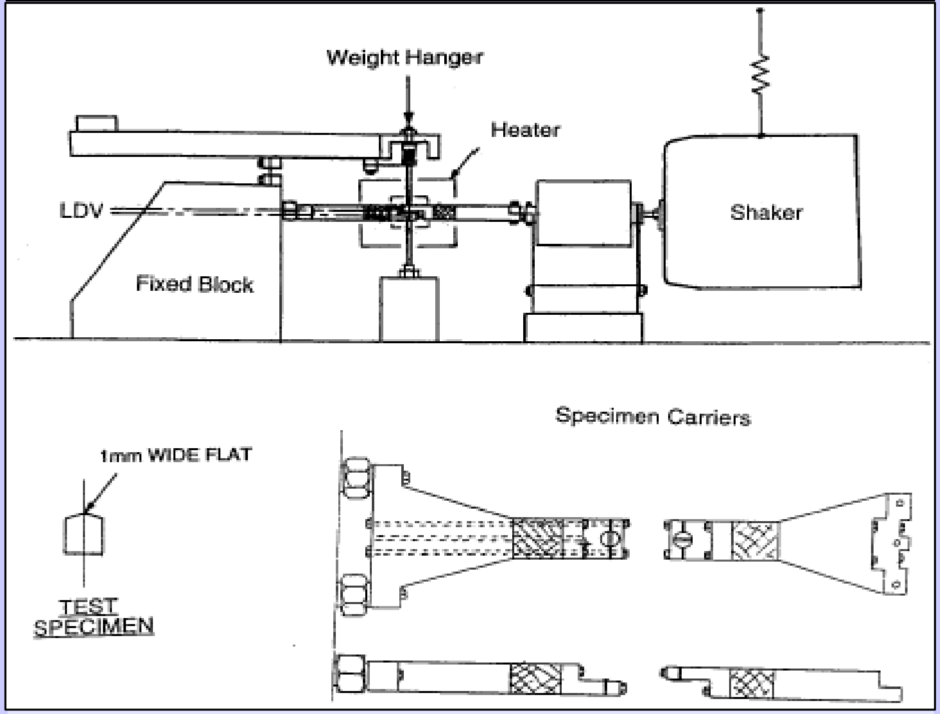
There is an extension to conform contacts (1990, Ciavarella, Farris, Hills&Nowell, …)

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

Contact mechanics - 4

The test rig at Imperial College 1999

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



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

Friction behaviour associated with fretting fatigue

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

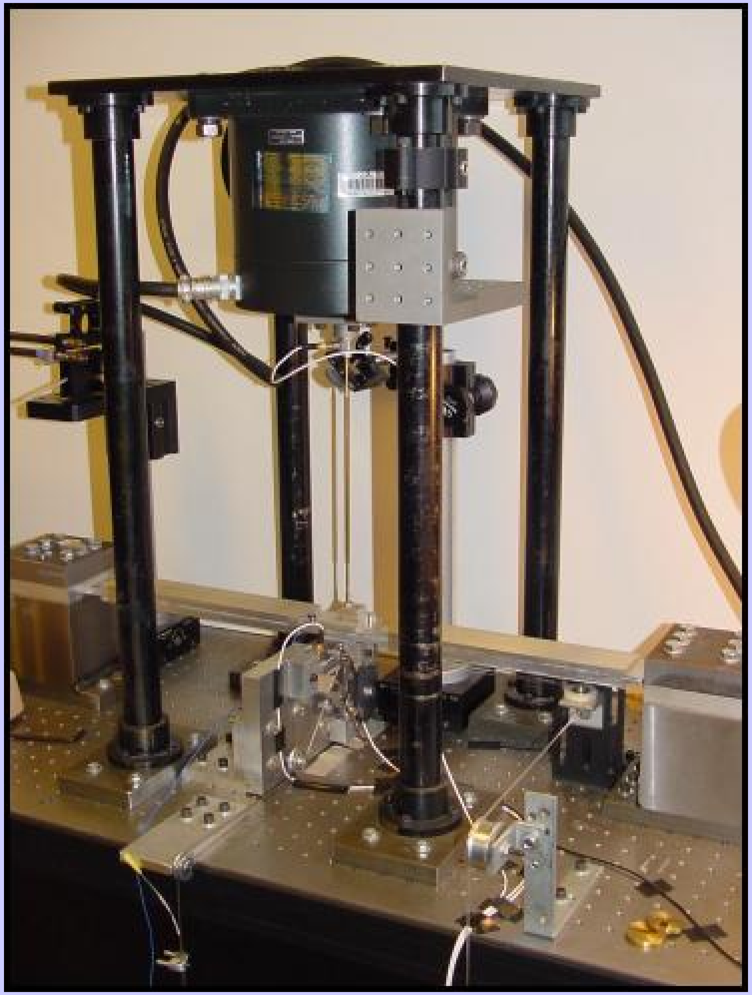
Tests at temperature up to 610°C in the contact region demonstrated that the friction coefficient increased with the wear of contact surfaces, and that the friction coefficient is dependent on the contact history.

Contact mechanics - 5

the Pittsburgh-Polito 1D test rig – room temperature

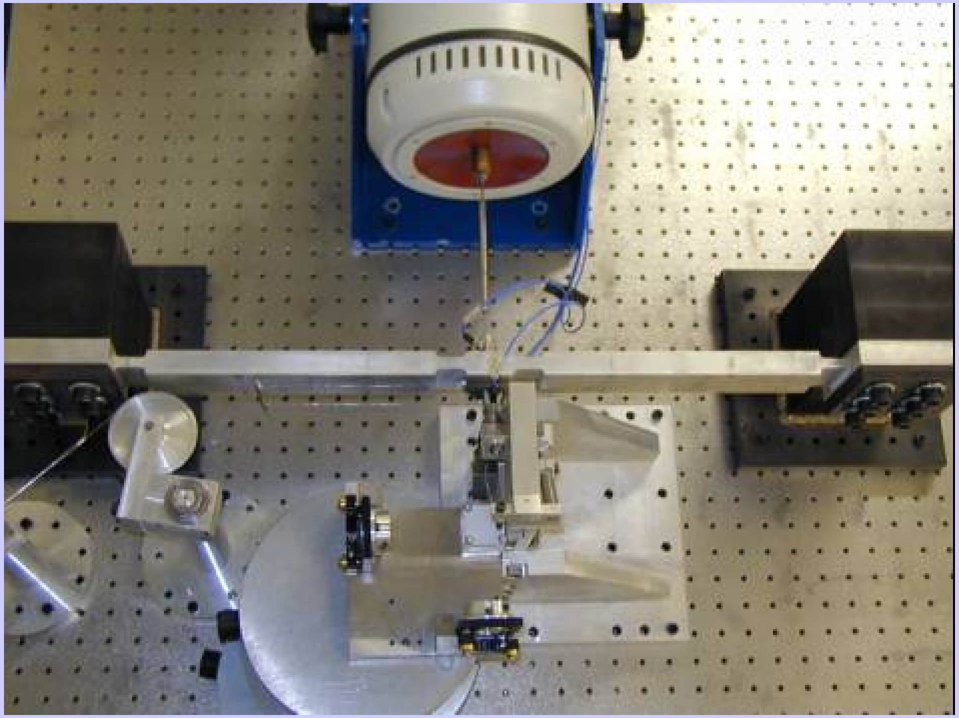
in co-operation with Prof. A. Akay

Carnegie Mellon University



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

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

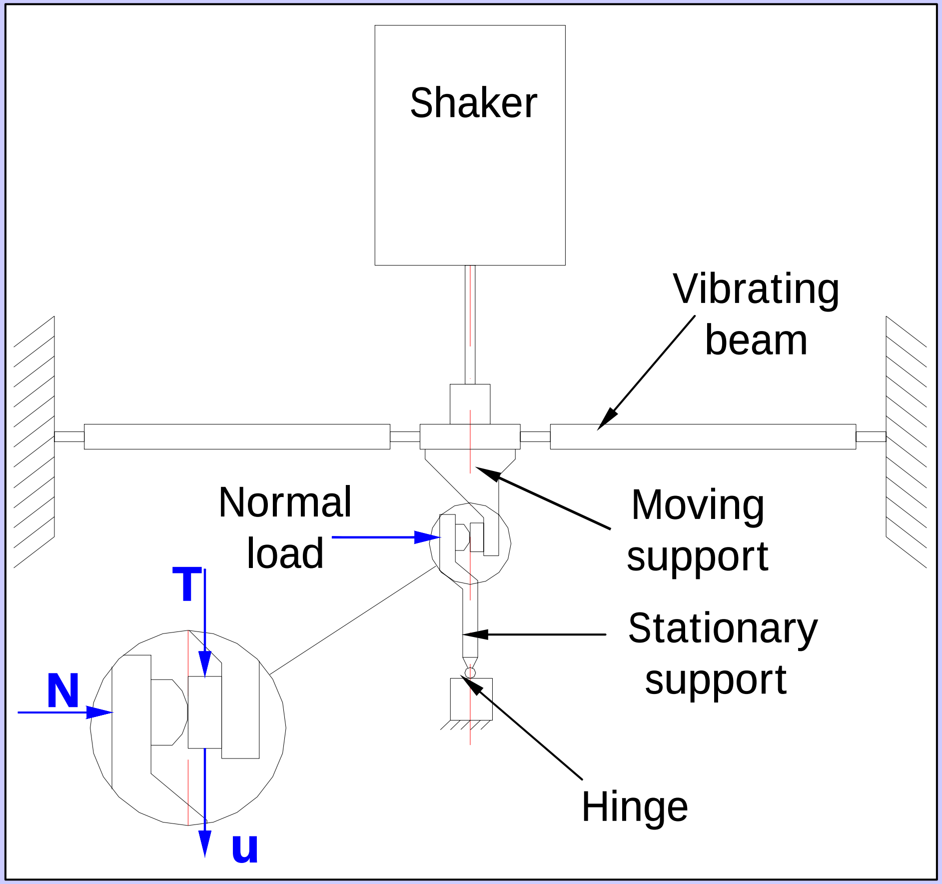


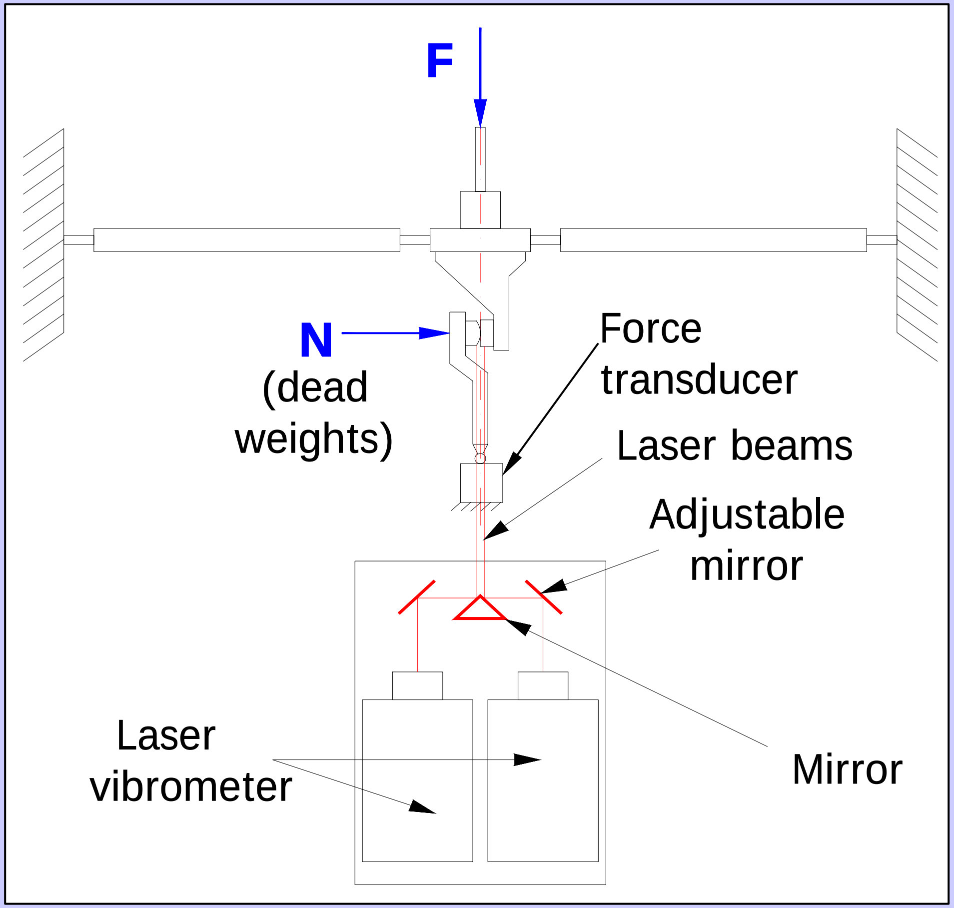
Contact mechanics - 6

the Pittsburgh-Polito 1D test rig – room temperature

Design Concepts

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





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

Contact mechanics - 7

the Pittsburgh-Polito 1D test rig – room temperature

Design Requirements at room temperature

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

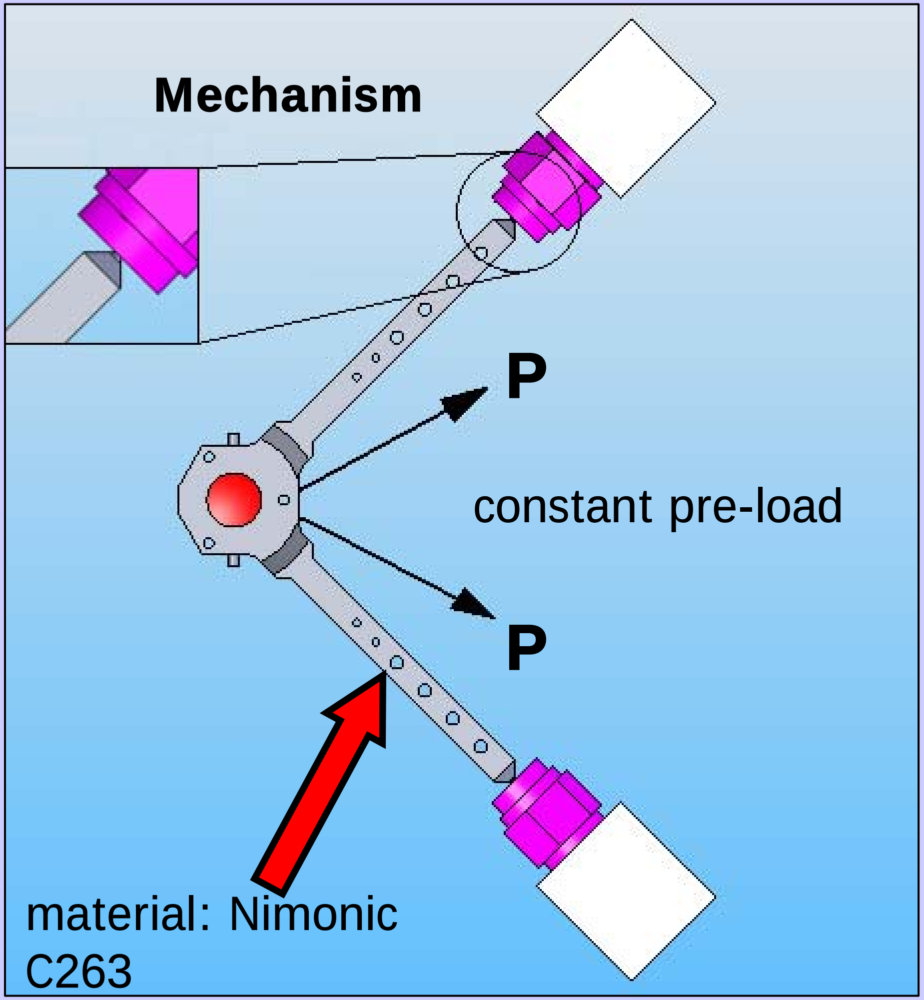
For high temperatures, there are additional requirements

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

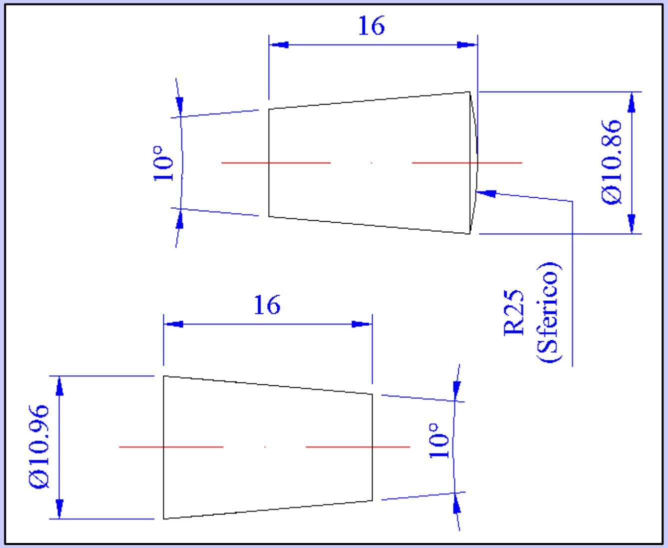
Contact mechanics - 8

the Pittsburgh-Polito 1D test rig – room temperature

Details of the rig



Conical Specimens



Contact mechanics - 9

data acquisition & control
high temperature

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

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

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

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

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

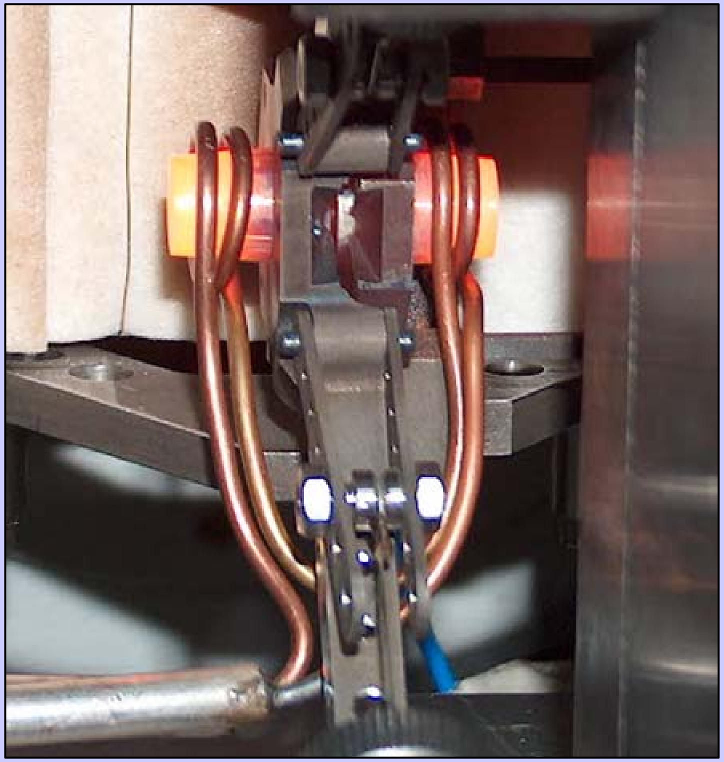
Contact mechanics - 10

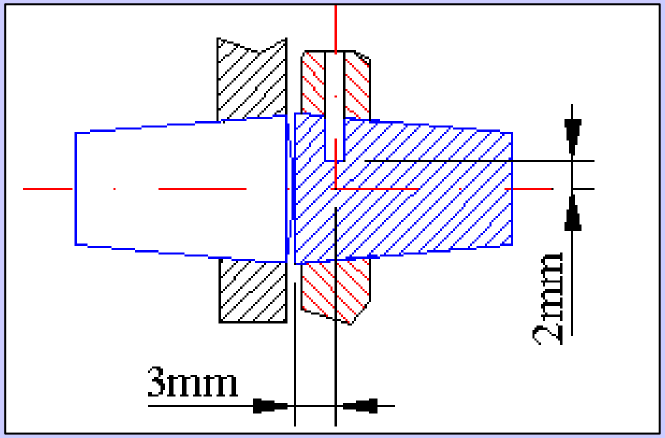
The heating system

- **<u>Electromagnetic induction system:</u>**
  - 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: **<u>k-type thermocouple</u>**, placed near the contact
- Temperature control: **<u>NI-card + Labview</u>**
- **<u>Error</u>** on temperature measurements: estimated by FEM thermal analysis

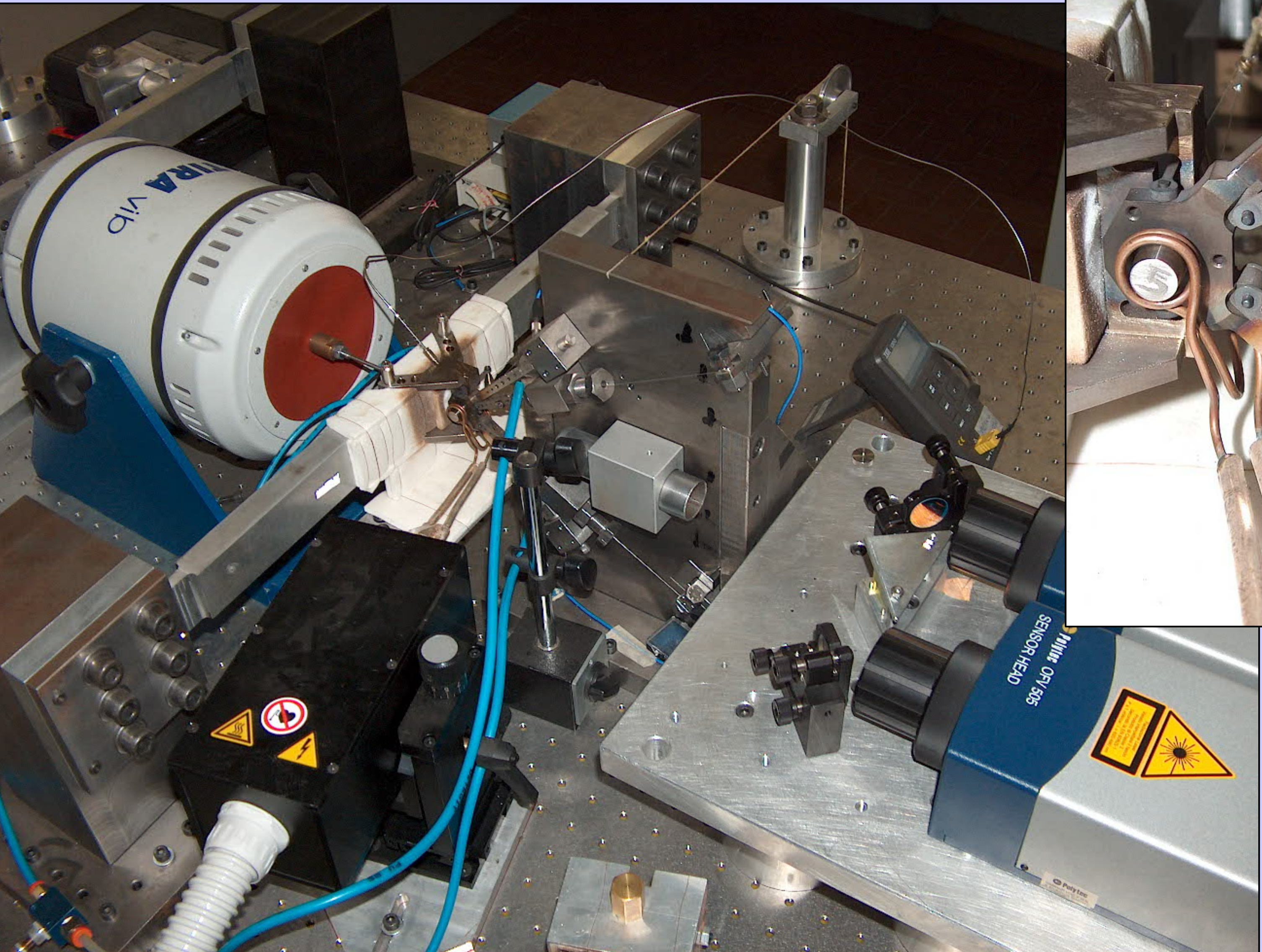
$$Temp_T = Temp_M \cdot (1 \pm 1{,}5\%) \pm 8°C$$

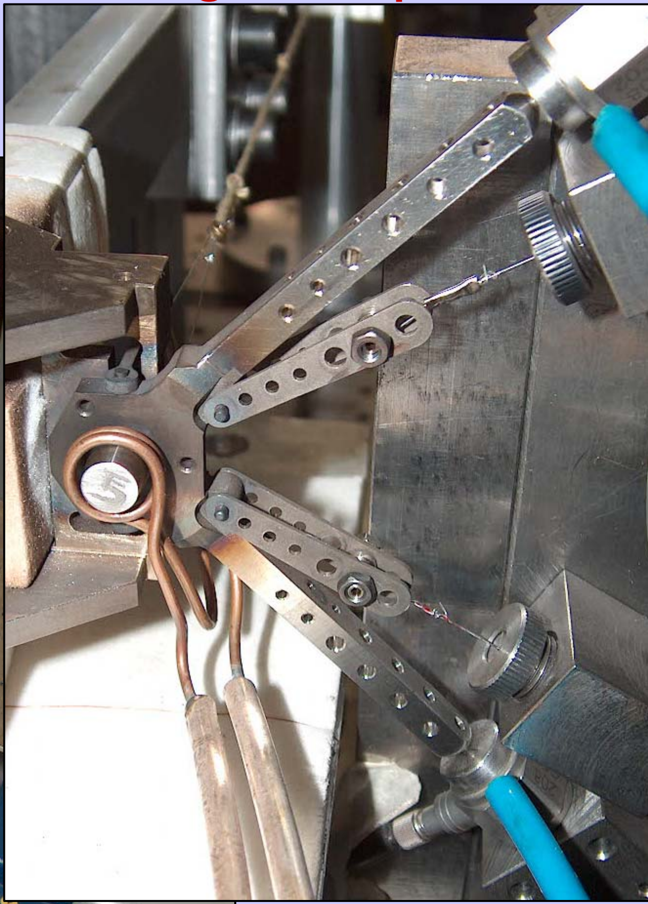




Contact mechanics - 11

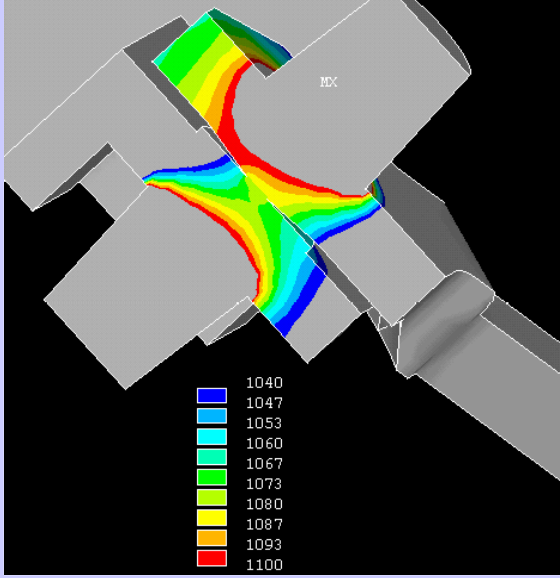
the Polito 1D test rig – high temperature

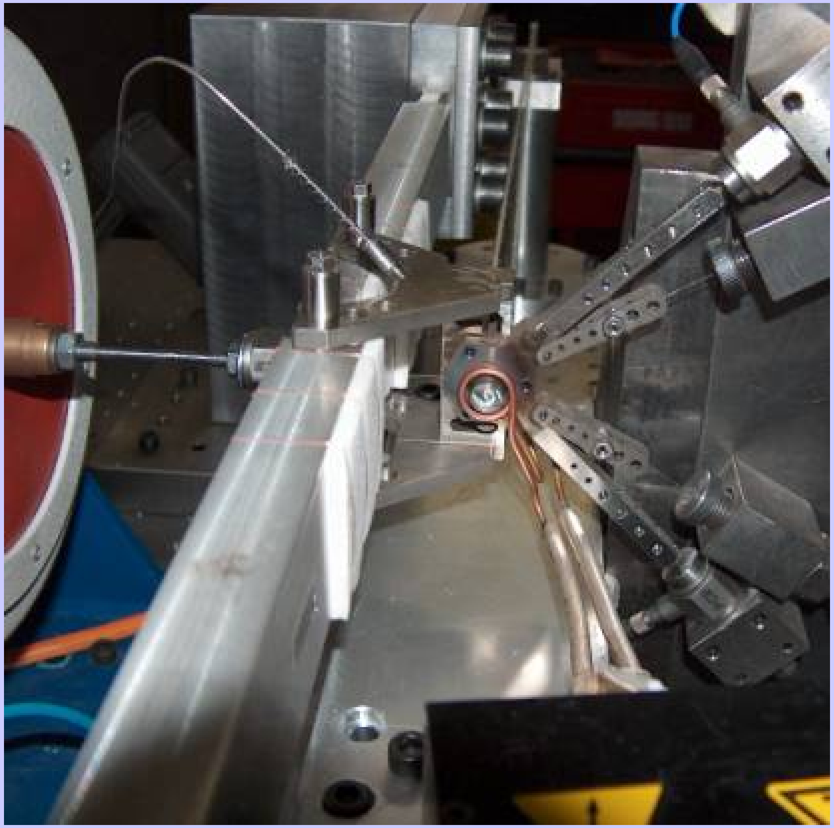


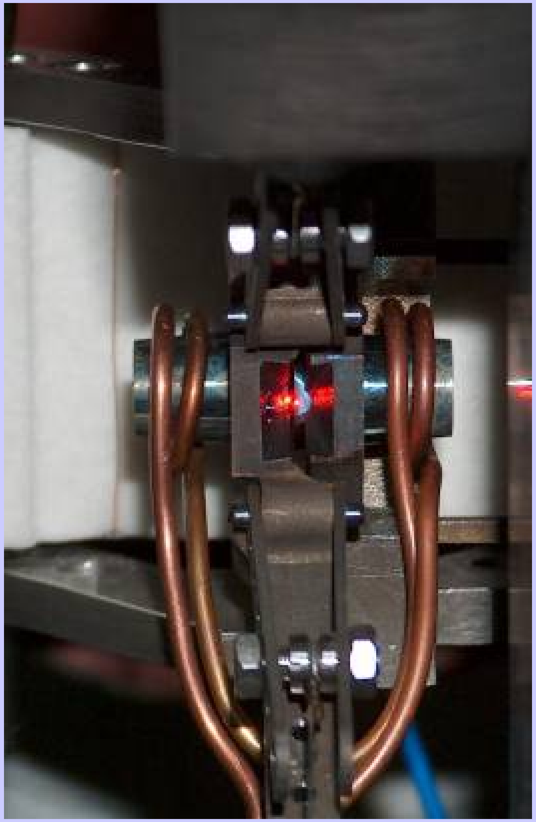


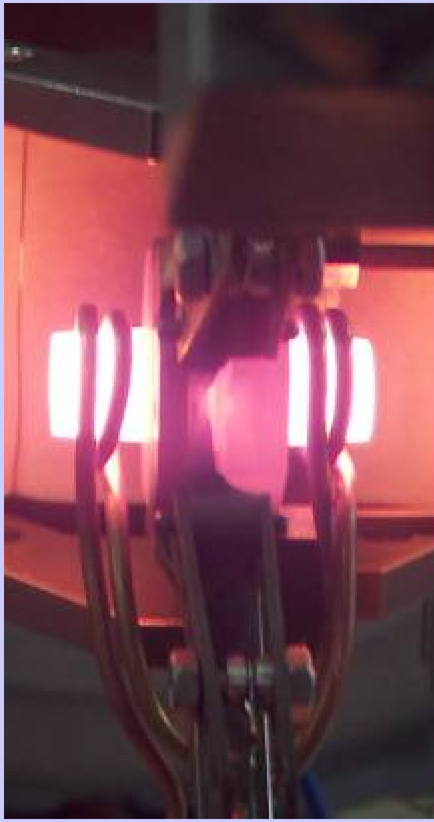
Contact mechanics - 12

the Polito 1D test rig
– high temperature









Contact mechanics - 13

the Polito 1D test rig – high temperature

Determination of contact parameters from hysteresis cycles

Dissipated energy:

⇒ It is the area of the hysteresis cycle.

Contact Stiffness:

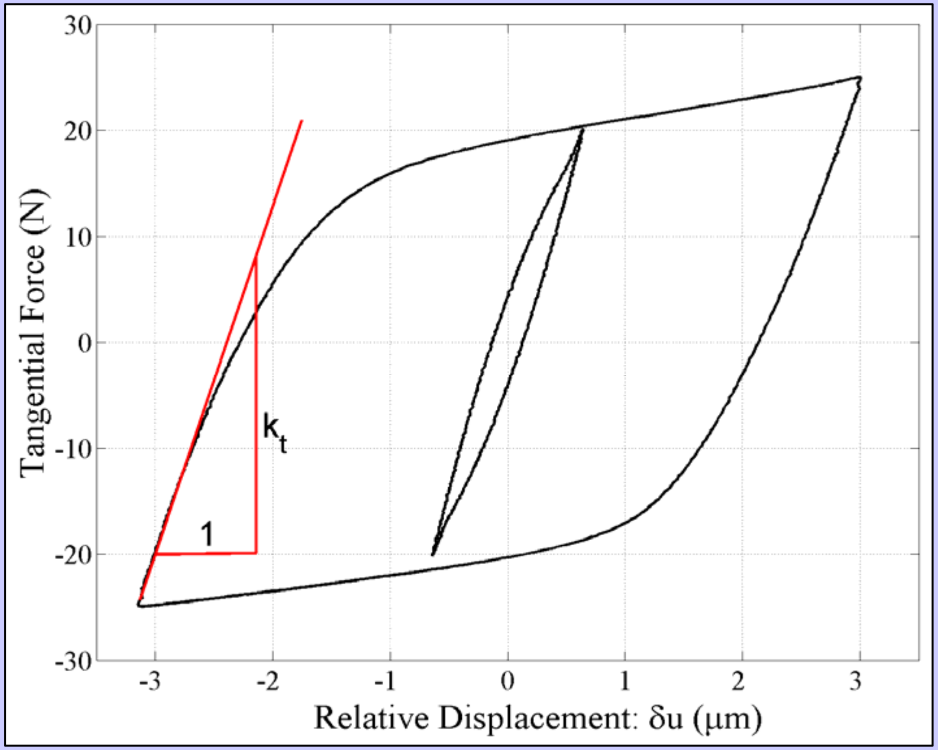
⇒ Slope of the curve after reversal of motion

Friction Coefficient:

⇒ Since it varies in gross-slip phase, it is calculated with Mindlin's theory, in terms of dissipated energy, normal loading and contact stiffness.

⇒ Calculation gives the "average" value.

⇒ Only for gross-slip cycles



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

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

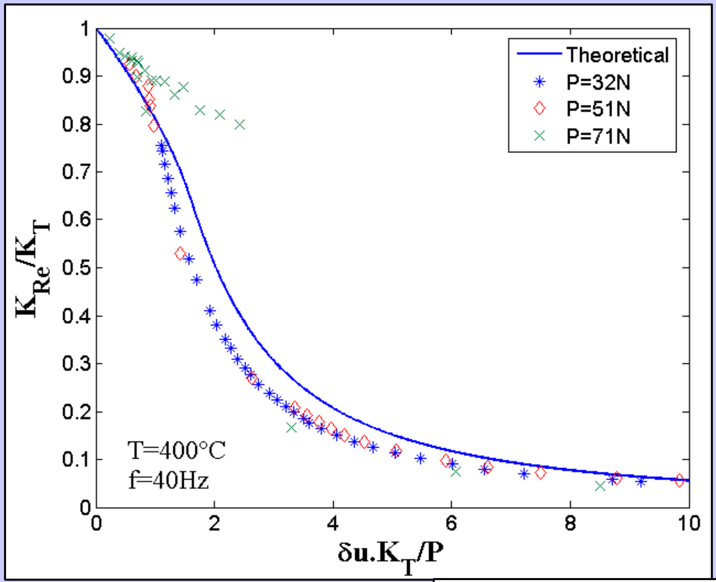
HBM real and imaginary parts

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

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

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

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

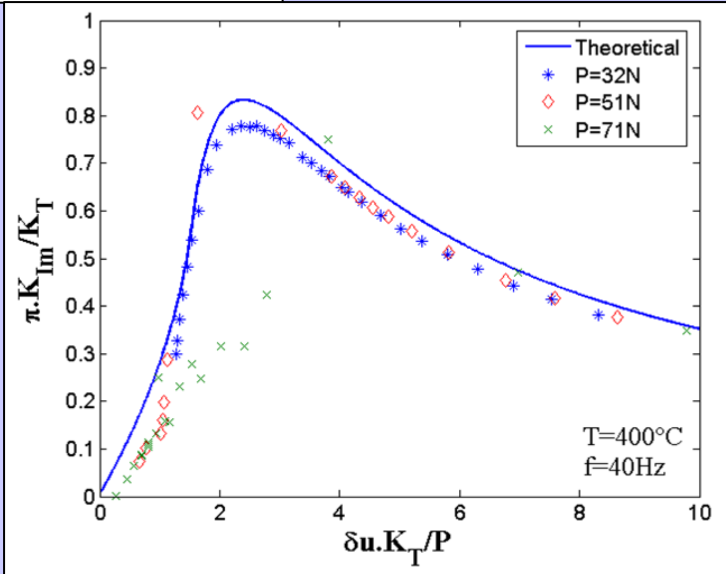


Dimensionless equiv. contact stiffness:

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

Hysteretic effective contact damping:

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



Contact mechanics - 15

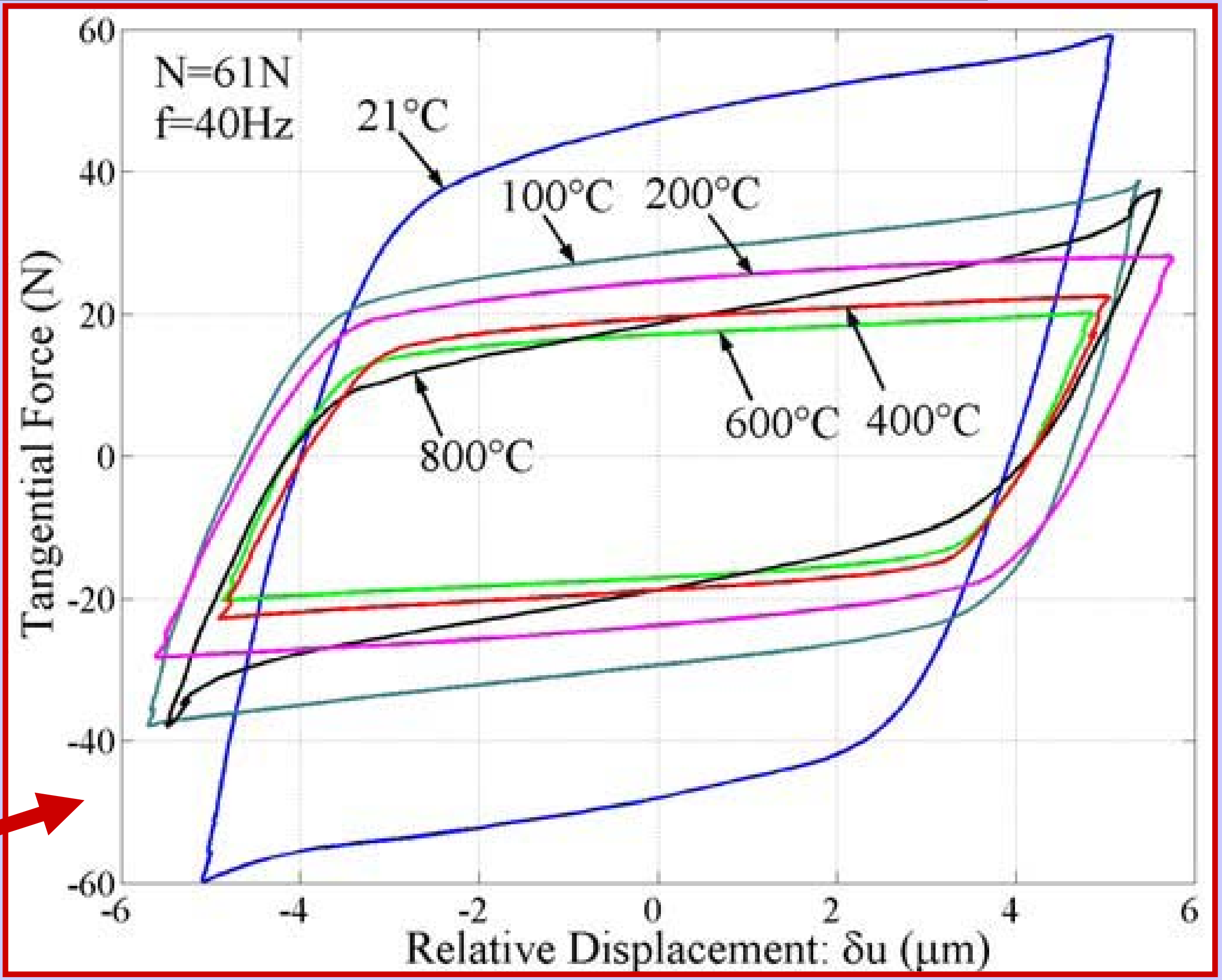
hysteresis cycles
at high temperature

First set of experiments: specimens of Inconel 100

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

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

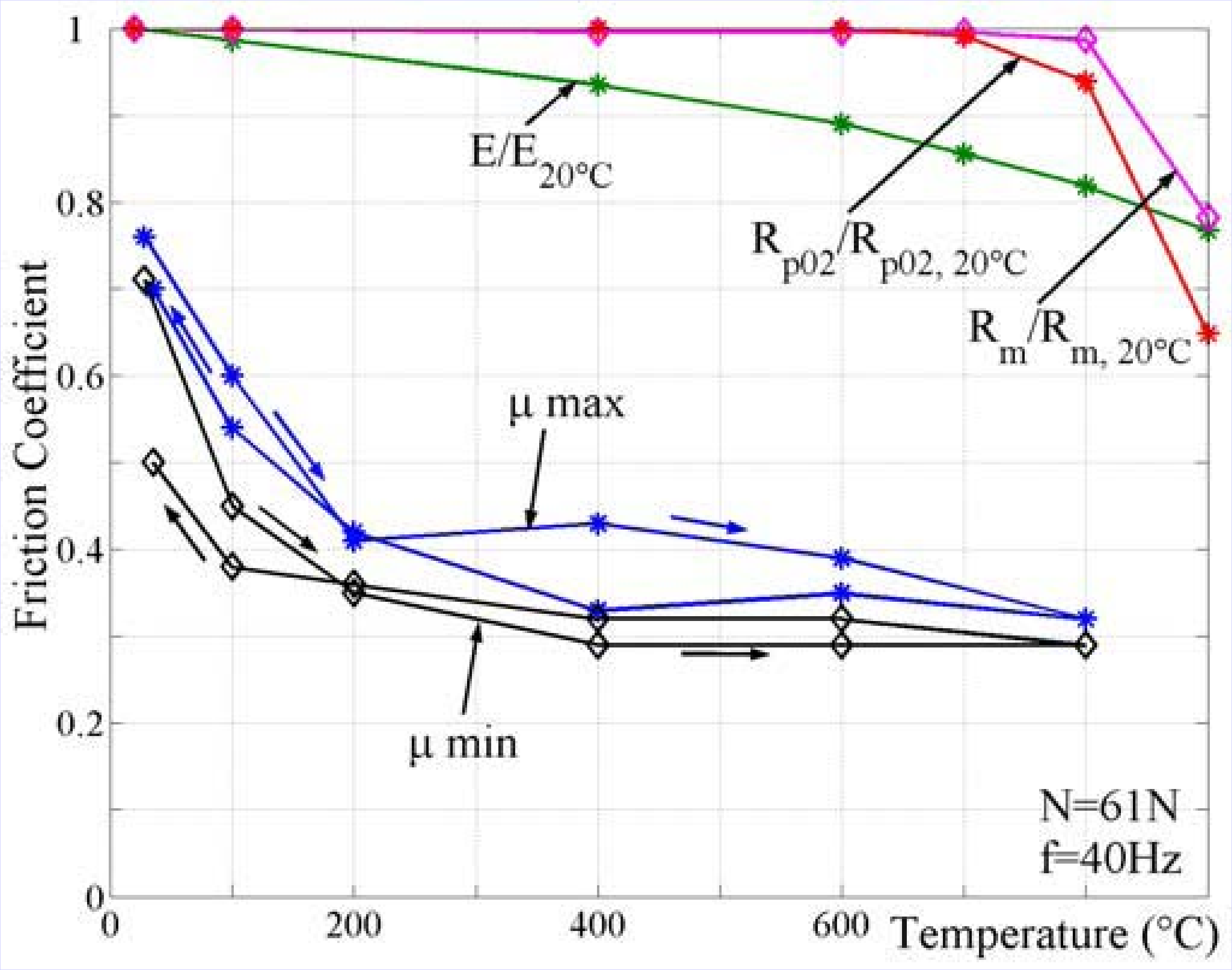
Temperature Range:
T=20°C up to 800°C



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

Contact mechanics - 16

hysteresis cycles at high temperature



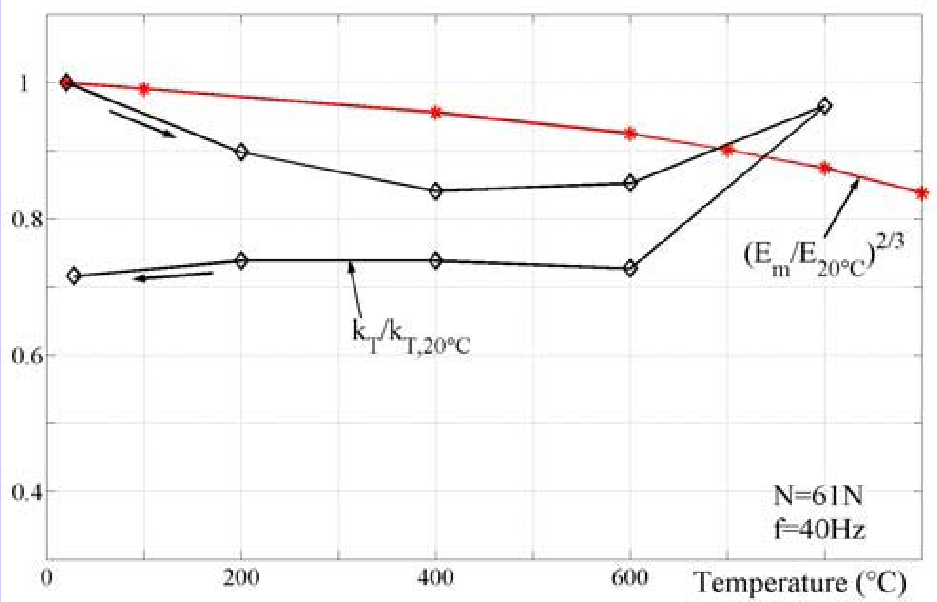
Friction coefficient & material parameters vs. temperature (Normal load: 61N)

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

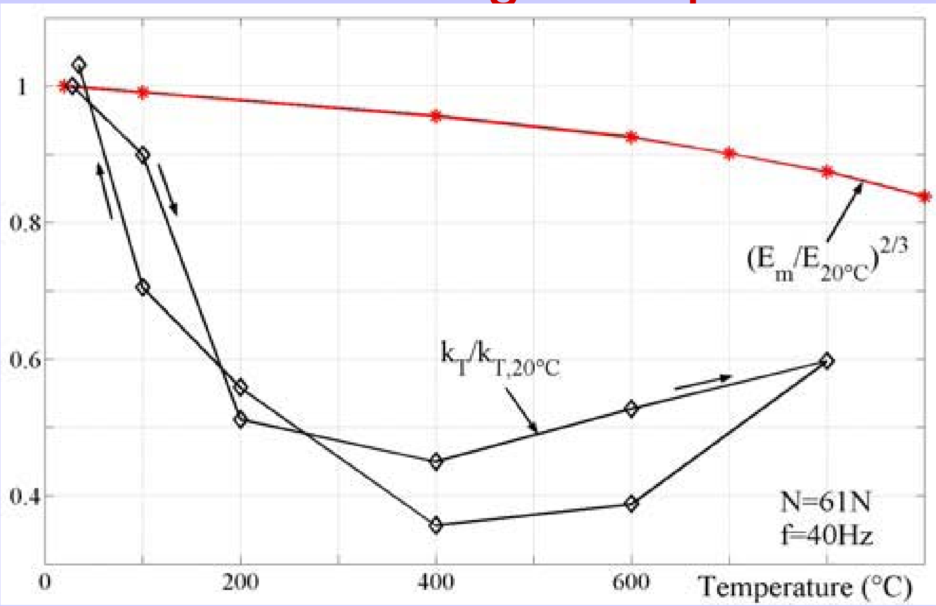
While the friction coefficient decreases rather sharply from room temperature up to 200°C and then becomes almost stable, yield and ultimate strength are practically constant up to almost 800°C and the Young modulus decreases almost linearly with temperature in the same temperature range. Therefore no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

Contact mechanics - 18

hysteresis cycles at high temperature



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



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

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

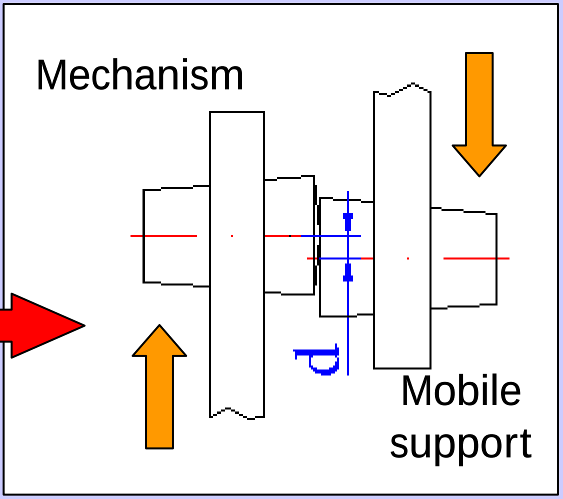
Contact mechanics - 17

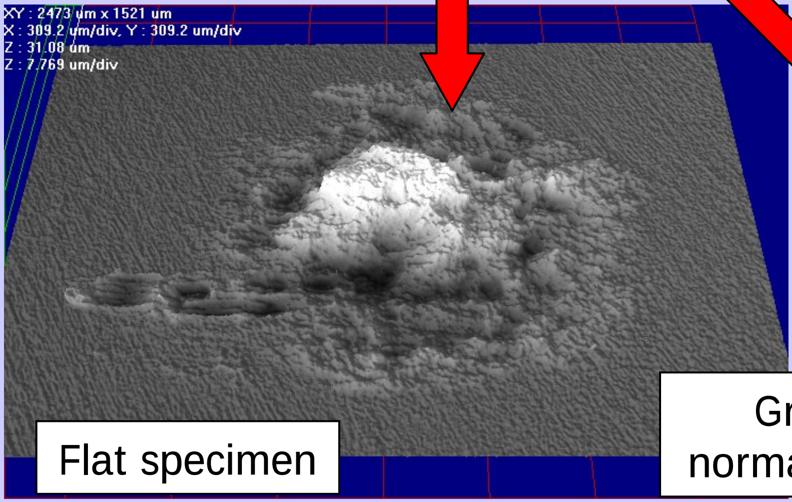
problems & developments

More problems found in the Test Procedure

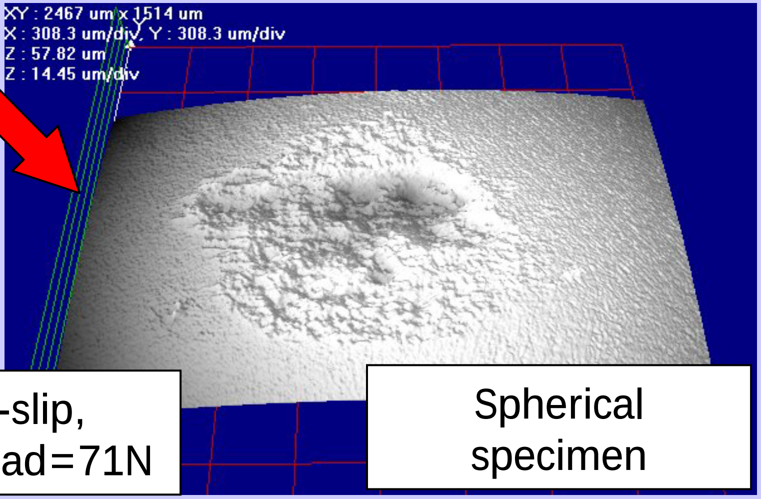
- Thermal expansions => contact point moves with temperature
  - Expansion of the mechanism
  - Expansion of the mobile support
  - Bending of the vibrating beam ???
- Wear changes the contact properties

20-800°C: d>1mm



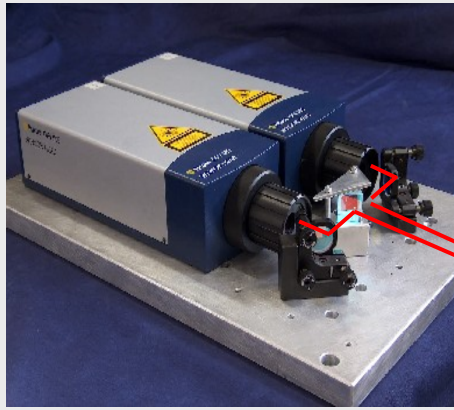


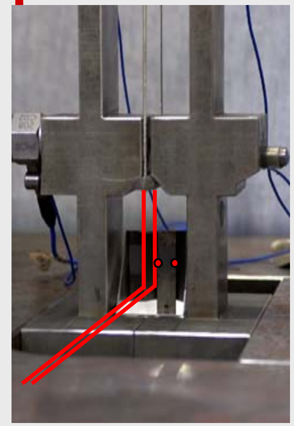
Gross-slip, normal load=71N

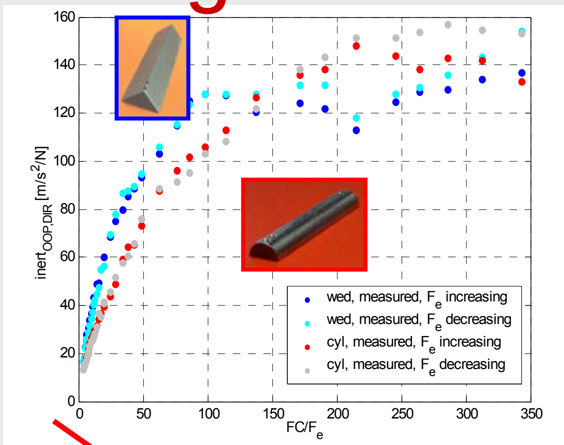


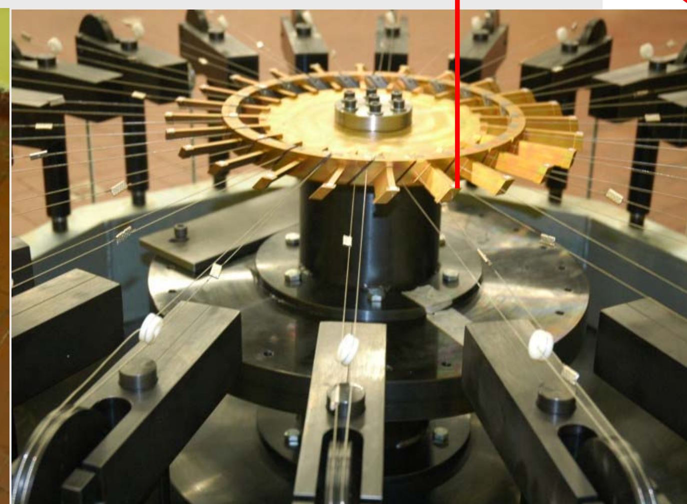
Materials under test now (2008/2009 – AVIO restricted access to data): RENE 77, 80, 108, 125; CMSX-4; Inconel 718 – with and without T800 coating (high roughness and hardness)

Modelling damping components -1
underplatform dampers – test rigs

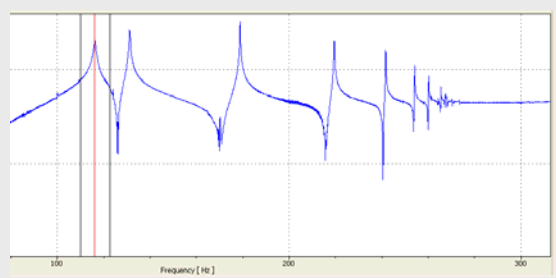


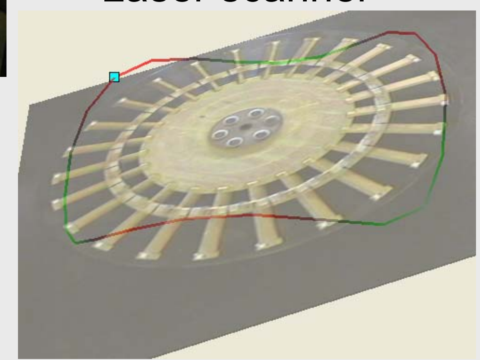






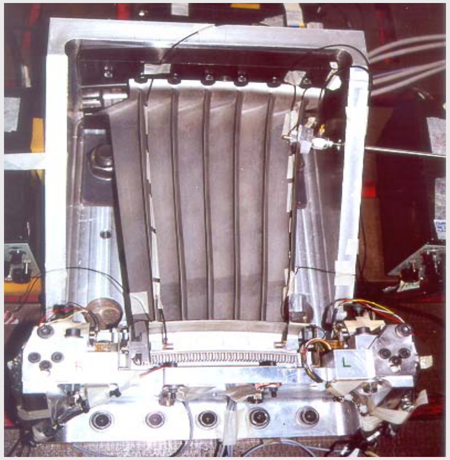
Laser scanner

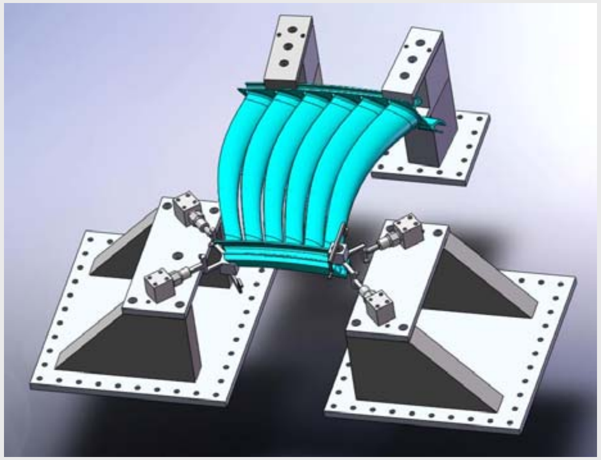


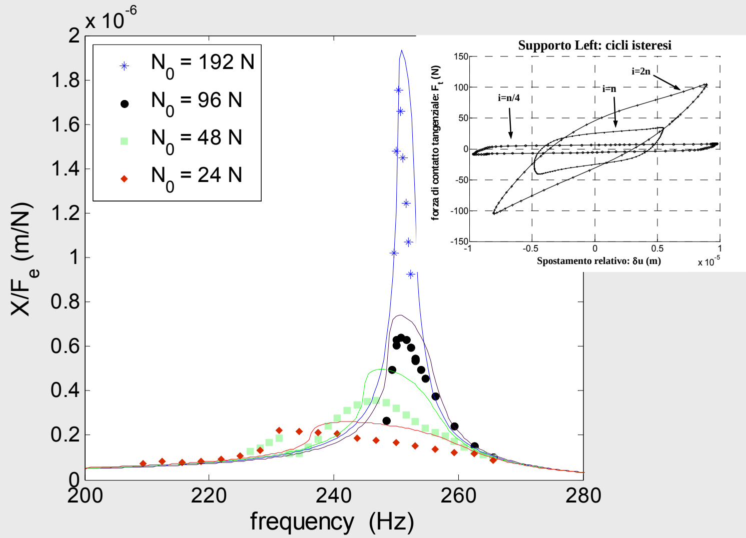


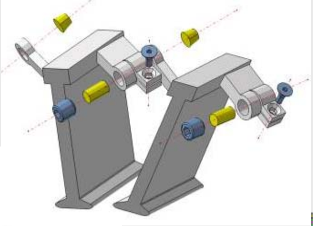
Non contact electromagnetic excitation

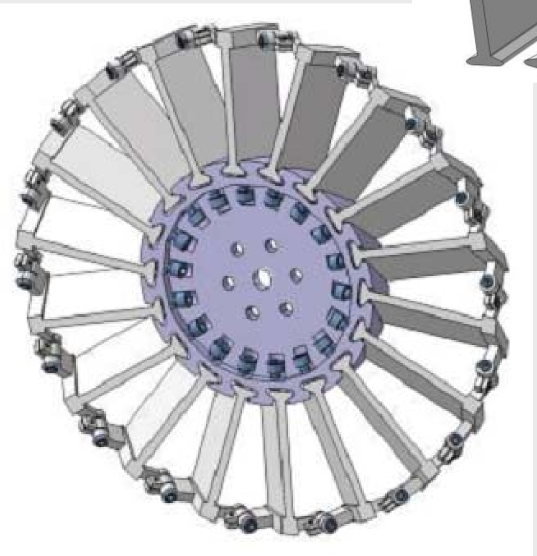
Modelling damping components -2
vane segments and shrouded blades – test rigs

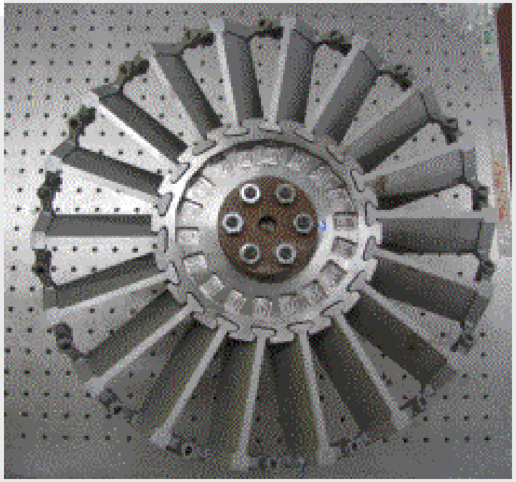




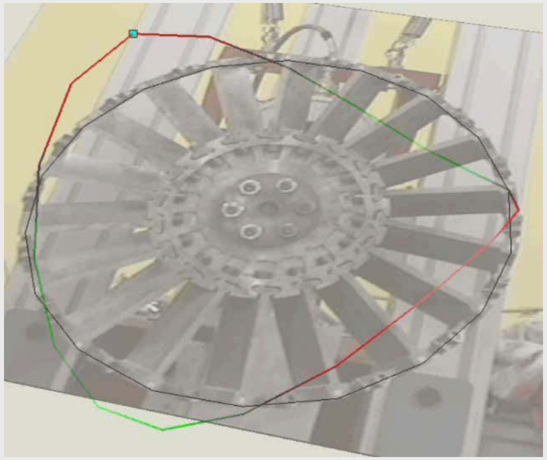








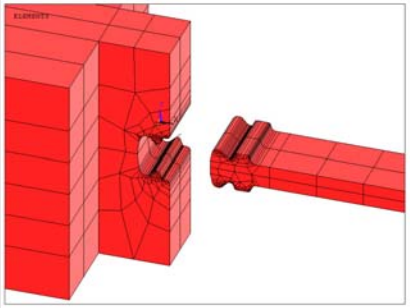
Laser scanner measurement

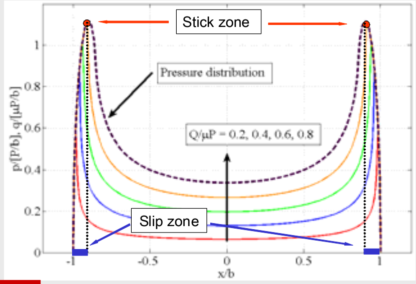


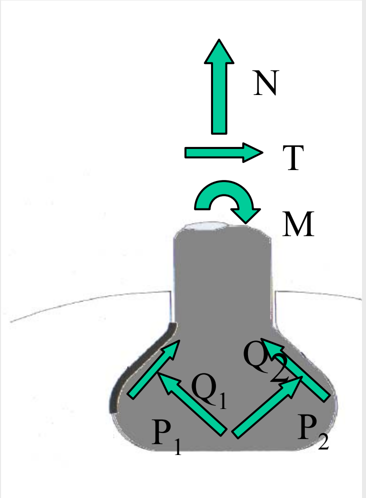
Modelling damping components -3
blade root damping characterization

The Test Rig

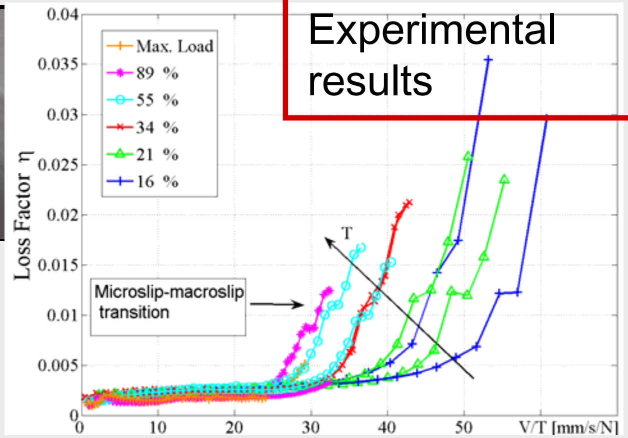
Numerical Contact model based on contact mechanics principles



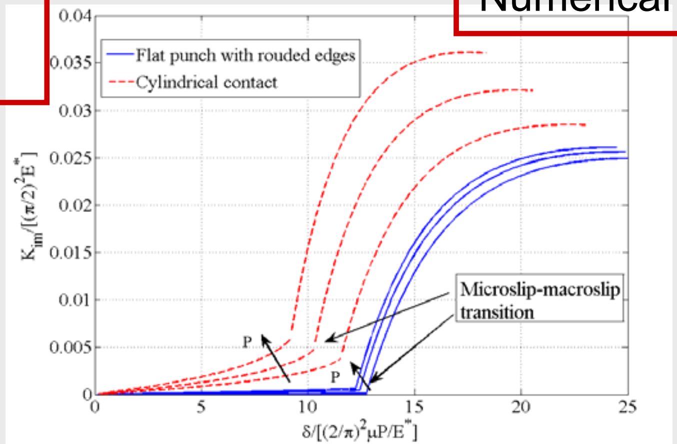




Experimental results

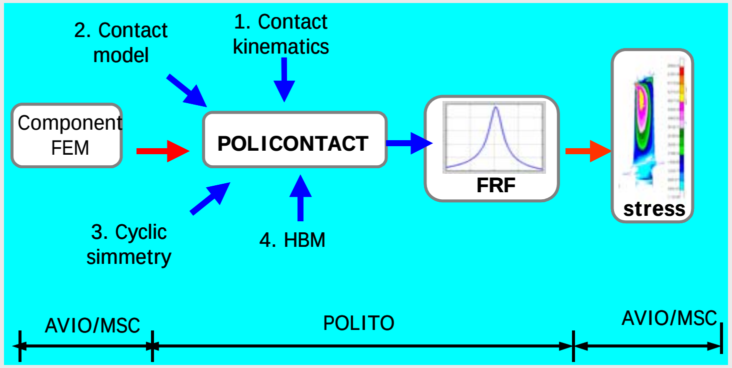


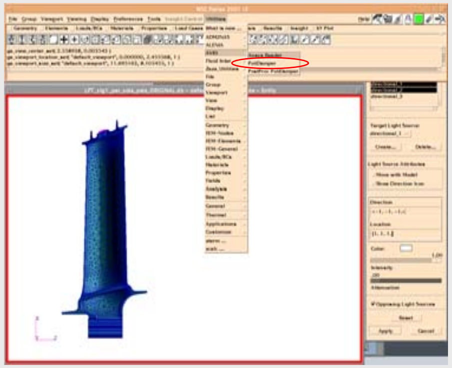
Numerical results



Dynamic response of turbine discs - 1

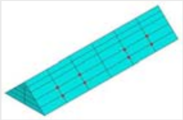
Non linear forced response calculation with friction damping
POLI.contact software

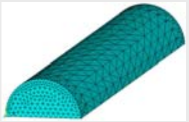




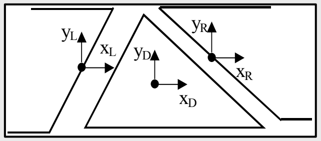
POLI.CONTACT in MSC.Patran and MSC.Nastran

underplatform damper



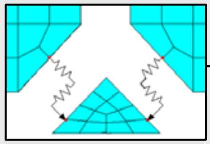


Rigid damper



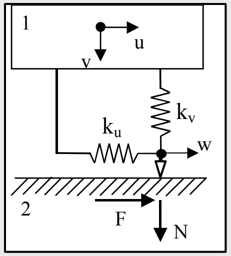
Kinematic assumptions

Elastic damper



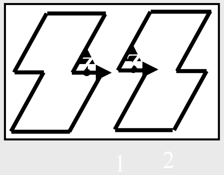
No kinematic assumptions

contact Model



shrouded blades and vane segments

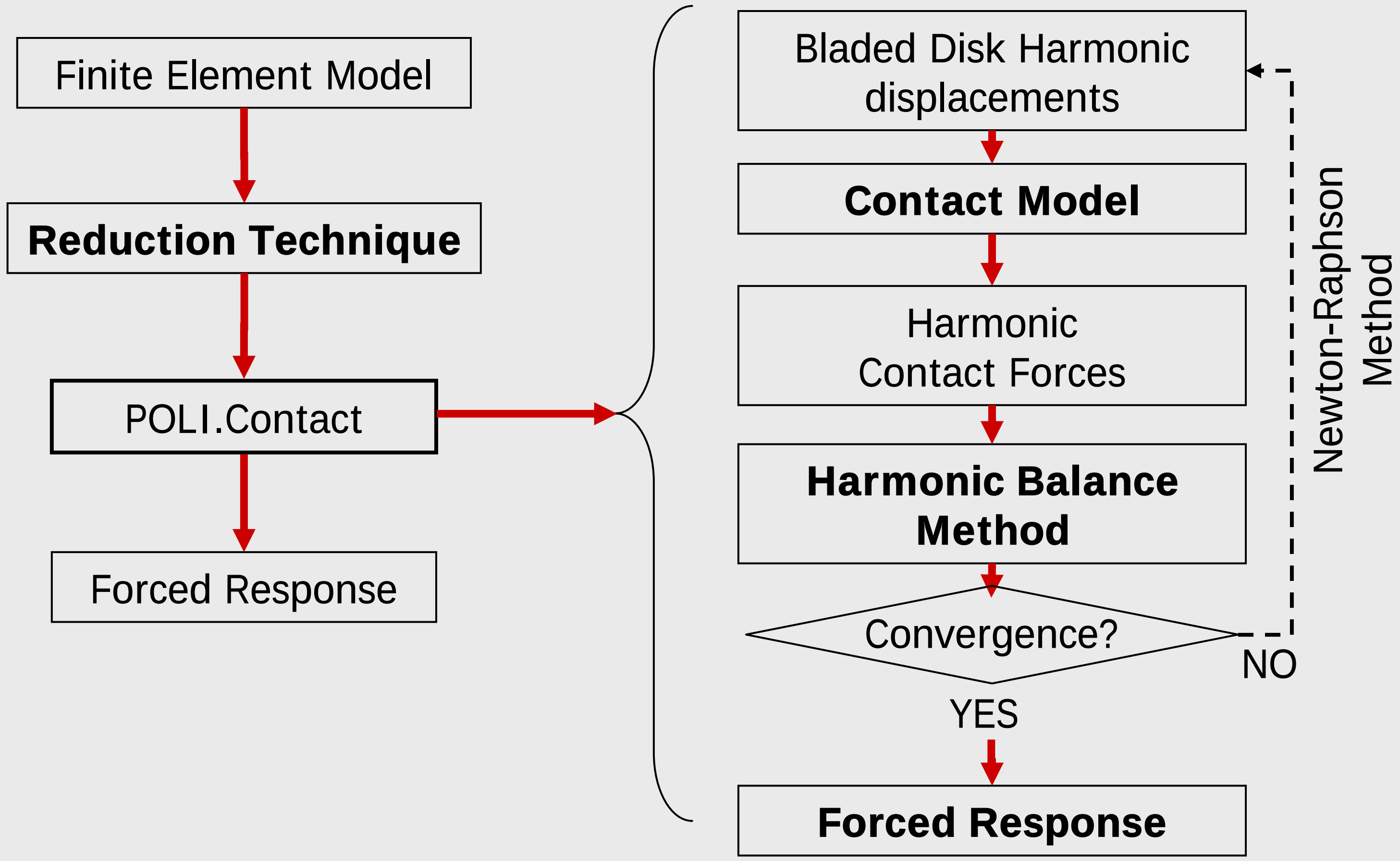
Direct blade-to-blade contact → simple contact kinematics



contact model

Dynamic response of turbine discs - 2

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



Dynamic response of turbine discs - 3

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

Reduction Matrix

N Physical dofs ← $\{X\} = [\Psi] \cdot \{q\};$ → **n** generalized dofs

$N >> n$

→

Reduced model

$$[K_R] = [\Psi]^T \cdot [K] \cdot [\Psi]$$

$$[M_R] = [\Psi]^T \cdot [M] \cdot [\Psi]$$

$$\{F_R\} = [\Psi]^T \cdot \{F\}$$

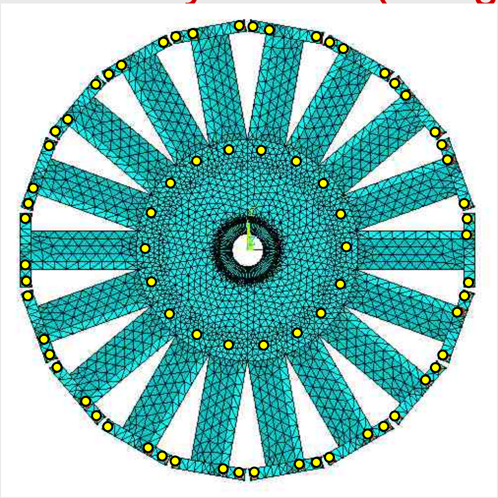
- Component Mode Synthesis (Craig-Bampton).

Physical dofs of master nodes

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

Modal dofs of slave nodes

=



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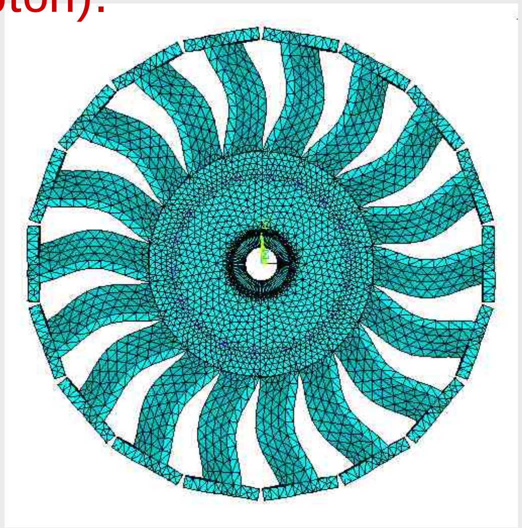
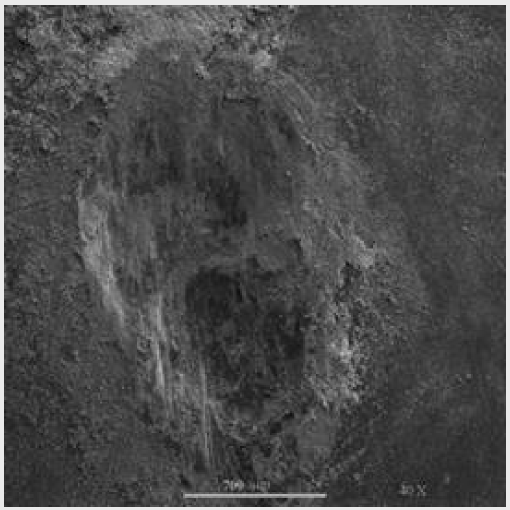


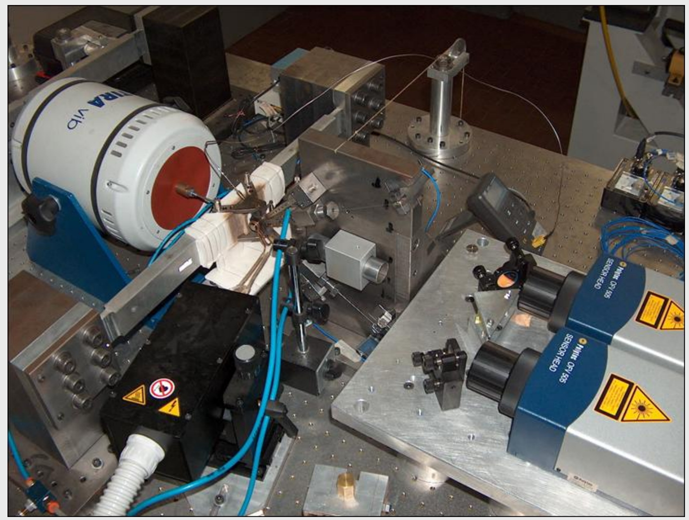
Complementary activities

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

Tribology, wear measurement

Measurement of wear on contact surfaces
Validation of theoretical wear models



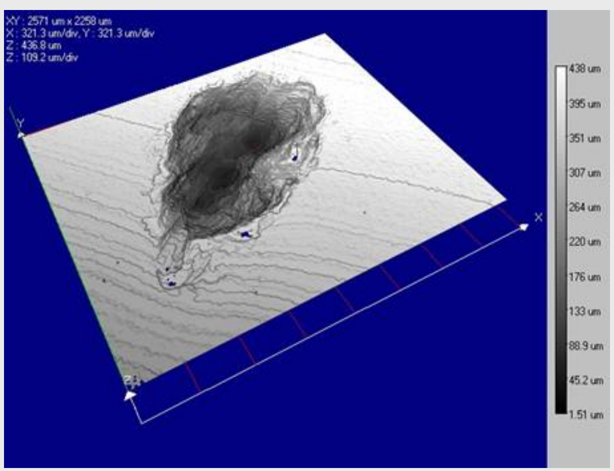


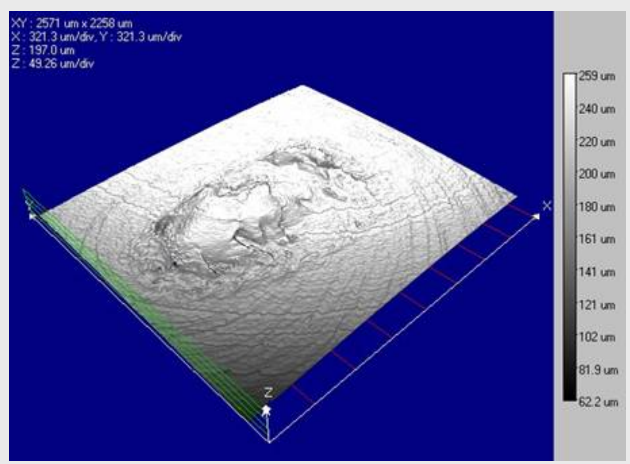
Point contact test

$10 \cdot 10^6$ cycles

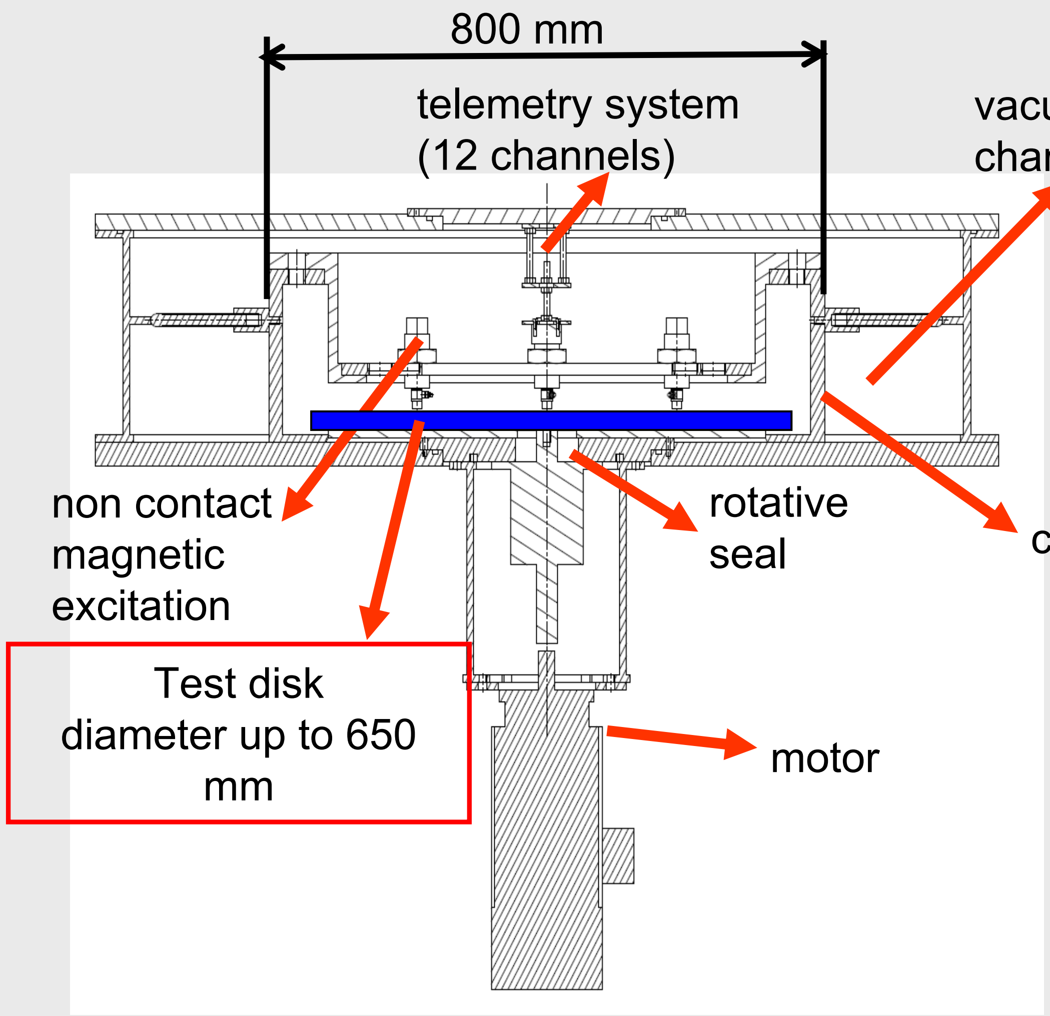
amplitude 30 μm

T = 900 °C

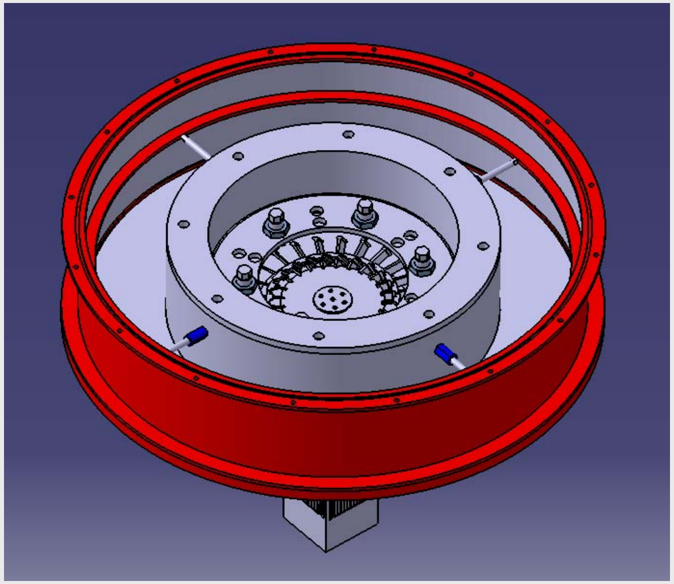




Spin-test rig (work in progress)

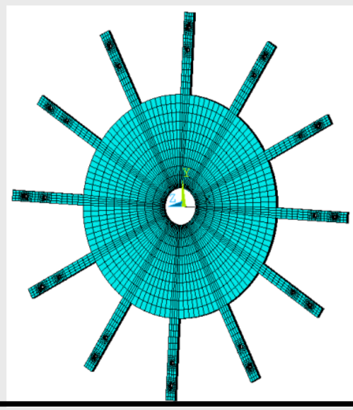


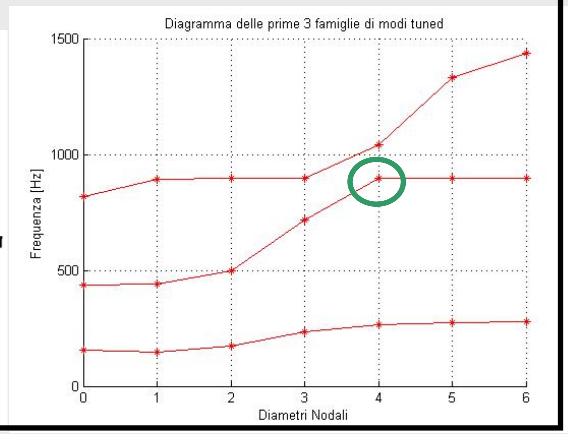
Rotation speed
up to 4000 rpm



Dynamics of turbine disks with asymmetrical effects (MISTUNING)

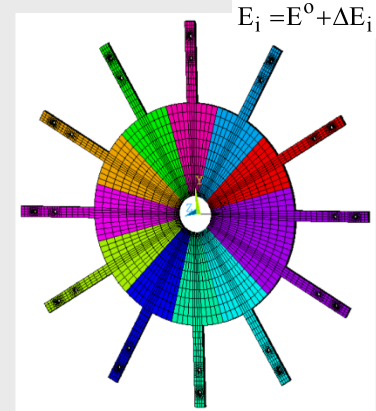
TUNED BLISK

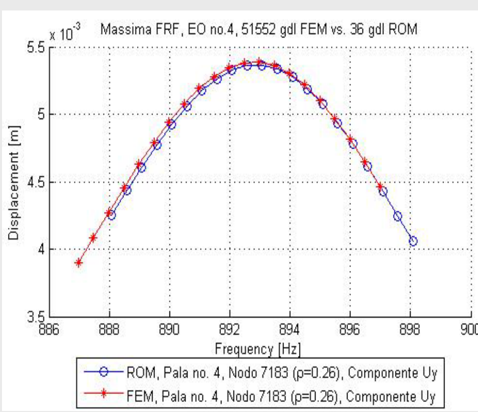




MISTUNED BLISK

$E_i = E^o + \Delta E_i$



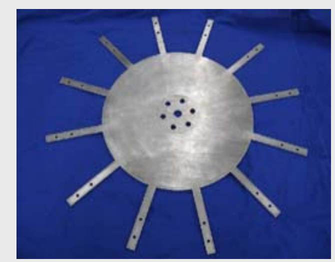


Magnification Factor: 1,42

Identification model of mistunnig

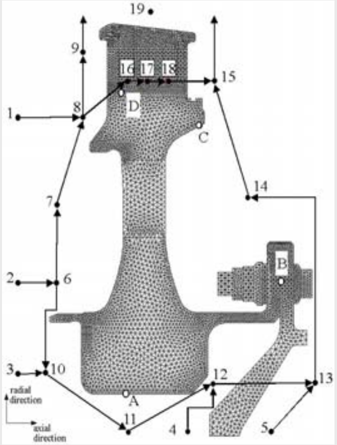
Comparison of different reduction techniques and improvement

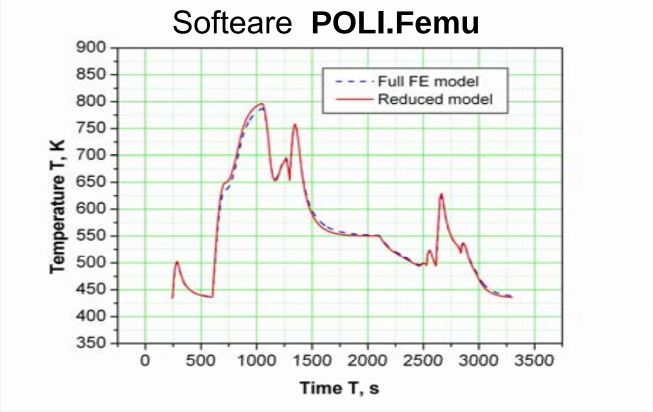
Experimemtal validation by means of dummy disks

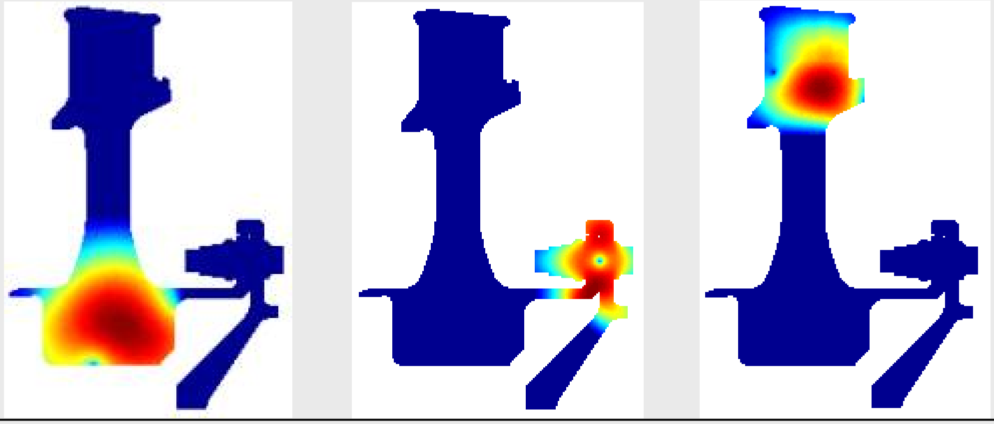


Real time evaluation of temperature and thermal stresses at critical locations of turbine disc (disc lifing)

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