Status of NIKHEF activities on crystalline suspensions

Eric Hennes

A.Bertolini, M.Doets, M. Jaspers, F. Schimmel, J. Geurtsen

ICRR: T. Sekiguchi,
Univ. of Sannio: R. DeSalvo

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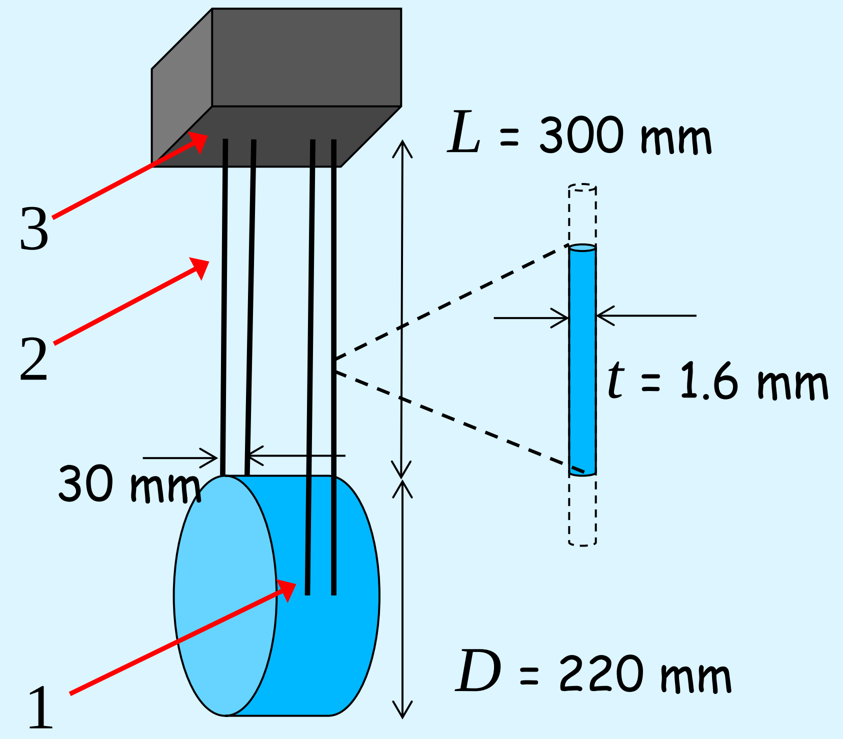
bKagra test mass suspension

Mirror and wires: both sapphire

To be addressed

- wire bonding
- wire tension equilibration
- vertical modes f_y= 100 Hz, f_{pitch} =23 Hz
  too high? (adVirgo: 10 times lower)
- pendulum dilution factor: 7.5
  too much thermal noise? (adLigo: >1000)
- wire profile



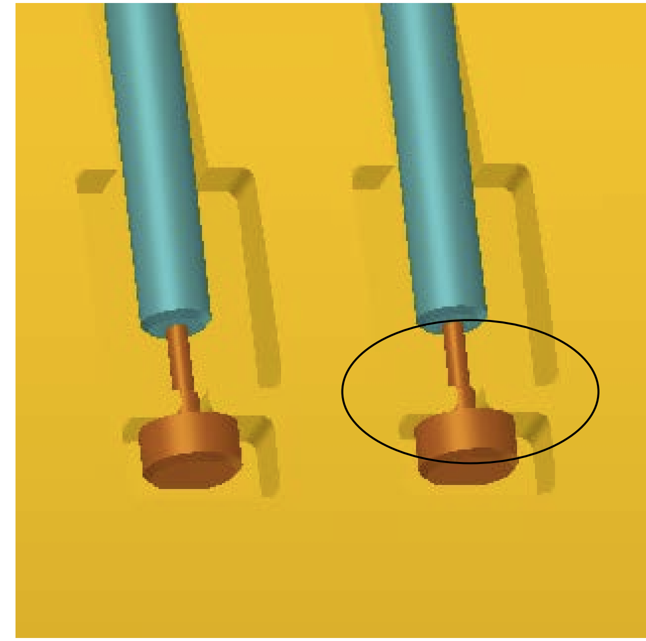
Investigated options (also valuable for ET)

1. Attach wire to mirror: suspension slots
2. Improve heat transfer & *z*-compliance: flat ribbon / segmented rod*
3. Increase vertical compliance with springs
   - Cantilever blade suspension
   - Silicon blade bending strength tests
   - Compressive beam suspension

*Non-uniform rod concept: Takanori Sekiguchi Elites f2f meeting, Tokyo, 2012/2/4

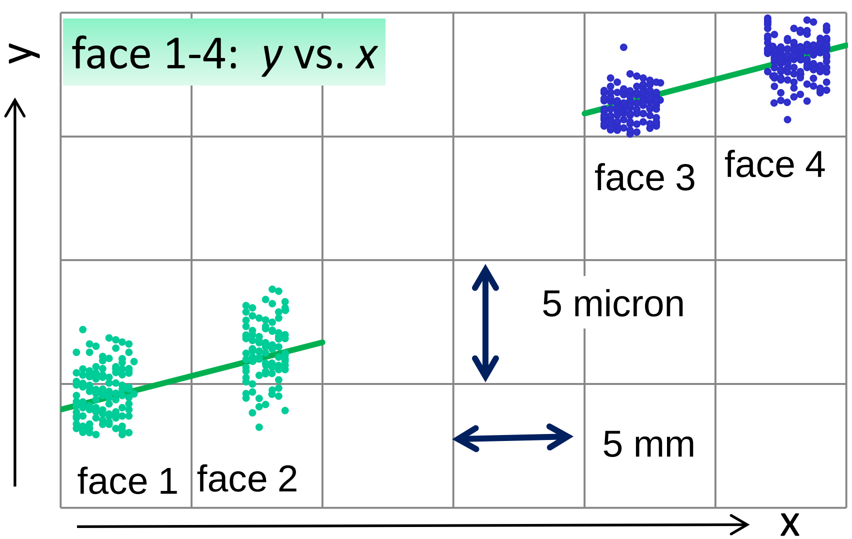
1. mirror-wire bonding concept

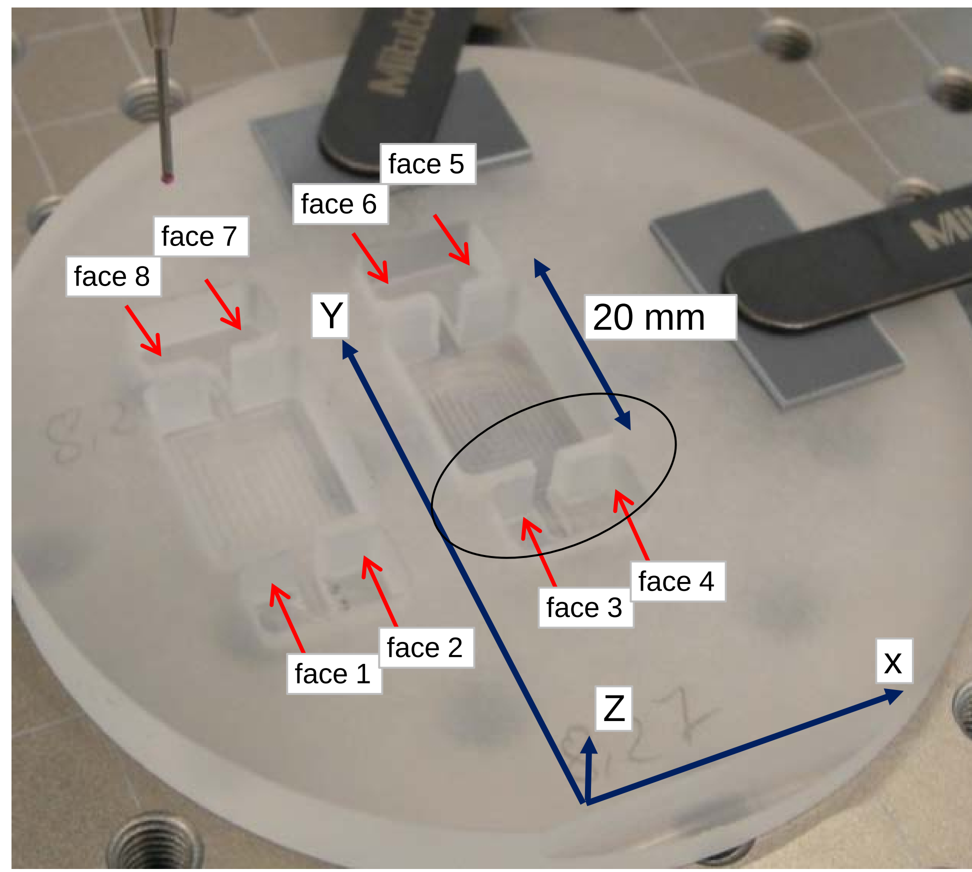
- Wire/rod with nail head attached
- Slot alcove machined in the mirror itself
- Contacting faces sufficiently aligned and flat
- Thin indium/gallium foil for thermal contact
- Purely compressive joint



Prototype sapphire slot:

- Made by CNC Mack (South-Germany)
- Ultra-sonically machined faces
- Face pairs aligned and flat σ_y= 0.8 μm

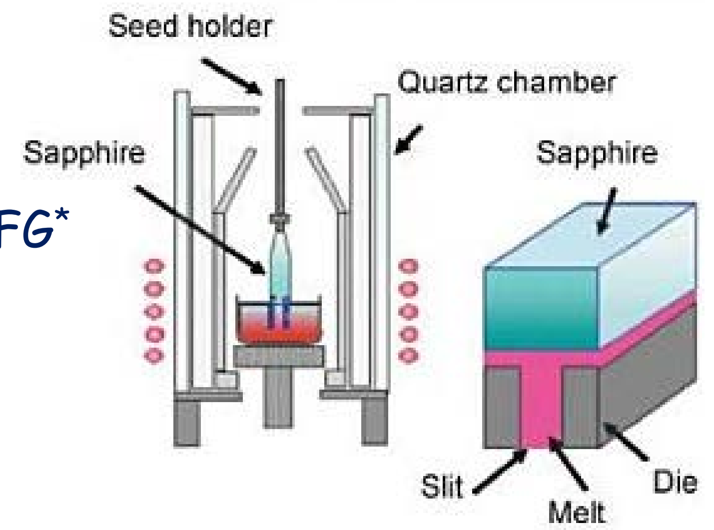


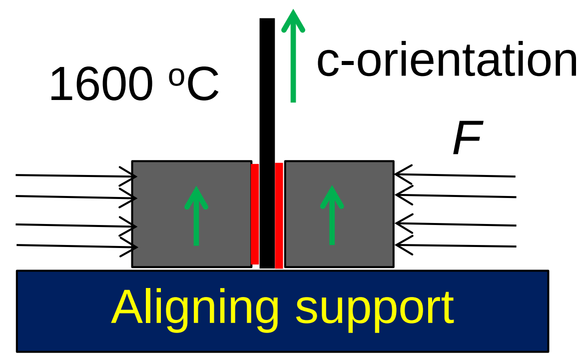


2. Flat Ribbon profile concept

- Mono-crystal grown in designed profile (Stepanov-EFG* method, tubes/ribbons up to 300 mm, 1-4 cm/hour)
- Low compliance in beam direction
- Polish flat sides → increase thermal conductivity

  (preventing diffusive reflection of phonons)
- 2 half nail heads, each 3 ultrasonically machined faces
- bonding to ribbon options: mono-crystalline braze by heat-pressing with sputtered microcrystals, or hydroxide catalysis bonding or ……?



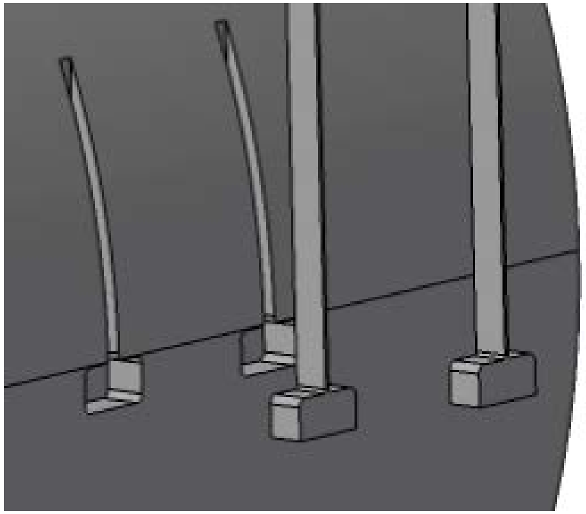


Bonding nail heads to ribbon

Test ribbons 100 x 5x 0.8 mm

- Made by Impex (Germany)
- 5 grades/types of surface treatment: ground, thermo-, chemical (CMP)-, diamond-, inspection- polished)
- Roughness measured with profiler (@ Amolf Amsterdam):

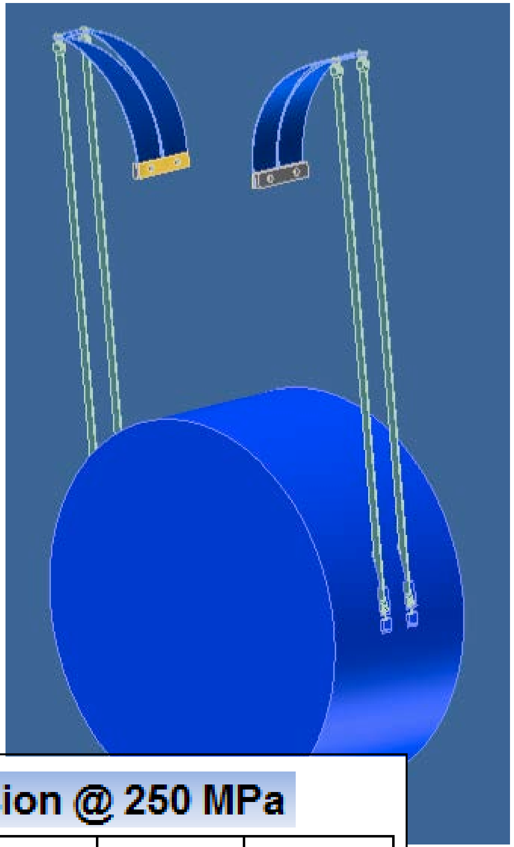
  between 700 and 0.4 nm RMS
- Thermal conductivity to be measured in Jena (hopefully soon!)

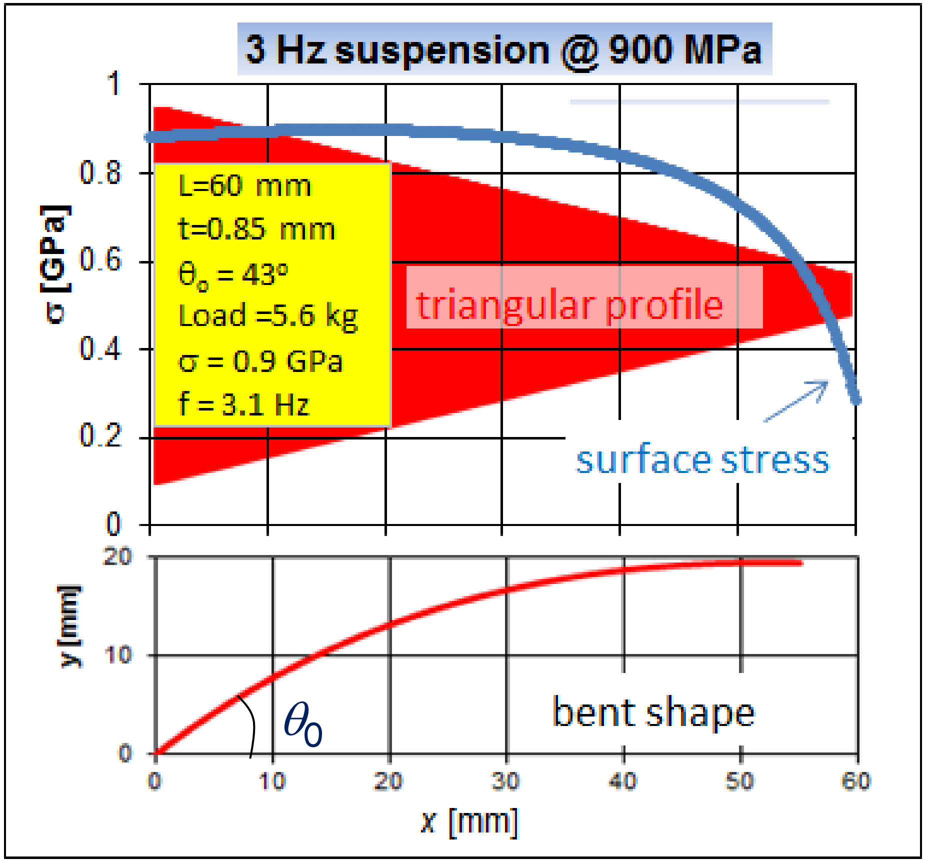


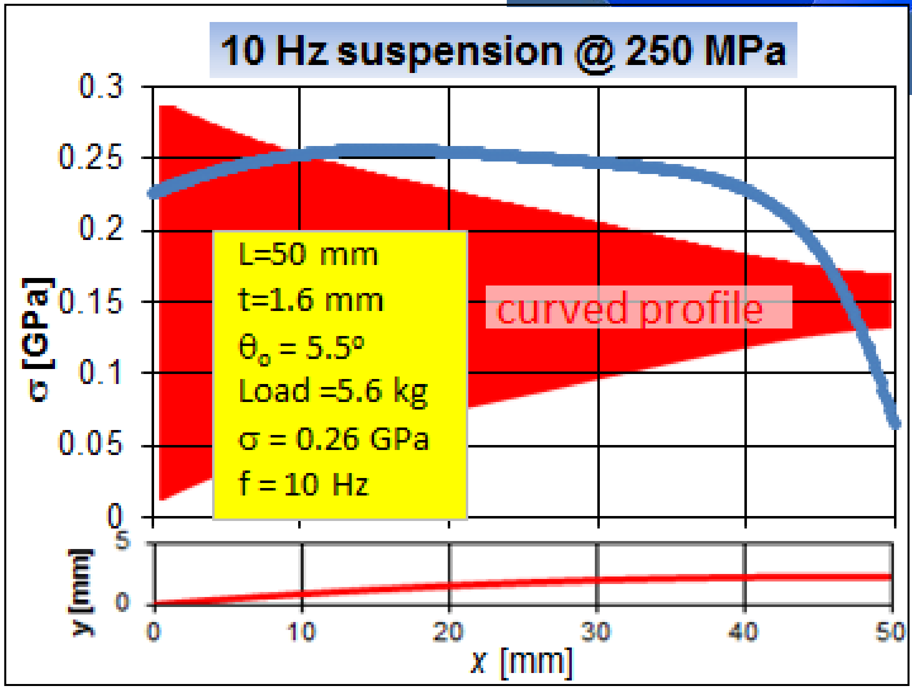
*Edge defined Film fed Growth

3.1. Triangular cantilever

- Candidates for compliant vertical suspension
- Obtainable compliance depends on allowed stress level
- Modeling by FEA or numerically
- For instance silicon blades for bKAGRA (5.5 kg/wire):





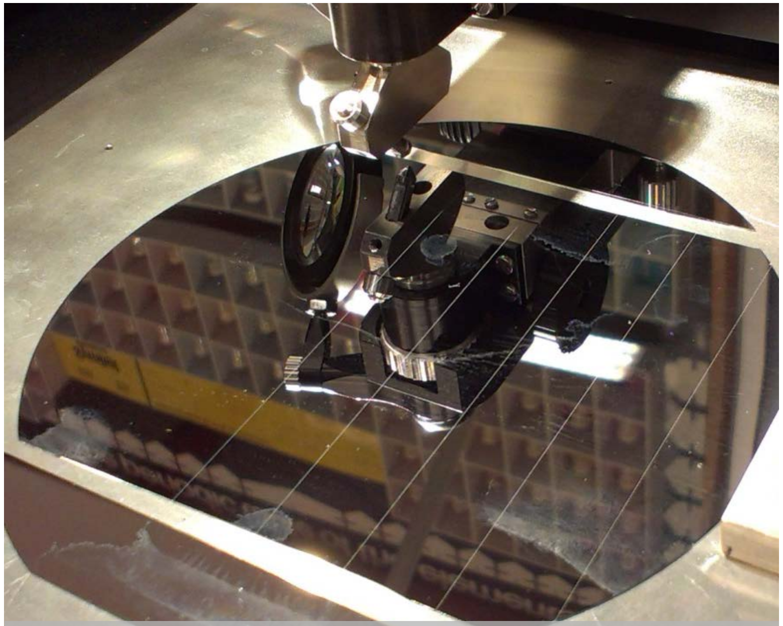


3.2. Silicon blades bending test

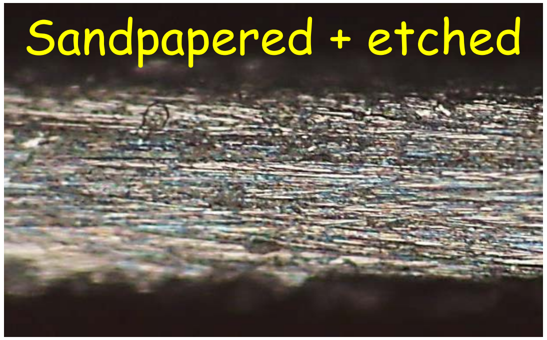
BSc student project @ Nikhef

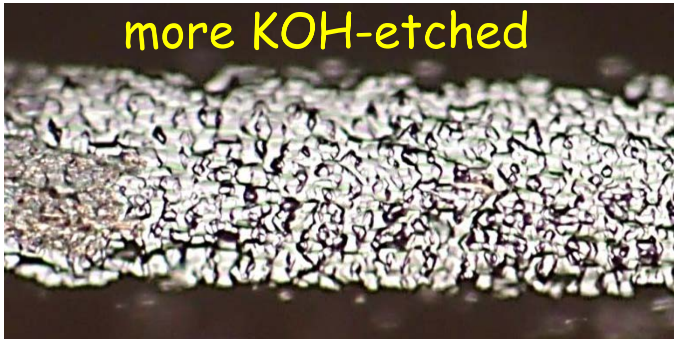
polished wafer samples

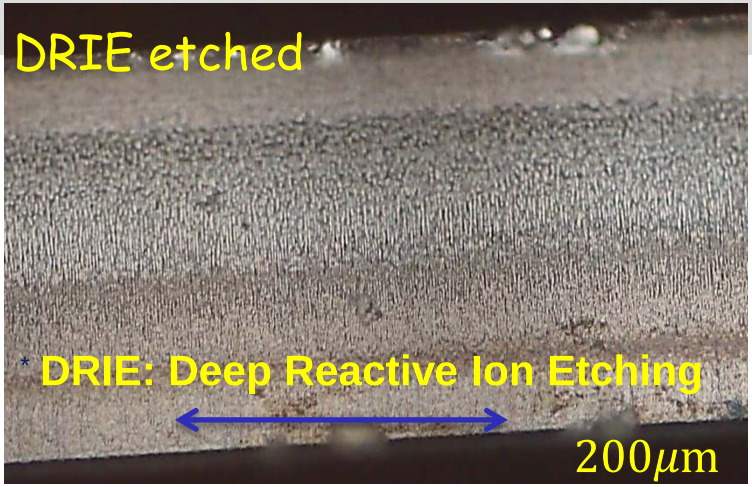
- From commercially available wafers
- Samples $L.w.t$ =50x5x0.4mm, 75x12x0.2 mm
- Hand-cut, machine-cut or DRIE* etched
- optionally sandpapered or KOH etched
- Microscopic/SEM edge inspection

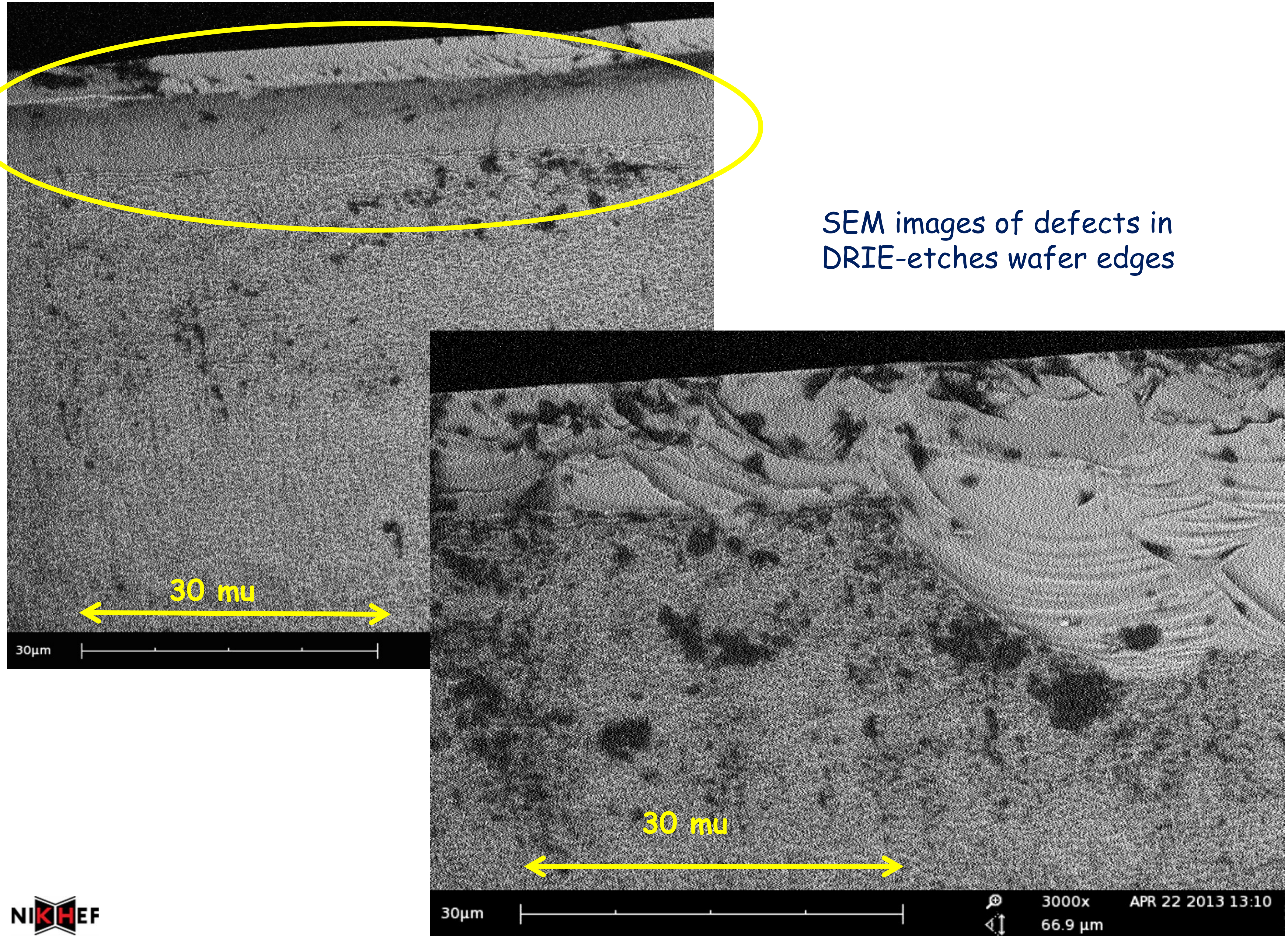

Wafer machine-cut @ Amolf Amsterdam

ion-etched wafer, 0.4 mm (TU









SEM images of defects in DRIE-etches wafer edges

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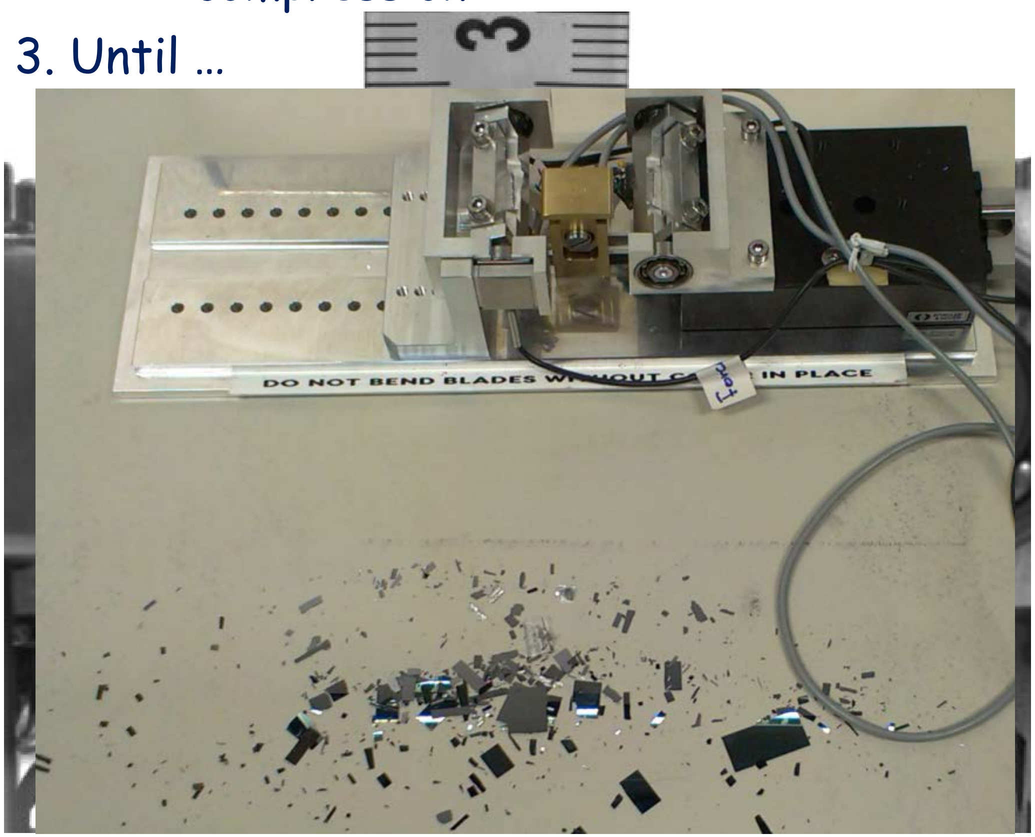
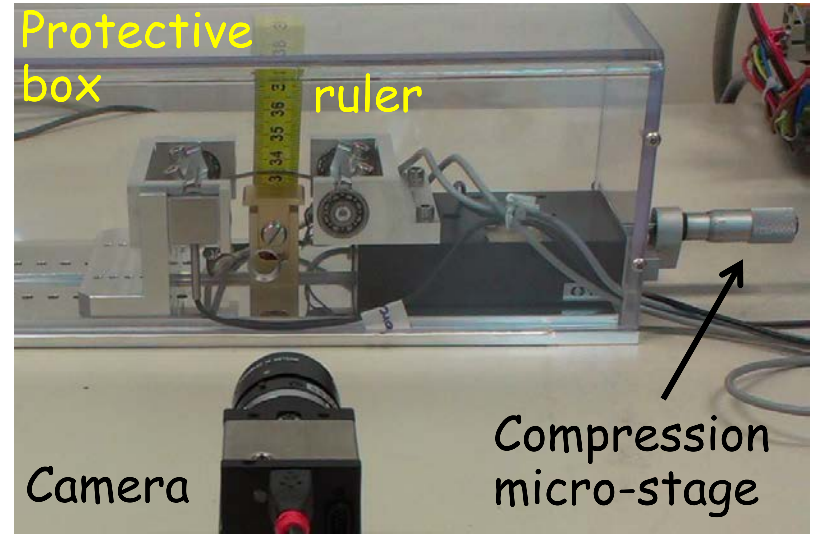
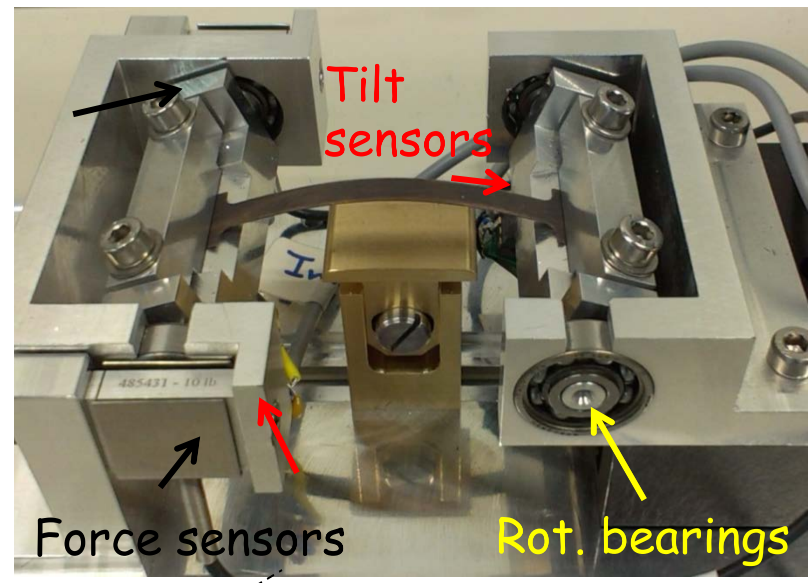
Bending test measurement setup

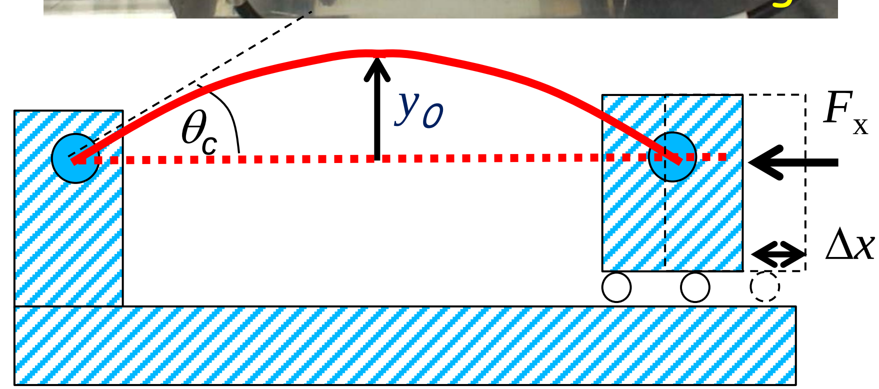
Stress-free at clamps

1. Mount flat blade in device, free length L
2. Measure after each compression step:
   - F_x compressive force
   - Θ_c clamp tilts
   - y_0 mid blade height
   - Δx compression
3. Until ...







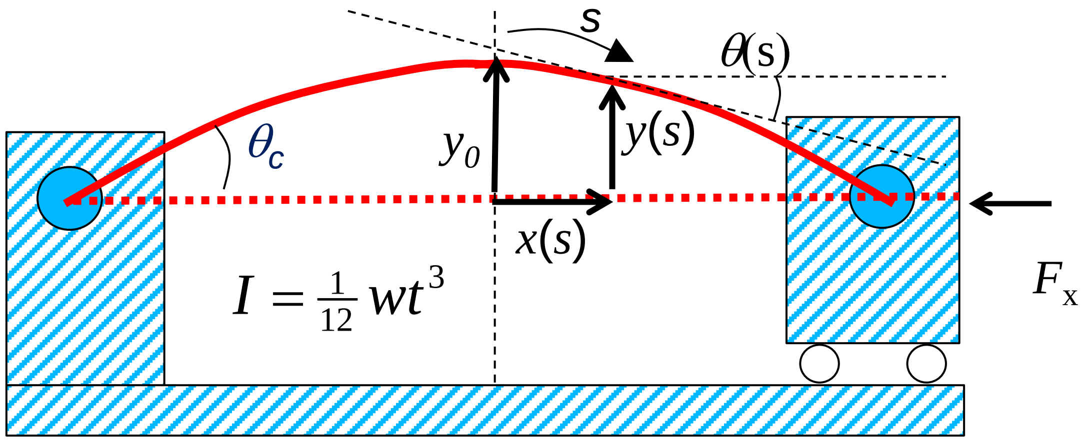


Bending test analysis

- Analysis based on (half) beam torque equilibrium

$$EI\frac{d^2\theta}{ds^2} = F_x \sin\theta, \quad s = 0....\tfrac{L}{2}$$

$$\frac{dy}{ds} = \sin\theta, \quad \frac{dx}{ds} = \cos\theta$$



- With boundary conditions

$$\theta(0) = x(0) = y(\tfrac{L}{2}) = 0$$

- plus one of the measurements (or more, if E and/or F_x are considered uncertain):

$$y(0) = y_0, \quad x(\tfrac{L}{2}) = (L - \Delta x)/2, \quad \theta(\tfrac{L}{2}) = -\theta_c$$

- Maximum stress estimations either directly or via model:

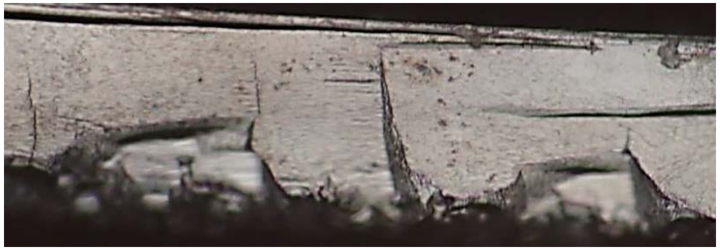
$$\sigma_m = \frac{y_0 F_x t}{2I} \qquad\qquad \sigma_m = \tfrac{1}{2} E t \frac{d\theta}{ds}(0)$$

Bending destructive test results (sofar..)

- (thin) flexures nicely cut along crystal axis can give up to 500 MPa strength (triangular blades not possible this way)
- DRIE-etching: allows for profiling, but strength limited to ~250 Mpa
- Hand-cut: large variation in strength
- Strong flexures can be selected
- To be done: serious edge polishing

Destructing 26 + more silicon blades..

	+ etched	DRIE-	Machine cut	
hand cut	edges (KOH)	etched	straight	inclined
max stress reached [Mpa] ±20				
270	197	220	484	122
540	189	350	502	179
170	200	250	567	239
150	broken during sand papering	250	569	
330		220	475	
290		340	515	
220		280		
280 ± 120	195 ± 5	270 ± 75	519 ± 40	180 ± 50

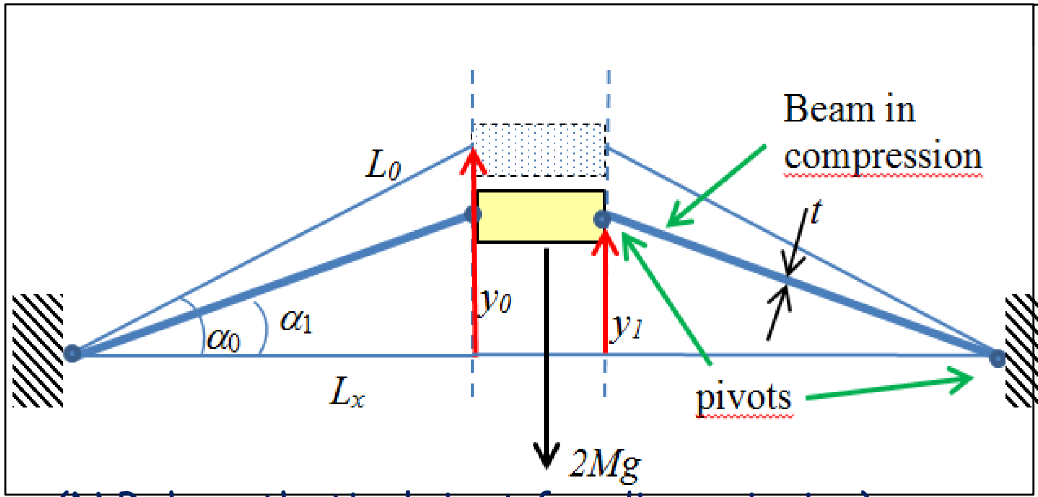


Compare : cut along crystal axis (left) or not (right)

3.3. Compressive beam suspensions

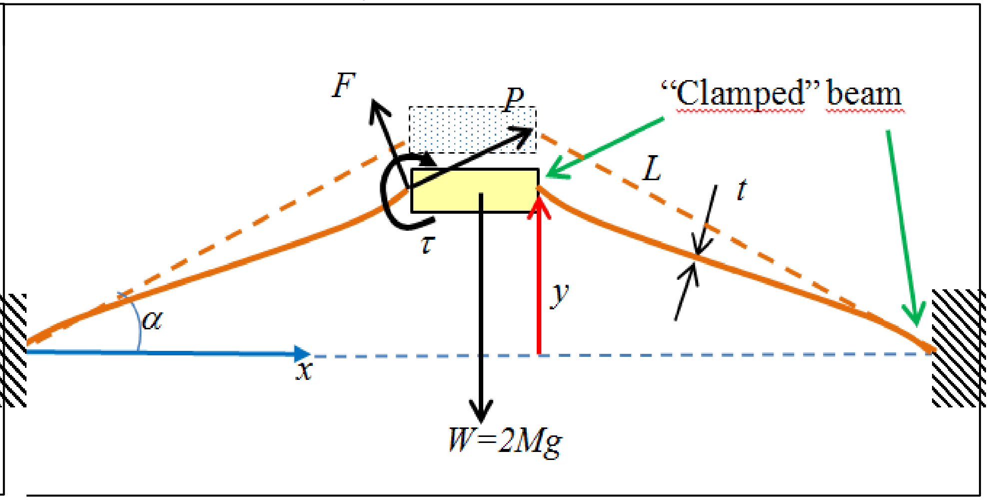
- Ceramics: high compressive, low tensile strength
- Beams in compression can withstand larger curvature, because …local stress = compressive offset (< 0) + local bending stress
- Moreover, compressive forces themselves can add-up to a soft spring (buckling)
- 2-step strategy:

(a) design optimization based on pure compressive (straight) beams



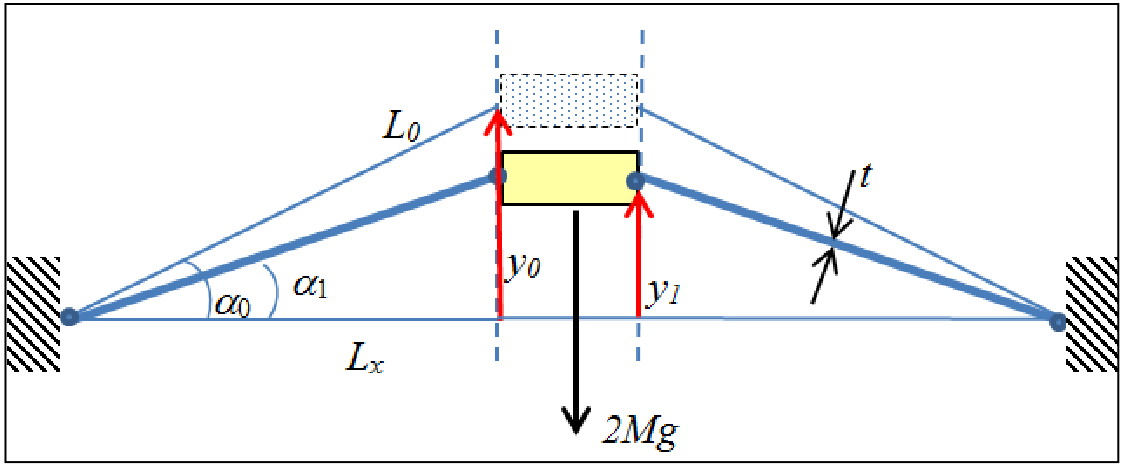
(N.B, hypothetical, just for dimensioning)

(b) extend this to clamped/monolithic beams in compression (William's toggle)



Purely compressive beam suspension design

For beam with rectangular cross-section



Given:

M : mass load per beam
L_0 : beam length
E : Young modulus
σ_m : compressive stress
f : required resonance frequency

Calculate:

t : beam thickness
w : width
α_0 : initial angle (or height y_0)
α_1 : final angle (or height y_1)

After some math…

$$\sin\alpha_0 = \sqrt{3}\, r\left[1 + \frac{kr}{6} + \left(\frac{5k^2}{72} - \frac{1}{2}\right) r^2 + O(r^3)\right]$$

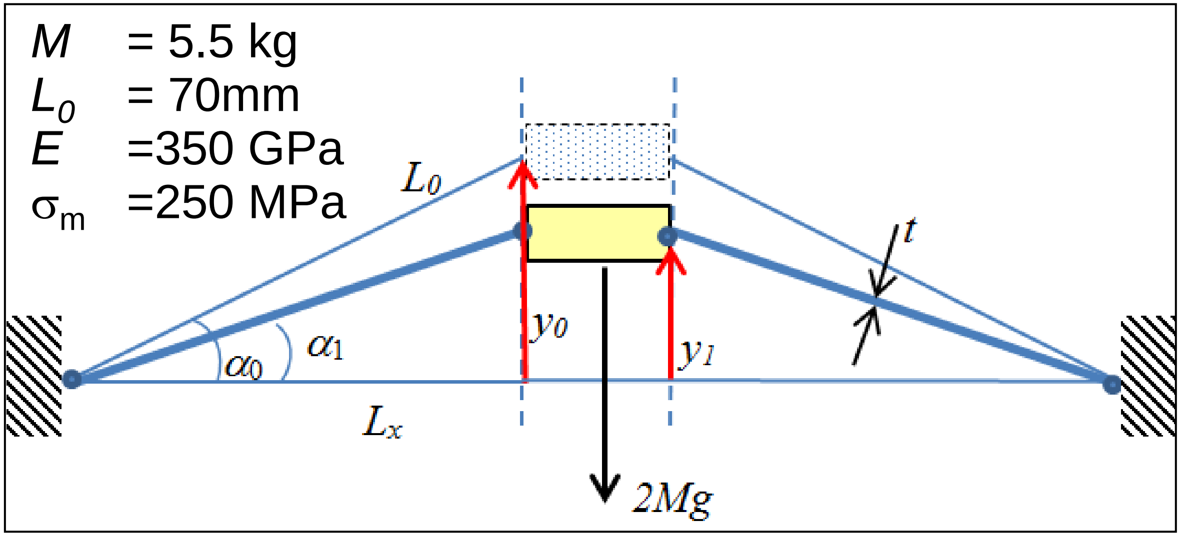
$$\sin\alpha_1 = r\left[1 + \frac{kr}{2} + \frac{k^2 r^2}{8} + O(r^3)\right]$$

$$t = L_0\sqrt{\frac{12}{\pi^2}\frac{\sigma_m}{E}}$$ beam buckling condition

$$w = \frac{Mg}{\sin\alpha_1 t\sigma_m}$$

with $r = \sqrt{\frac{\sigma_m}{E}}$, $k = \frac{Mg}{Ewt}$

Purely compressive beam suspension analysis: Applied to bKAGRA



sideward buckling

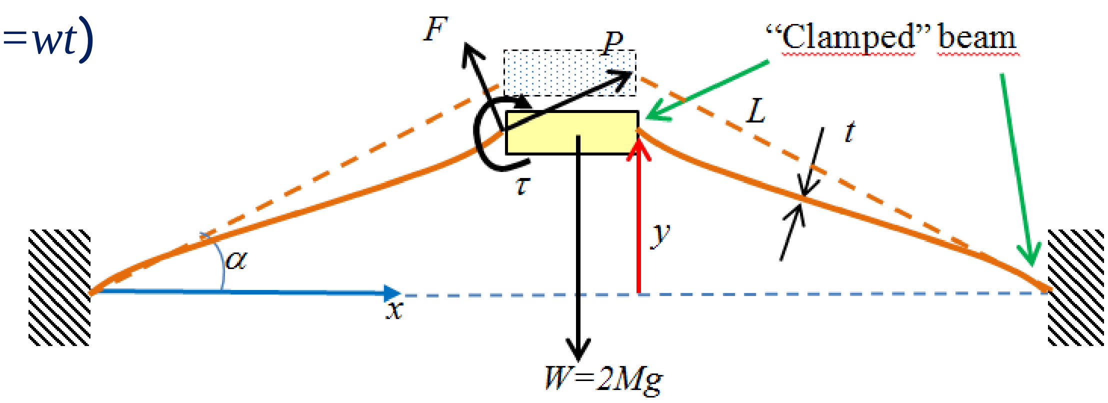
impossible	red
too risky	orange
possible	blue
safe	green

frequency	f	0	5	10	16	[Hz]
width	w	4	3.6	2	1.7	[mm]
thickness	t	2	2	2	2	
initial height	y_0	3.2	3.3	3.8	5	
equilibium	y_{eq}	1.9	2.05	2.7	4.3	
instability	y_{min}		1.8	1.6	1.3	
margin	Δy	0	0.25	1.1	3	

Selected "seed" for next model

Including bending: Williams* toggle

For a given geometry ($\alpha \ll 1$, L, EI, $A{=}wt$) and vertical deflection $\delta = y - y_0$, the forces are:



$$P = \frac{AE}{L}\left(\delta \sin\alpha - \frac{6\delta^2}{10L}\right)$$

$$F = \frac{P}{L}\frac{\mu(\rho)}{1-\mu(\rho)}\delta$$

$$W = 2(F + P\sin\alpha)$$

with $\mu(\rho) = \frac{\pi}{2}\sqrt{\rho}\cot\frac{\pi}{2}\sqrt{\rho}$, $A{=}wt$

$$\rho = P/P_E\ ,\quad P_E = \frac{\pi^2 EI}{L^2}$$

Surface stress distribution

$$\sigma_b(x) = \frac{Etk}{2P}(k\tau\cos kx - F\sin kx)$$

with $\tau = P_E\frac{\rho}{2(1-\mu(\rho))}\delta$ beam torque

$$k^2 = \frac{P}{EI}$$

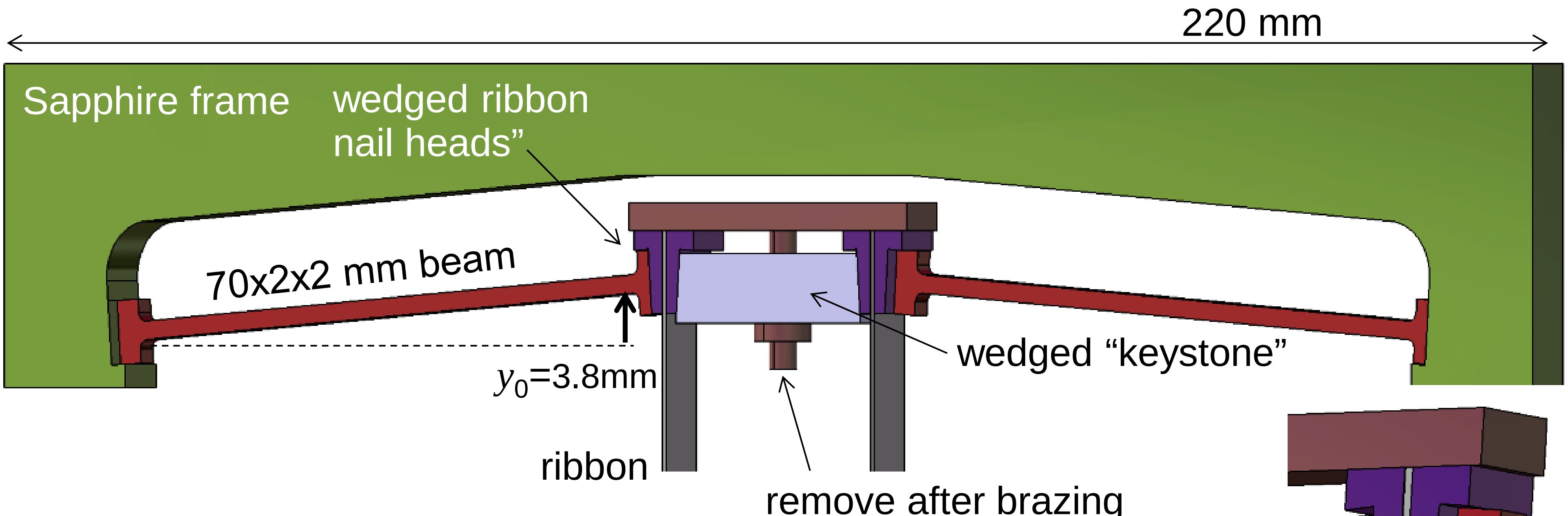
"wave" number

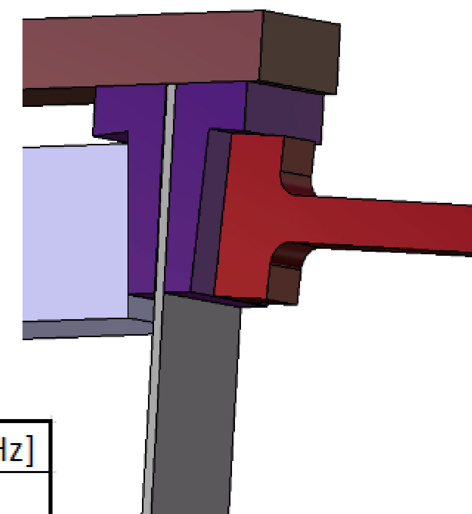
Average (compressive) stress

$$\sigma_c(x) = \frac{P}{A}$$

*Williams, F.W., 1964. An approach to the non-linear behavior of the members of a rigid jointed plane framework with finite deflections. *The Quarterly Journal of Mechanics and Applied Mathematics*, **17**(4):451-469.

Williams toggle, applied to bKAGRA



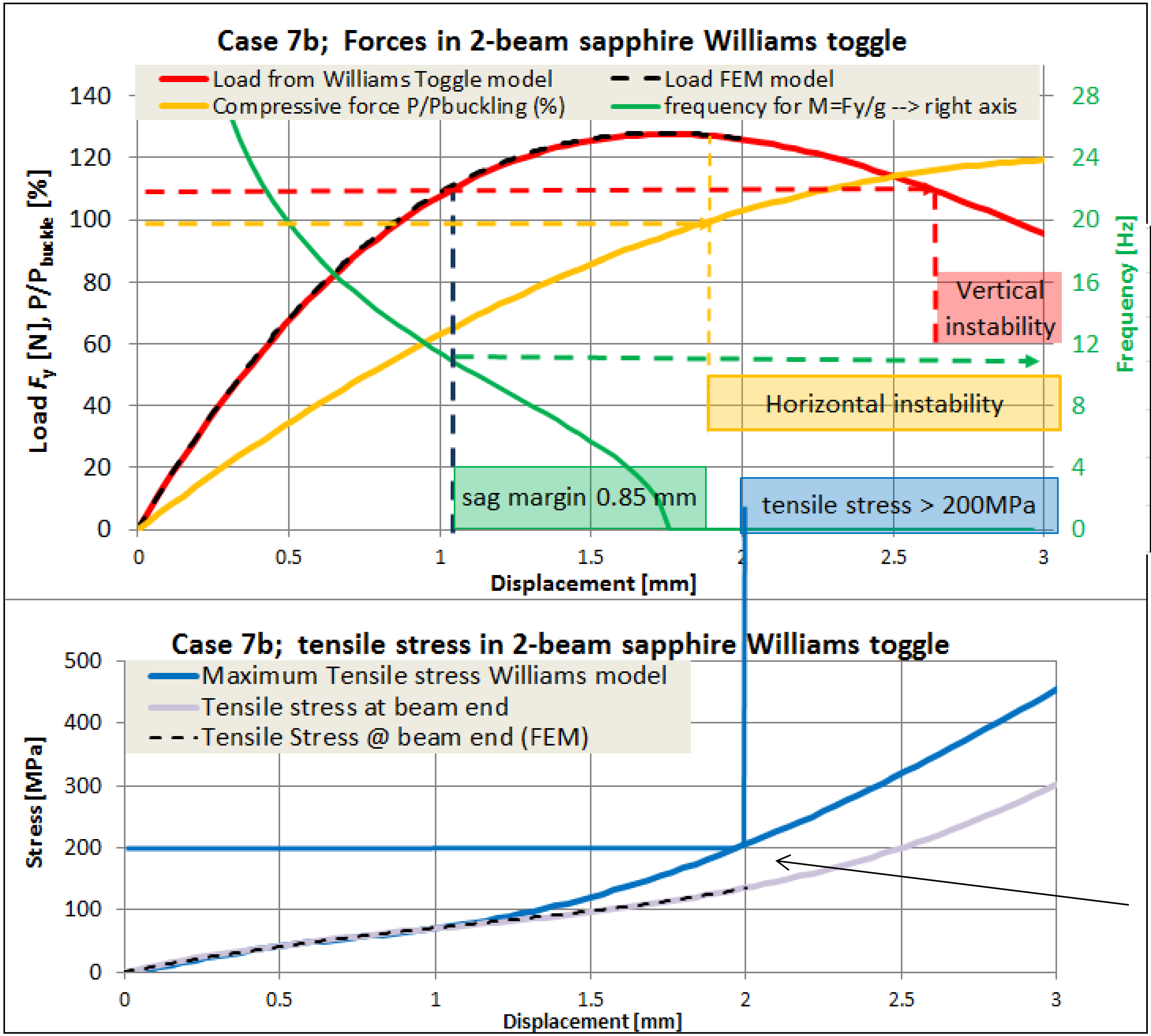


- All materials sapphire to cope with thermal expansion
- Adopt parameters from pure compressive 10 Hz beam study
- Allow some maximum tensile stress

frequency	f	10	[Hz]
width	w	2	
thickness	t	2	
initial height	y_0	3.8	[mm]
equilibium	y_{eq}	2.7	
instability	y_{min}	1.6	
margin	Δy	1.1	

See slide 12

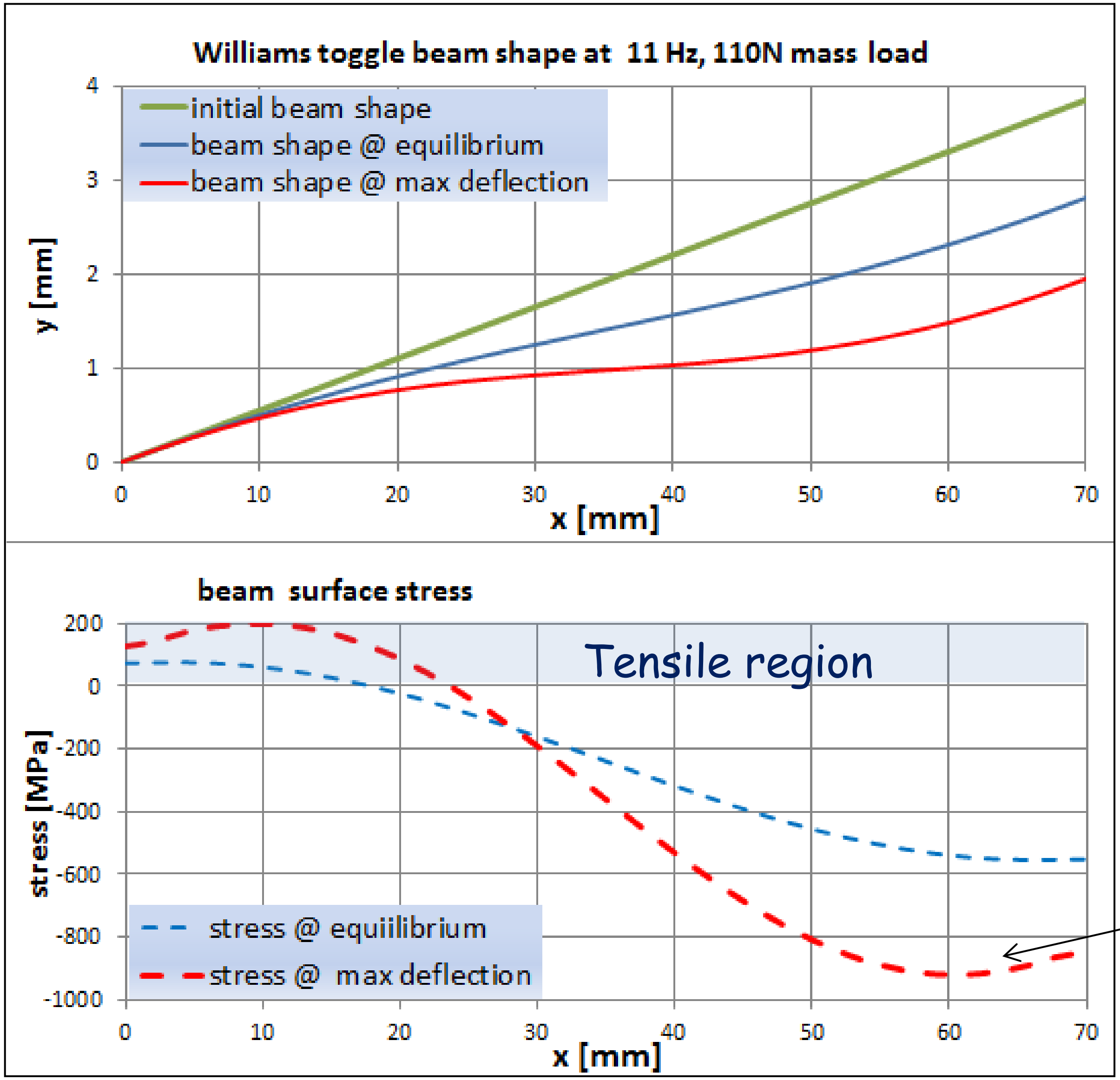
Williams toggle, model results (1)



		Compressive only	+ bending	
frequency	f	10	11	Hz
width	w	2		mm
thickness	t	2		
initial height	y_0	3.8		
equilibium	y_{eq}	2.7	2.75	
minimum	y_{min}	1.6	1.90	
margin	Δy	1.1	0.85	

- Dynamic behaviour of the models is almost the same
- Bending makes it just a bit stiffer
- Max stress not at beam end

Williams toggle, model results (2)



Beam shape

Surface stress

- Max stress ≠ stress @ beam end
- Compressive: 950 MPa

Summary

- In-mirror micron-accurate (USM-) machined sapphire suspension slots seem feasible

- sapphire grown ribbons (from Impex) have roughness down to 4 nm rms

  The surface-dependence of their thermal conductivity will be tested in Jena

- Silicon cantilever blade suspension requires strength: 250 MPa for 10 Hz
  500 MPa for 3 Hz

- Silicon Bending tests sofar (0.2-0.4 mm) give:

270 (80) MPa	DRIE-etched blades
520 (40) MPa	cut along crystal axis

- Compressive beam suspension was studied (Williams toggle)

  For KAGRA 10 Hz suspension conceivable with tensile stress < 200 MPa

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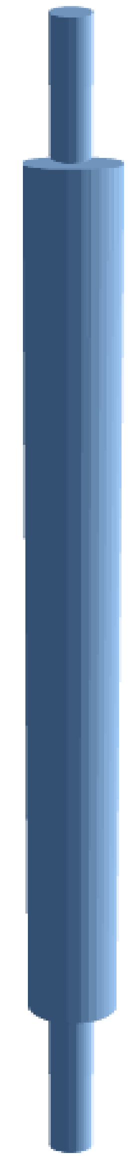
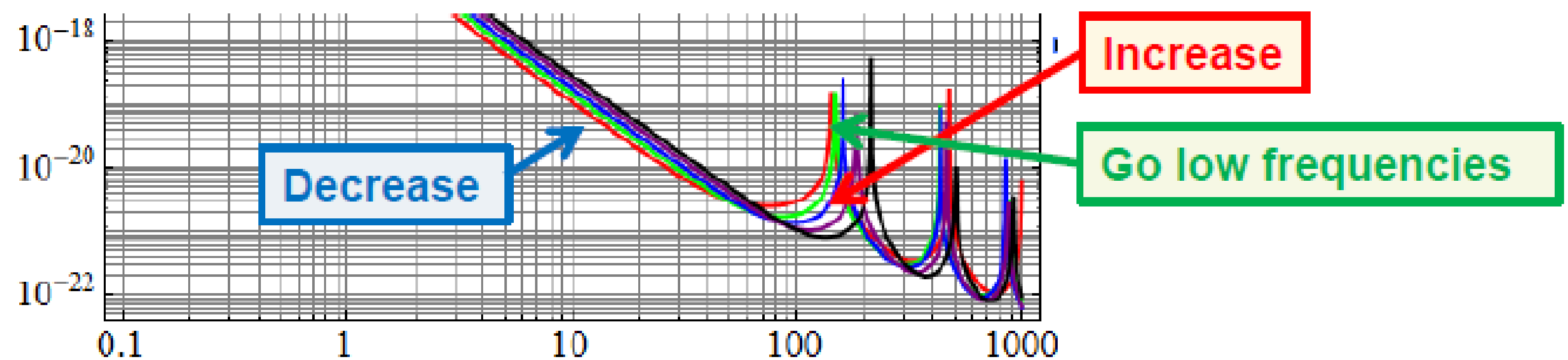
Backup

From Sekiguchi

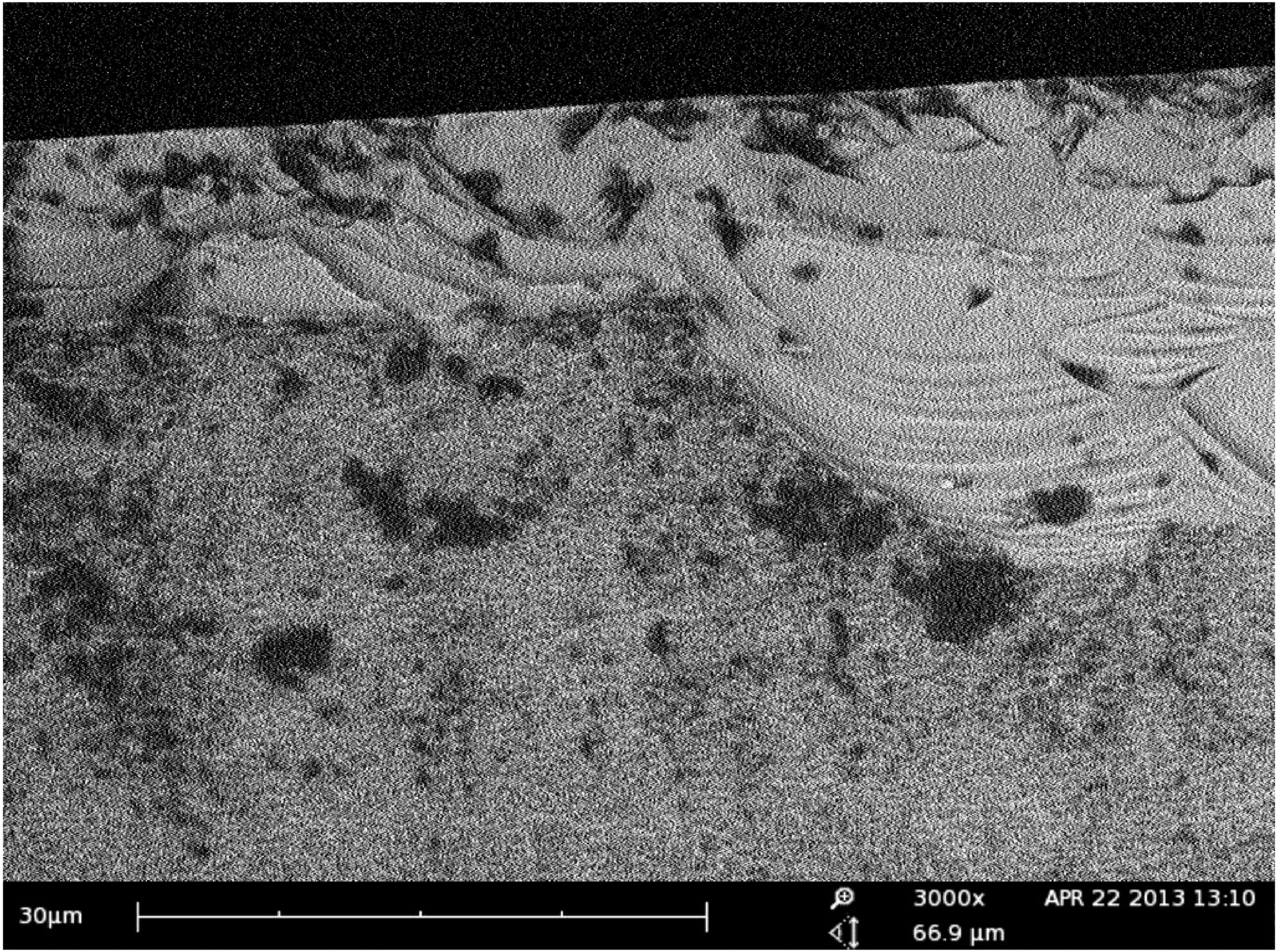
*Non-uniform rod concept: Takanori Sekiguchi
Elites f2f meeting, Tokyo, 2012/2/4



- There is a **trade-off** of using non-uniform fibers :
  - The suspension thermal noise at low frequencies (<50 Hz) decreases, by the “dissipation dilution” effect.
  - However, the suspension thermal noise at high frequencies (~100 Hz) increases and the resonant frequencies of the violin modes get lower.



- Resonances of the violin modes and vertical mode still remain at ~100 Hz.



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