EPFL

https://sciences.ncsu.edu/news/james-webb-space-telescope-qa-with-astrophysicist-bordoloi/

MICRO 372 - Advanced Mechanisms for Extreme Environments

Chapter 4b

Advanced mechanisms design

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Kinematic analysis - definitions

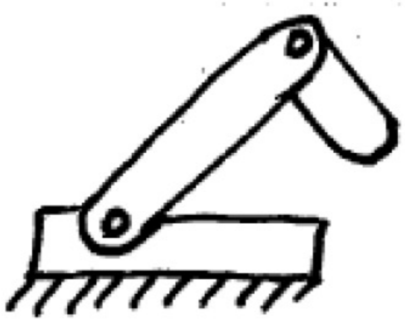
- **Kinematics**: architectural skeleton of a mechanism
- **Internal mobilities**: sum of DOF of individual flexure joints of a mechanisms
- **Kinematic chains**: paths from output to input of a mechanism
- **Kinematic loops**: closed kinematic chains within the mechanism
- **External DOFs**: DOF of the output of a mechanism
- **Internal DOFs**: DOF of intermediate stages in a mechanism
- **Overconstaints**: number of times the output or intermediate stages are constrained several times for a given DOF
- **Isostaticity**: feature of a kinematic with no overconstraint neither internal DOF

Kinematic analysis – Mobility equation

- Grübler formula using segments and joints numbers:

$$DOF = \sum Mo - 6(\underbrace{k - n + 1}_{= L})$$

- External + internal DOF → DOF
- Internal mobilities of joints → $\sum Mo$
- Number of joints → k
- Number of segments → n
- Fixed base → $+1$
- Number of loops → L



k = 2
n = 3
Mo = 2

Kinematic analysis – Mobility equation

- Grübler formula (with kinematic loops):

$$DOF = \sum Mo - 6L$$

- Modified formulation in 2D (planar anaylsis):

External DOF — Internal DOF — Internal mobilities of joints — Kinematic loop — Overconstraints

$$DOF_e + DOF_i = \sum Mo - 3L + OC$$

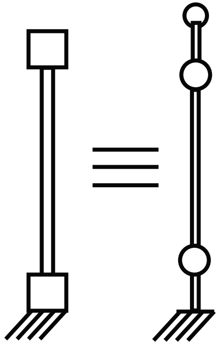
- Modified formulation in 3D (volumetric analysis):

$$DOF_e + DOF_i = \sum Mo - 6L + OC$$

Kinematic analysis – Model of a blade

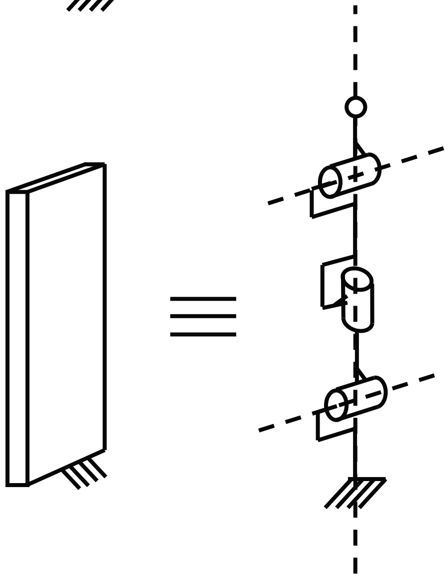
In 2D:

- the kinematic equivalent of a blade is 2 pivots



In 3D:

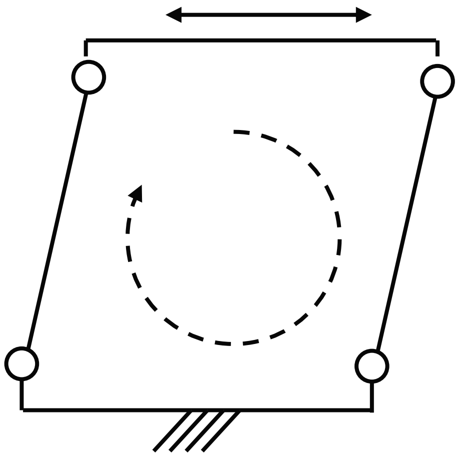
- the kinematic equivalent of a blade is 3 pivots



Kinematic analysis - Parallelogram stage

- DOFe = 1
- DOFi = 0
- Mo = 4
- L = 1
- OC = 0 or 3?

depends on the analysis type (2D or 3D)



External DOF, Internal DOF, Internal mobilities of joints, Kinematic loop, Overconstraints

$$DOF_e + DOF_i = \sum Mo - 3L + OC$$

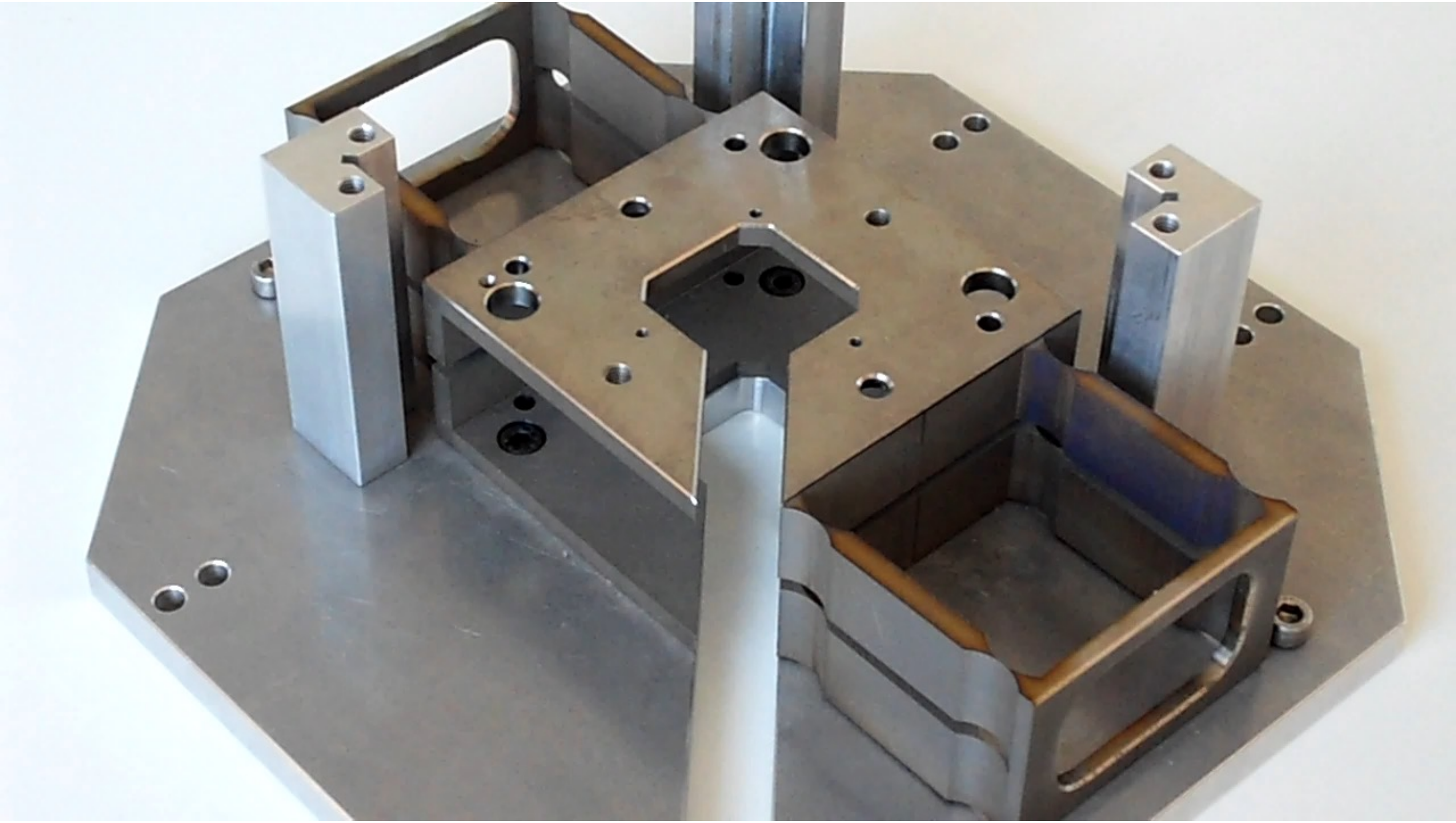
$$1 + 0 = 4 - 3 \cdot 1 + 0$$

$$DOF_e + DOF_i = \sum Mo - 6L + OC$$

$$1 + 0 = 4 - 6 \cdot 1 + 3$$

Each time a planar kinematic loop of blades is closed, 3 out-of-plane overconstraints appear!

Kinematic analysis – Overconstrained linear stage



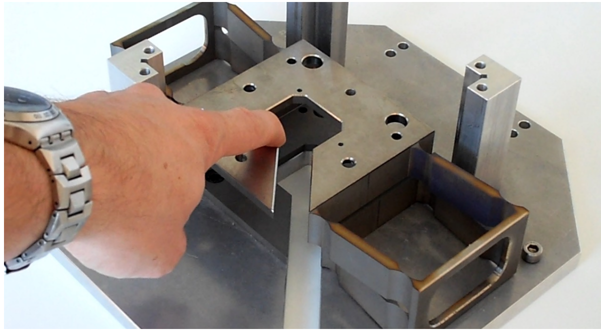
Kinematic analysis - Overconstrained linear stage

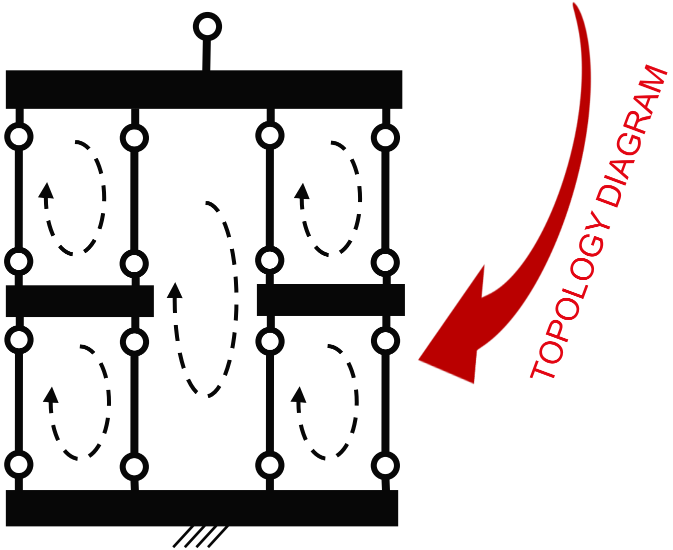
- External Degree of Freedom DOFe: 1
- Internal Degree of Freedom DOFi: 2
- Internal motions of joints Mo: 16
- Kinematic loops L: 5
- Overconstraints: ?

$$DOF_e + DOF_i = \sum Mo - 6L + OC$$

$$1 + 2 = 16 - 6 \cdot 5 + OC$$

$OC = 17$ (12 OC in parallelograms and 5 OC on output)



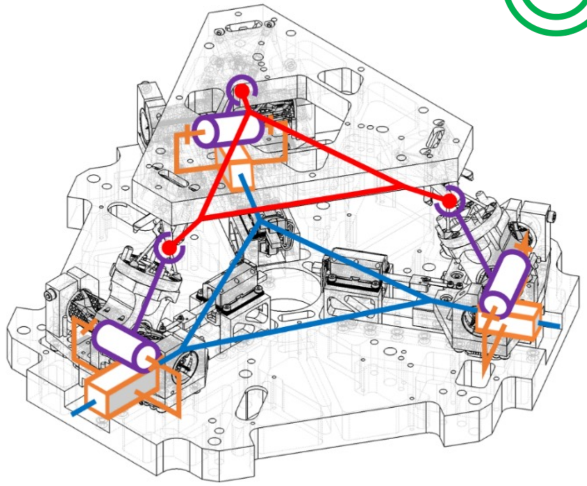


Kinematic analysis – Isostatic PULSAR tripod

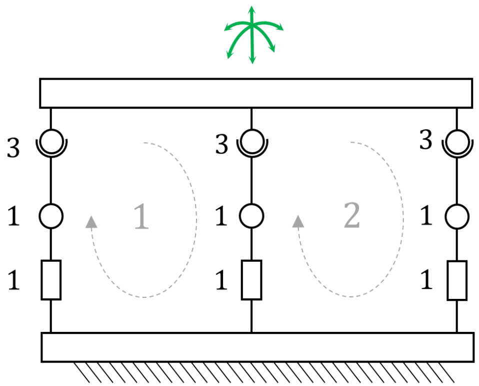
- External Degree of Freedom DOFe: 3
- Internal Degree of Freedom DOFi: 0
- Internal motions of joints Mo: 15
- Kinematic loops L: 2
- Overconstraints: 0

$$DOF_e + DOF_i = \sum Mo - 6L + OC$$

$$3 + 0 = 15 - 6 \cdot 2 + 0$$

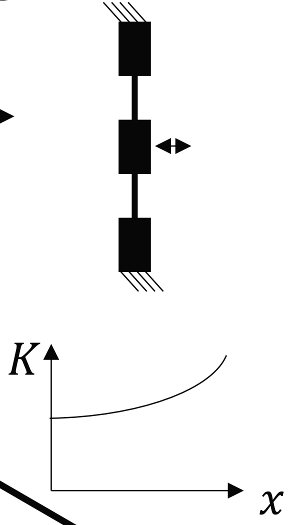


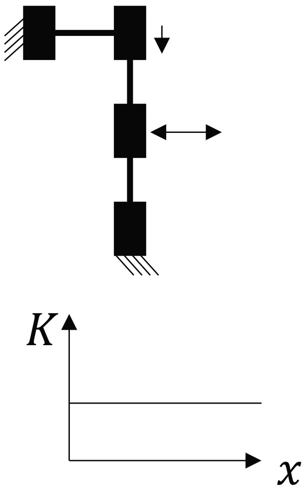
TOPOLOGY DIAGRAM

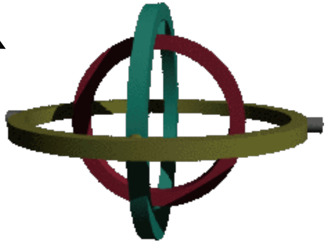


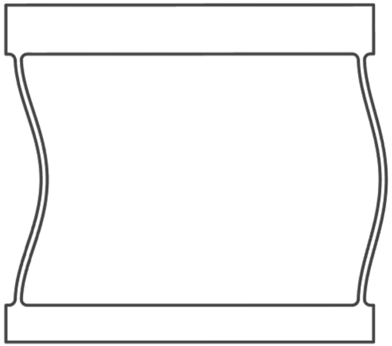
Kinematic analysis – Comments

- **Internal local DOFs** with **limited motion range** versus **internal DOF** with **large motion range**
- **Internal DOFs** of the mechanism versus **eigenfrequencies** of the flexures
- Check of the **sub-systems isostaticity**
- Kinematic analysis **does not consider** the **geometry of the mechanism**, only the **topology**
- Kinematic analysis **does not consider** the **singularities of the mechanism (e.g. gimbal lock)**
- **Kinematic analysis depends on hypothesis**: 2D versus 3D, number of DOF per flexure, etc..
- Interpretation matters!









Kinematic analysis - Conclusions

Internal DOFs:

- Can induce **uncontrolled vibrations**
- **Strokes and stress** on flexures are **not well controlled**
- **Parasitic motion** on output **may be induced** by uncontrolled motion of internal DOFs
- A **nightmare** to **increase the dynamics** of the mechanism!

Overcontraints:

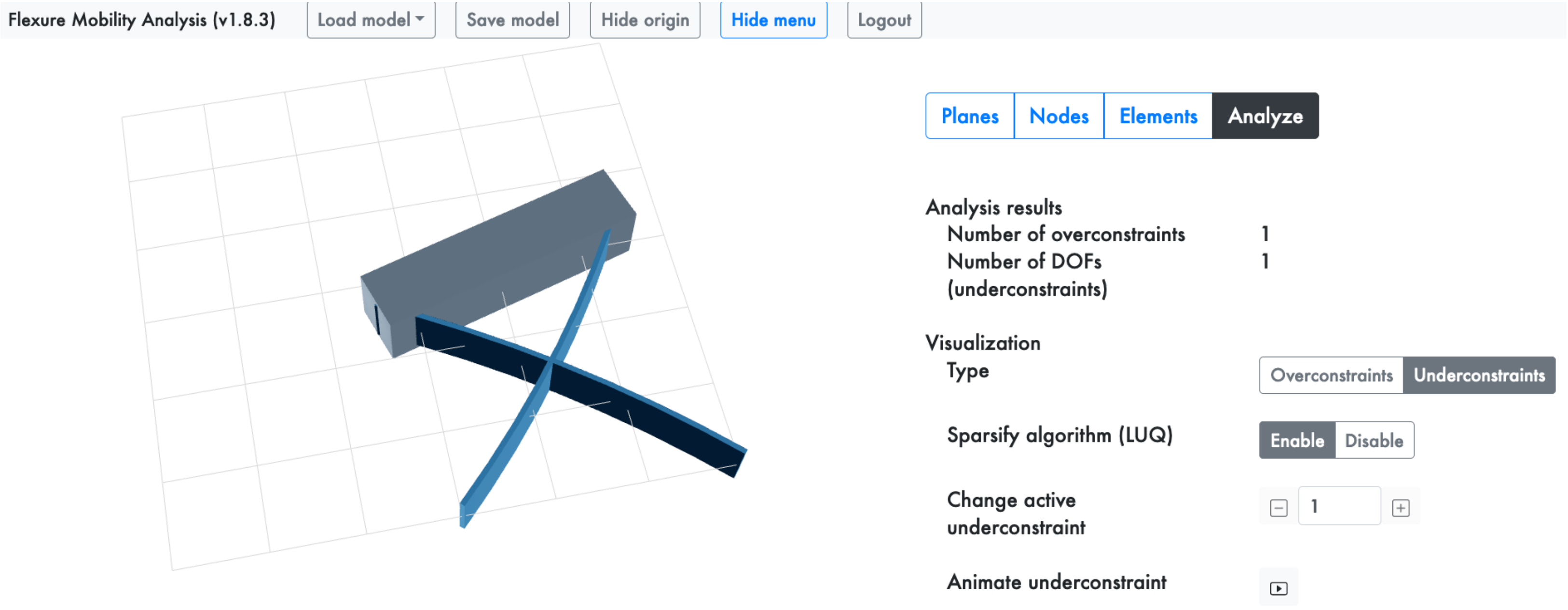
- Large **variations of stiffness** can appear
- Possibly **increases** significantly **the stress level**, even at rest position
- Makes the mechanism **less immune to tolerances variations** (manufacturing and assembly)
- **Can be beneficial** to improve transversal stiffnesses!

Kinematic analysis - Conclusions

- It is important to **carry out a kinematic analysis** when choosing the architecture of the mechanism, **at an early stage of development**.
- It is preferable to **avoid overconstraints and internal DOFs**.
- We **can accommodate overconstraints**, in particular out-of-plane OC (when a kinematic loop closes). We can also sometimes take advantage of this, for example to stiffen certain transverse stiffnesses or increase robustness.
- **Internal DOFs can be tolerated**, especially if the masses of the moving parts are low, the associated natural frequencies are high, and the system dynamics are low.

Kinematic analysis online tool (demo)

whatthedof.com



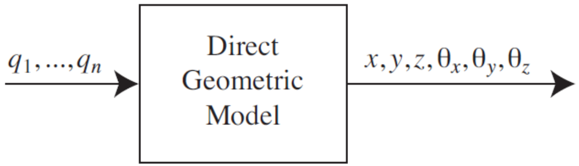
Geometric modeling

- **Direct Geometric Model** and **Inverse Geometric Model**
- **How to calculate your model?**
  - Analytical implementation
  - Validation through CAD sketches
- **Useful to** :
  - Calculate the motion ranges on actuators, on joints and on output (workspace)
  - Calculate the resolutions on actuator and on output
  - Calculate the parasitic motion
  - Optimize a system
  - Calibrate a system
  - Control a system

Direct Geometric Model and Inverse Geometric Model

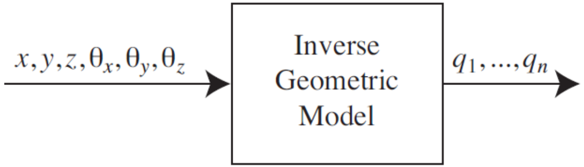
- The Direct Geometric Model (**DGM**) is the mathematical relation that give the output (or tool) positions as a function of the input (or motor) positions:

$$\vec{x}_i = \mathrm{DGM}(\vec{q}_i)$$



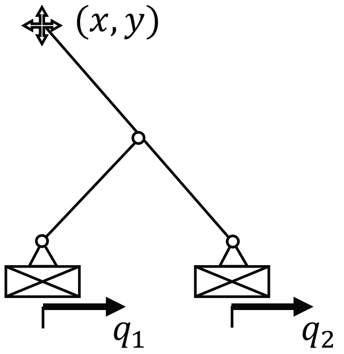
- Inverse Geometric Model (**IGM**) is the mathematical relation that give the input (or motor) positions as a function the output (or tool) positions:

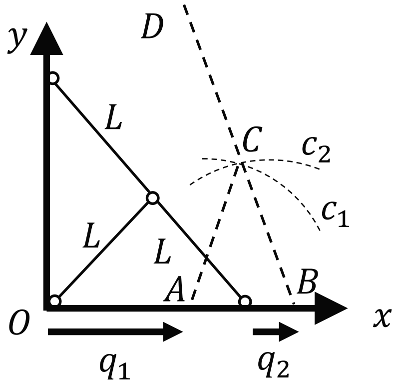
$$\vec{q}_i = \mathrm{IGM}(\vec{x}_i)$$



- x_i are tool coordinates and $\mathrm{q_i}$ are motor coordinates

Geometric modeling: DGM of the lambda mechanism





$$[x, y] = DGM([q_1, q_2])$$

$$A: (q_1, 0), B: (\sqrt{2}L + q_2, 0)$$

$$c_1: (center\ A, radius\ L)$$
$$c_2: \ (center\ B, radius\ L)$$

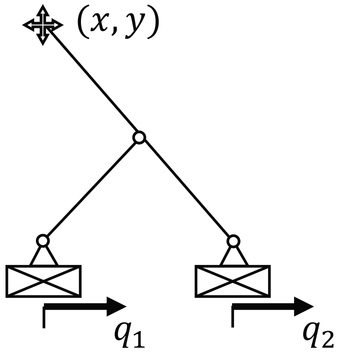
$$C: intersection(c_1, c_2)$$

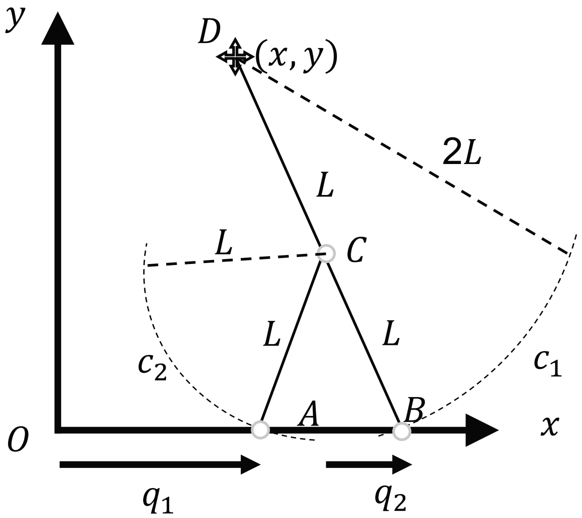
$$D: \ \overrightarrow{OD} = \overrightarrow{OB} + \overrightarrow{BD} = \overrightarrow{OB} + 2\overrightarrow{BC} = 2\overrightarrow{OC} - \overrightarrow{OB}$$

☞ EXERCICE 8

Geometric modeling : IGM of the lambda mechanism (principle only)

- Trace a circle **c1** (center: **D**, radius: **2L**)
- Intersect with line **y = 0** → **B**
- Take the middle point of **BD** → **C**
- Trace a circle **c2** (center: **C**, radius: **L**)
- Intersect with line **y = 0** → **A**
- Or.. use the symmetry of **A** and **B** regards to **C**
- Calculate **q1** and **q2**





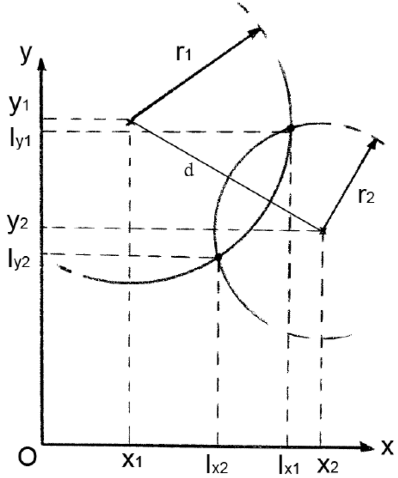
Geometric modeling: DGM and IGM setup

- Define carefully and precisely your **reference axes** and **geometric parameters**
- Draw a 2D (or 3D) **CAD sketch** (it can also be used to animate your CAD model)
- Some **generic models** may exist for known kinematics (e.g., hexapods, delta, ..)
- If it is not the case **do the math**
- No need to **solve analytically the equations**, unless you want for example to implement a fast model on controller
- Use a **CAD sketch to verify your** mathematical **model**
- **Circle intersections** can be calculated in planar models:

$$d = \sqrt{(x_2 - x_1)^2 + (y_2 - y_1)^2}.$$

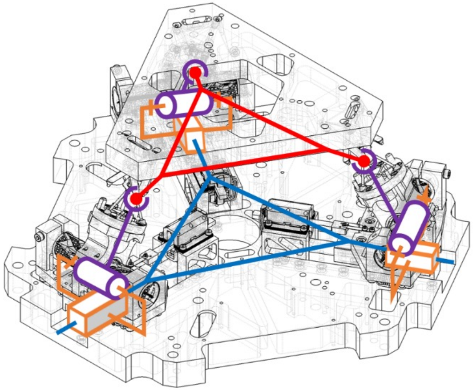
$$Ix_{12} = \frac{x_1 + x_2}{2} + \frac{(x_1 - x_2)(r_1^2 - r_2^2)}{2d^2} \pm \frac{y_2 - y_1}{2d^2}\sqrt{((r_1 + r_2)^2 - d^2)(d^2 - (r_1 - r_2)^2)}$$

$$Iy_{12} = \frac{y_1 + y_2}{2} + \frac{(y_1 - y)(r_1^2 - r_2^2)}{2d^2} \pm \frac{x_2 - x_1}{2d^2}\sqrt{((r_1 + r_2)^2 - d^2)(d^2 - (r_1 - r_2)^2)}$$

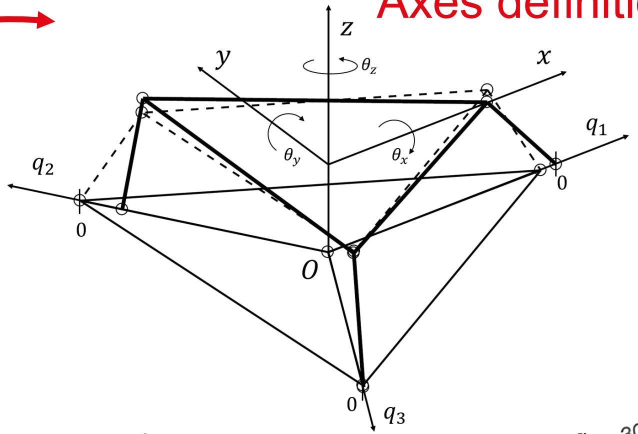


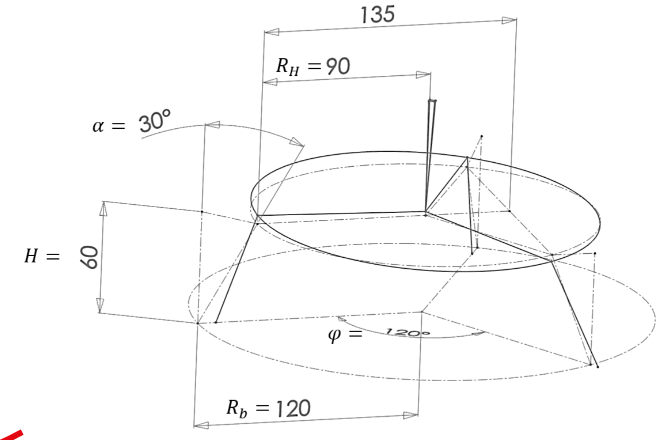
Geometric modeling: example of PULSAR IGM

Kinematic model

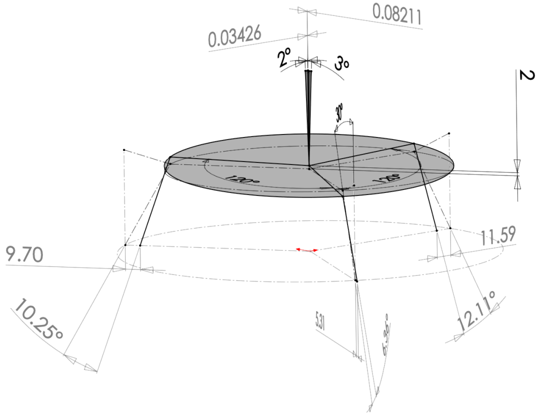


Axes definition





Geometric parameters definition



3D sketch

Geometric modeling: example of PULSAR IGM

The initial pivot positions are respectively for each axis:

$$P_{01} = (R_B, 0{,}0)^T \quad P_{02} = (R_B \cos(\tfrac{\pi}{3}), R_B \sin(\tfrac{\pi}{3}), 0)^T \quad P_{03} = (R_B \cos(\tfrac{2\pi}{3}), R_B \sin(\tfrac{2\pi}{3}), 0)^T$$

The initial gimbal center positions are: (with $H_v = H \cos \alpha$)

$$P_1 = (R_H, 0, H_v)^T \quad P_2 = (R_H \cos(\tfrac{\pi}{3}), R_H \sin(\tfrac{\pi}{3}), H_v)^T \quad P_3 = (R_H \cos(\tfrac{2\pi}{3}), R_H \sin(\tfrac{2\pi}{3}), H_v)^T$$

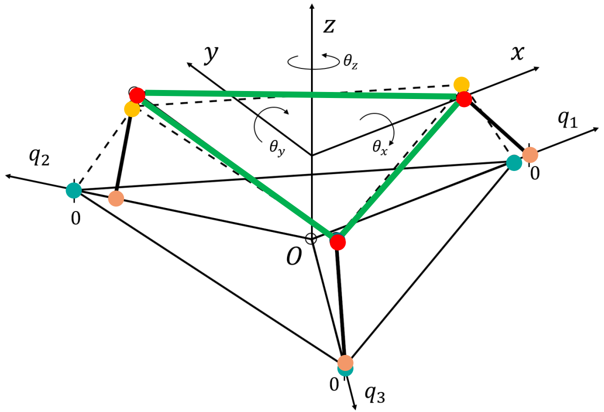
The triangle formed by P_1, P_2 and P_3 is equilateral of length $L = \|P_2 - P_1\|$.

The gimbal points $P_{1,new}$, $P_{2,new}$ and $P_{3,new}$ are obtained by applying two rotational matrices $R_x(\theta x)$ and $R_y(\theta y)$ around the x- and y-axis respectively and a vertical translation of z to P_1,P_2 and P_3:

$$P_{i,new} = R_y(\theta y) R_x(\theta x)\left(P_i + \begin{bmatrix}0\\0\\z\end{bmatrix}\right)$$

Where: $$R_x(\theta x) = \begin{bmatrix}1 & 0 & 0\\ 0 & \cos\theta x & -\sin\theta x\\ 0 & \sin\theta x & \cos\theta x\end{bmatrix}$$

and $$R_y(\theta y) = \begin{bmatrix}\cos\theta y & 0 & \sin\theta y\\ 0 & 1 & 0\\ -\sin\theta y & 0 & \cos\theta y\end{bmatrix}$$



F. Cosandier, J. Rouvinet, A. Ummel, D. Nguyen, V. Schaffter, and B. Nussbaumer, « Development of an am flexure-based tripod mechanism for the fine positioning of segmented telescope mirror tiles », in Proc. ESMATS, Online, Sept. 2021.

Geometric modelling: example of PULSAR IGM

Then, the new pivot positions are for $i = 1, 2, 3$:

$$P_{i,mot} = \lambda_i \cdot t_i$$

Where λ_i is a scalar and t_i the motor axis direction:

$$t_1 = (1,0,0) \qquad t_2 = (\cos(\frac{\pi}{3}), \sin(\frac{\pi}{3}), 0)$$

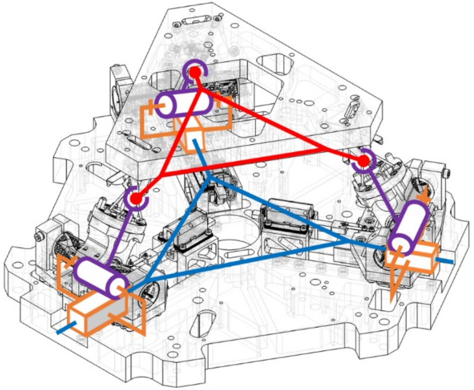
$$t_3 = (\cos\left(\frac{2\pi}{3}\right), \sin(\frac{2\pi}{3}), 0)$$

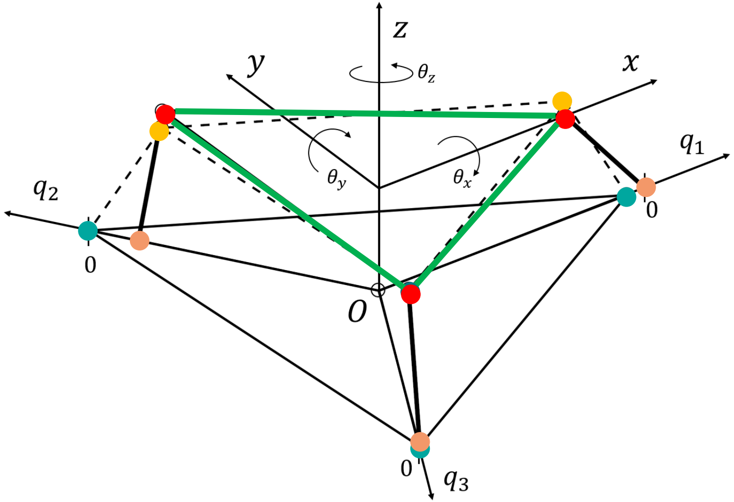
$$\lambda_i = t_i \cdot P_{i,new} + \sqrt{(t_i \cdot P_{i,new})^2 - (\|P_{i,new}\|^2 - H^2)}$$

Finally, the motor positions q1, q2 and q3 are obtained:

$$q_i = \|P_{0i} - P_{i,mot}\| \cdot \mathrm{sgn}(\lambda_i - \|P_{0i}\|)$$

N.B.: the model is not exact!





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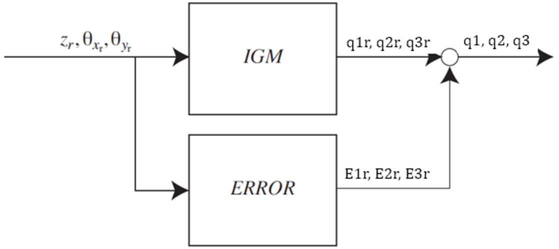
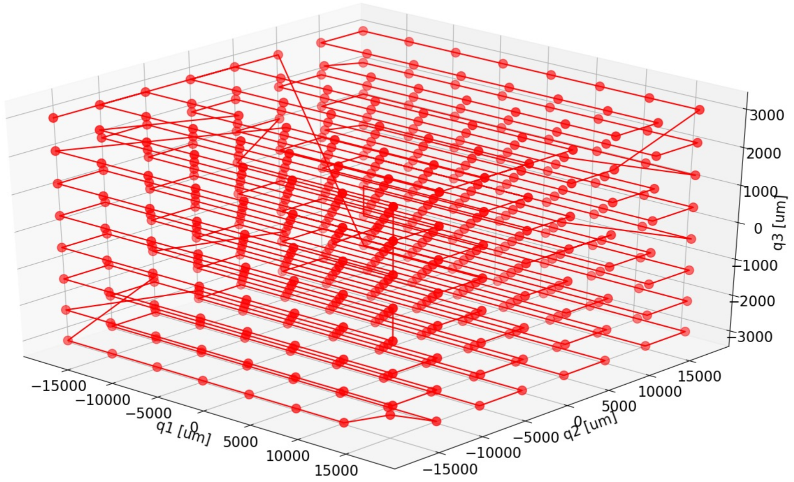
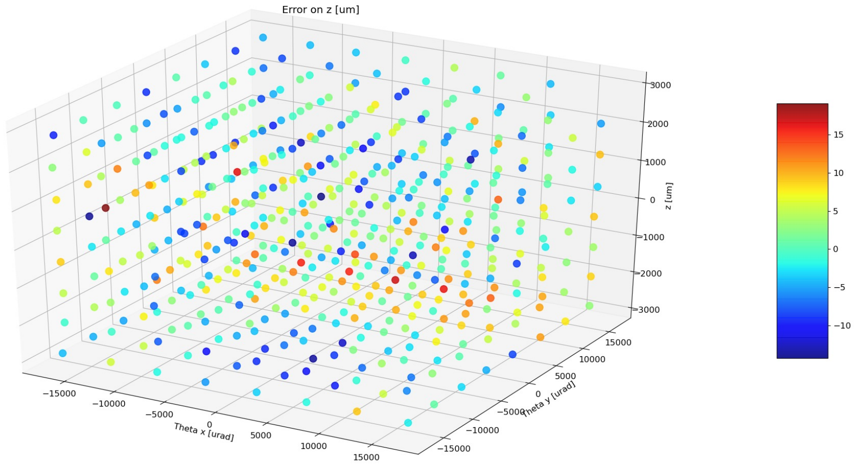
Geometric modelling: example of PULSAR calibration

- Position measurement is made over the whole workspace
- Error model is a fit at Xth order on N axis, including the cross axes terms
- You obtain a model with (many) coefficients

$$q1, q2, q3 = \mathrm{IGM}(z_r, \theta x_r, \theta y_r) + error(z_r, \theta x_r, \theta y_r)$$





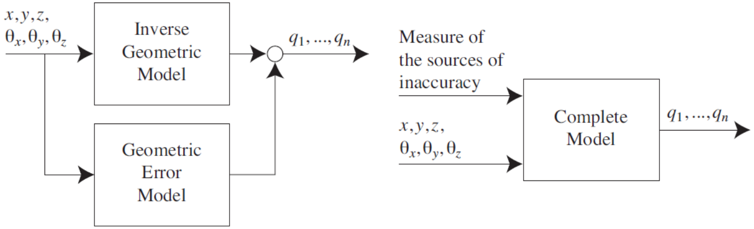


Geometric modelling: example of PULSAR calibration

- **Position measurement** is performed on the **whole working volume**
- **Error model is fitted** at Xth order on N axis, including the cross axes terms
  - Standard fit function is used
  - You get a model with many coefficients
- It is also possible to get the **full IGM** instead of just the **error model** through this method

Poly	B1	B2	B3
x	0.000336	-0.13751	0.137711
y	-0.15983	0.078817	0.080533
z	1.735941	1.74542	1.751814
x*x	-2.88E-08	3.42E-07	3.46E-07
y*y	5.36E-07	1.01E-07	1.00E-07
z*z	6.17E-05	5.57E-05	5.63E-05
x*y	0	-4.21E-07	4.37E-07
y*z	-1.09E-05	4.79E-06	4.92E-06
x*z	0	-8.23E-06	8.32E-06
x*x*x	0	-9.38E-13	1.16E-12
y*y*y	0	0	0
z*z*z	3.50E-09	2.77E-09	2.89E-09
x*x*y	0	2.04E-12	2.45E-12
x*x*z	0	4.47E-11	5.08E-11
y*y*x	0	-1.23E-12	1.87E-12
y*y*z	9.03E-11	1.66E-11	1.89E-11
z*z*x	0	-6.14E-10	6.43E-10
z*z*y	-8.35E-10	3.58E-10	3.93E-10
x*y*z	0	-5.12E-11	5.86E-11
Offset	-124.361	-99.3174	220.4672

$$\begin{aligned}\Delta\theta_x = \; & a_1 X + a_2 Y + a_3 Z + a_4 X^2 + a_5 Y^2 + a_6, Z^2 + a_7 XY + a_8 XZ + a_9 YZ + \\ & + a_{10} X^3 + a_{11} Y^3 + a_{12}, Z^3 + a_{13} XYZ + a_{14} X^2 Y + a_{15} X^2 Z + a_{16} Y^2 X + \\ & + a_{17} Y^2 X + a_{18} Y^2 Z + a_{19} Z^2 X + a_{20} Z^2 Y + a_{21}\end{aligned}$$



https://infoscience.epfl.ch/record/166120?ln=fr

Flexure optimization

- **Flexure model: FEM or analytical**

  - **FEM model**
    - Faster to set up, longer to solve than analytical
    - **Example**: COMAM flexure lattices

  - **Analytical model**
    - Must be validated on a FEM nominal case (no need for high precision)
    - Run on millions of parameter sets
    - **Example**: Vibration isolation platform gimbal

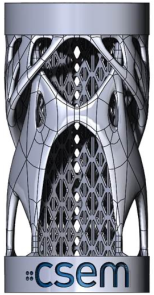
- **Model inputs**
  - Geometric parameters, material
  - Discrete or continuous values depending on the optimization method

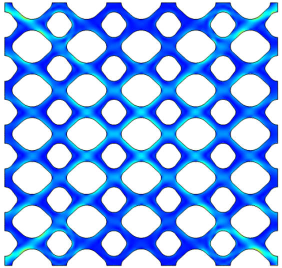
Flexure optimization

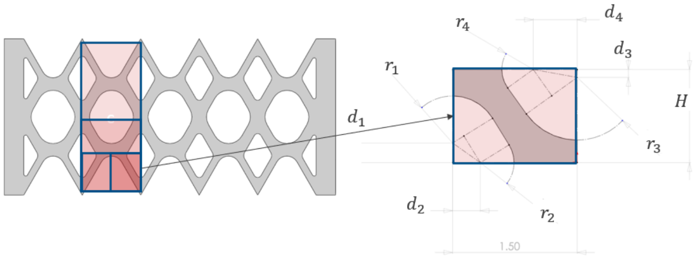
- **Optimization methods:**
  - Brute force
  - Monte-Carlo
  - Latin square
  - Genetic algorithm
  - Gradient based
  - ...
- **Input filters**
  - Based on impossible geometries
  - Based on overhang angle for AM
  - ..
- **Model outputs:**
  - Mass
  - Max stress
  - Stiffnesses
  - Eigenmodes
  - Buckling loads
  - ..
- **Output filters**
  - Limit stress
  - Required stiffnesses
  - ..

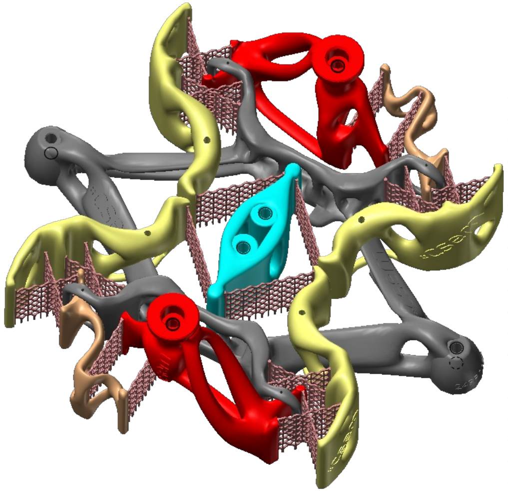
Flexure optimization – Example 1

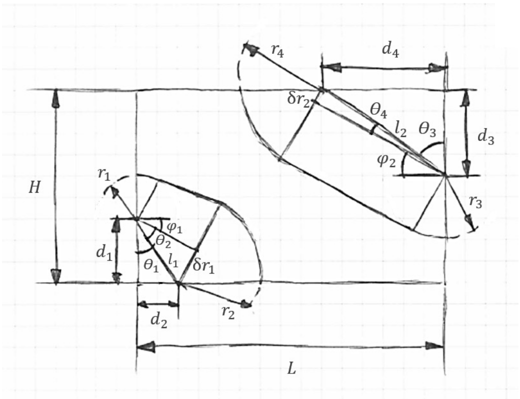
- Example: lattice flexures
- Optimized through FEM
- Monte-Carlo parameters sets generation
- Unit cell of mesh geometric parameters as inputs

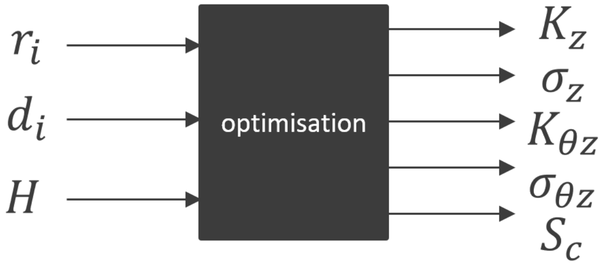








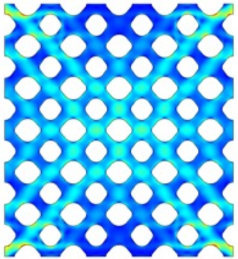
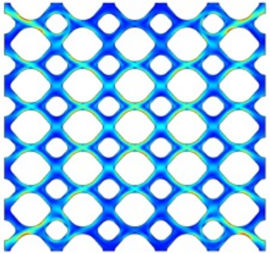
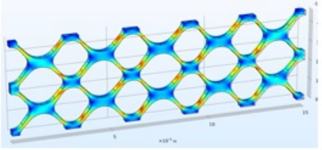
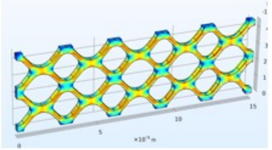




Flexure optimization – Example 1

- Outputs:
  - Rotation and vertical stiffnesses and stress (K_z, Sigma_z, K_rz, Sigma_rz)
  - Surface constancy (CS)
- Stiffness K_z is transformed into compliance C_z, so to be minimized

Symbol	K_z	C_z = 1/K_z	sigma_z	K_rz	sigma_rz	CS
Min/max	↗	↘	↘	↘	↘	↘
Weight		0.6	0.8	0.9	0.5	0.4

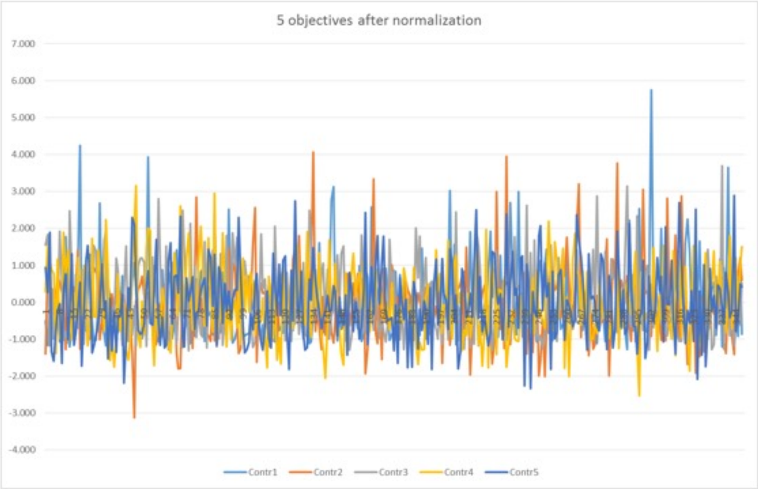





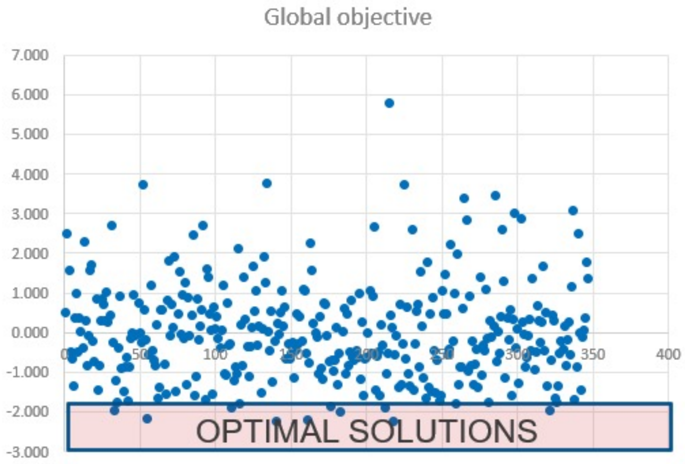

K_z max (sol 291) Sigma_z min (sol 45) K_rz min (sol 302) Sigma_rz min (sol 296) CS min (sol 242)

Flexure optimization – Example 1

- **Objective function**: summation of normalized and weighted contributions
- **Weights** are chosen based on the relative importance and criticality of the achievement of each objective

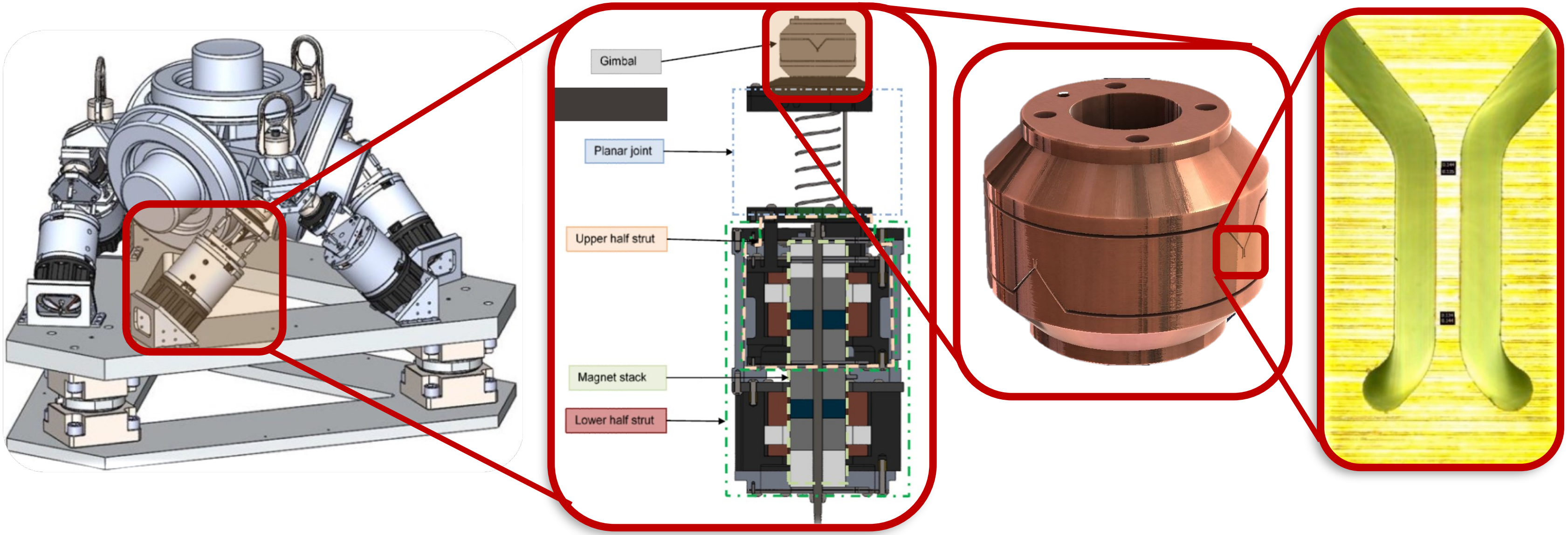
$$G_{obj} = w_{C_z}\frac{C_z/\overline{C_z} - 1}{STD\left(C_z/\overline{C_z} - 1\right)} + w_{\sigma_z}\frac{\sigma_z/\overline{\sigma_z} - 1}{STD\left(\sigma_z/\overline{\sigma_z} - 1\right)} + w_{K_{\theta z}}\frac{K_{\theta z}/\overline{K_{\theta z}} - 1}{STD\left(K_{\theta z}/\overline{K_{\theta z}} - 1\right)} + w_{\sigma_{\theta z}}\frac{\sigma_{\theta z}/\overline{\sigma_{\theta z}} - 1}{STD\left(\sigma_{\theta z}/\overline{\sigma_{\theta z}} - 1\right)} + w_{CS}\frac{CS/\overline{CS} - 1}{STD\left(CS/\overline{CS} - 1\right)}$$





Flexures optimization - Example 2

- **Vibration isolation and suppression system** for precision payloads in space
- **Hexapod** including six electromagnetic damping struts
- **Gimbal flexure optimization** made **analytically**



Flexures optimization - Example 2

- Baseline and specifications

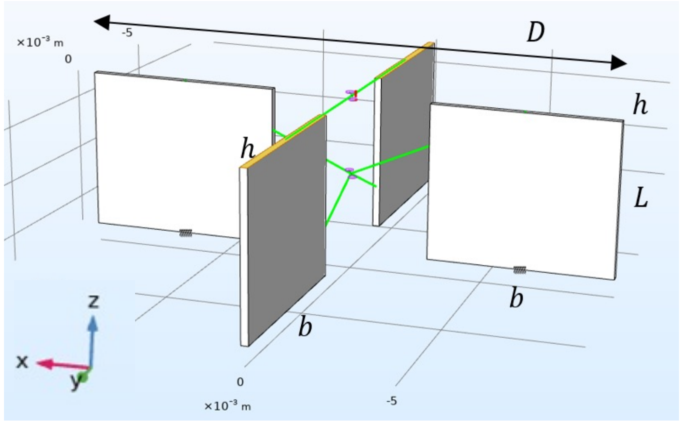
Input parameters			
Material		17-4 PH	
Young's Modulus	E	**196**	GPa
Pivot blade Outer Diameter	D	**20**	mm
Blade length	L	**12**	mm
Blade thickness	h	**0.26**	mm
Blade width	b	**9**	mm
Pivot internal diameter	di	**2**	mm
Joint overall length (flat-flat)	l		mm
Max angle stroke		**0.018**	rad

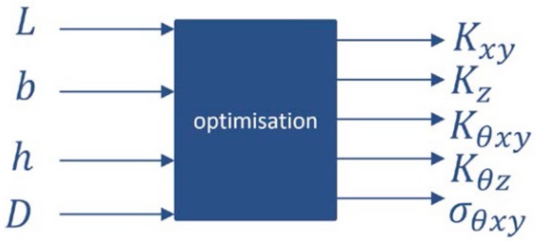
Analysis stiffness values (results)		
K_lateral_x	**35782**	N/m
K_lateral_y	**35782**	N/m
K_axial_z	**3.8E+07**	N/m
K_bending_rx	**0.430**	Nm/rad
K_bending_ry	**0.430**	Nm/rad
K_torsion_rz	**1.15**	Nm/rad

Target stiffness values (specs)		
K_lateral_x	1E+06	N/m
K_lateral_y	1E+06	N/m
K_axial_z	12000000	N/m
K_bending_rx	1.15	Nm/rad
K_bending_ry	1.15	Nm/rad
K_torsion_rz	600	Nm/rad

Flexure optimization – example 2 (optimization procedure)

- Parameter definition
- Analytical formulation of the stiffnesses
- Validation of the stiffnesses formulae
- Generation of sets of parameters
  - Define parameters boundaries
  - Monte Carlo set up (uniform distribution on each param.)
- Outputs assessment
  - Calculate Kxy, Kz, Krxy, Krz for each set of parameters
  - Calculate the functional von Mises stress for each set of parameters
- Optimization process
  - Define the weights for each individual objective
  - Define a global objective function including the weights and normalizations
  - Find the optimum of the global objective function
- Verification of the optimal solution (reanalysis)

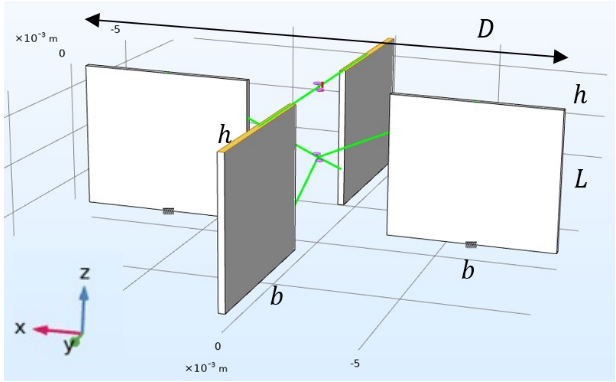




Flexure optimization – example 2

Analytical modeling and model validation through FEM:

- Lateral stiffness $$K_{xy} = \frac{2Eb^3h^3}{L^3(h^2 + b^2)}$$
- Axial stiffness $$K_z = \frac{Ebh}{L}$$
- Bending stiffness $$K_{\theta xy} = \frac{Ebh^3(D-b)^2}{L(2h^2 + 6(D-b)^2)}$$
- Torsional stiffness $$K_{\theta z} = \frac{Ebh^3(D-b)^2}{4L^3} + \frac{Gbh^3}{3L}$$
- Bending stress $$\sigma_{\theta xy} = \frac{\theta_{xy}Eh}{2L}$$



Axis	Stiffness FEM	Stiffness analytical	Unit	Error [%]
x	35782	35854.4	N/m	-0.20%
y	35782	35854.4	N/m	-0.20%
z	38220000	38220000	N/m	0.0000%
rx	0.43055	0.430531824	Nm/rad	0.0042%
ry	0.43055	0.430531824	Nm/rad	0.0042%
rz	1.15	0.90	Nm/rad	21.86%

Flexure optimization – example 2

- What is the value of the **bending stiffness** $K_{\theta\,xy} = ?$

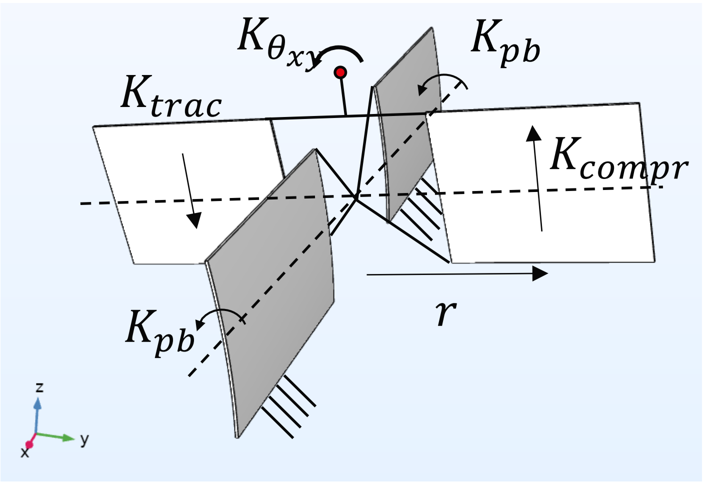
$$\frac{1}{K_{\theta\,xy}} = \frac{1}{2K_{trac\,\theta}} + \frac{1}{2K_{pb}}$$

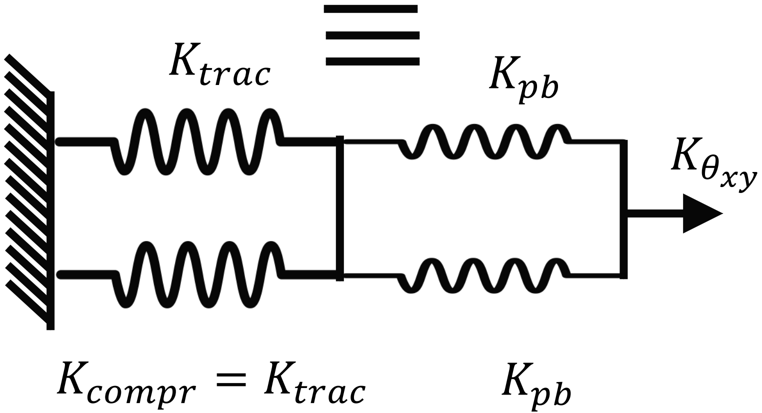
$$K_{trac\,\theta} = \left(\frac{C}{\theta} = \frac{F \cdot r}{z/r} = \right) K_{trac} \cdot r^2$$

$$r = \frac{D-b}{2}$$

$$K_z = \frac{Ebh}{L} \qquad K_{pb} = \frac{Ebh^3}{12L}$$

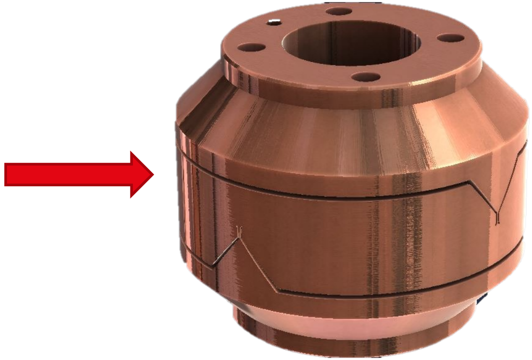
$$K_{\theta\,xy} = \frac{Ebh^3(D-b)^2}{L(2h^2 + 6(D-b)^2)}$$





Flexure optimization – example 2

- This correspond to a **design** with:
  - Very short leaf springs
  - Thin leaf springs
  - Large diameter
  - Medium inner diameter
- All **stiffness requirements** are fulfilled
- Optimal solution **parameters**:

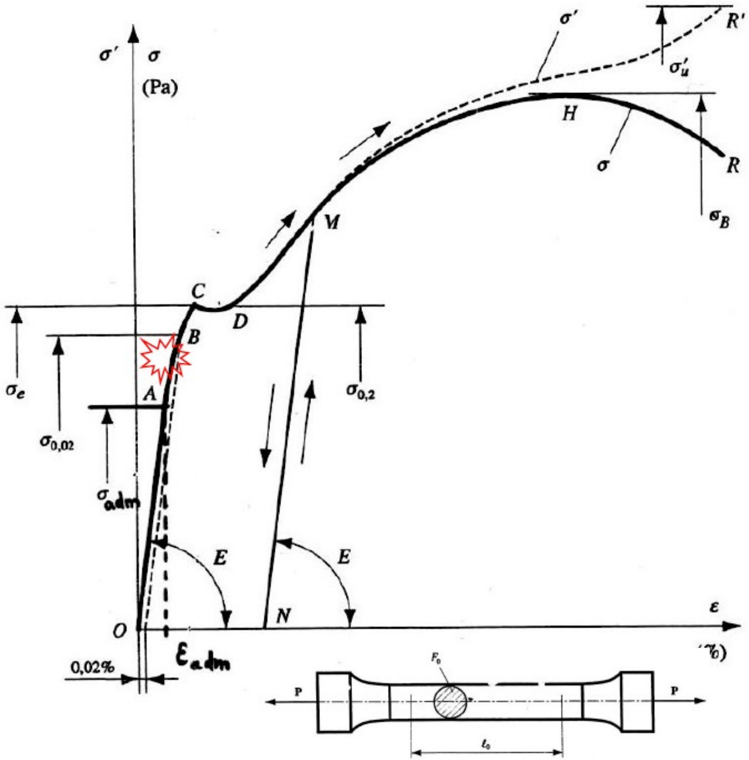


Property	Symbol	Value	Unit
Height	L	1	mm
Width	b	12.5	mm
Thickness	h	140	um
Diametre	D	41	mm

	Axis	Spec	Stiffness FEM	Unit	Ratio spec
MIN	x	1.00	5.35	MN/m	5.35
	y	1.00	5.35	MN/m	5.35
	z	12.00	63.92	MN/m	5.33
MAX	rx	1.15	0.96	Nm/rad	0.83
	ry	1.15	0.96	Nm/rad	0.83
MIN	rz	600	627	Nm/rad	1.05

Fatigue and material aspects

- The stroke of flexure is proportional to : $\frac{\sigma_{adm}}{E}$

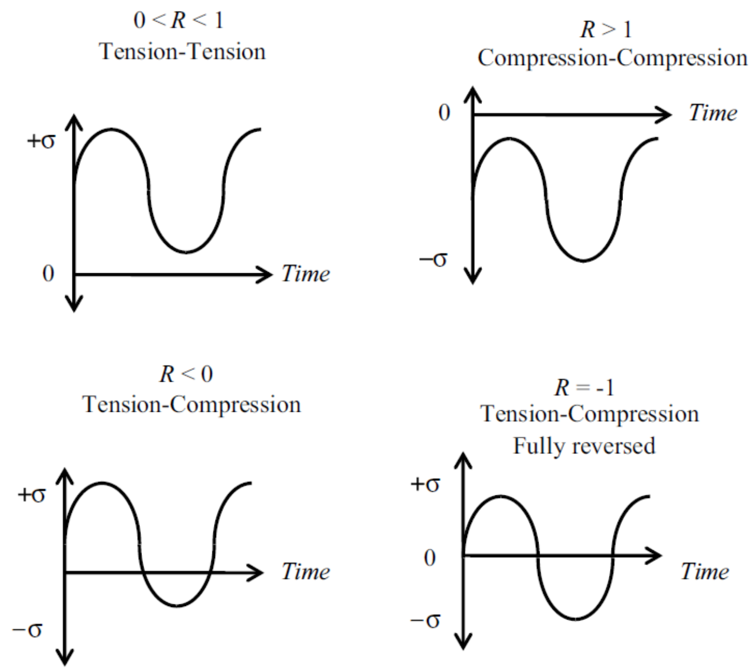


Good materials for flexures:

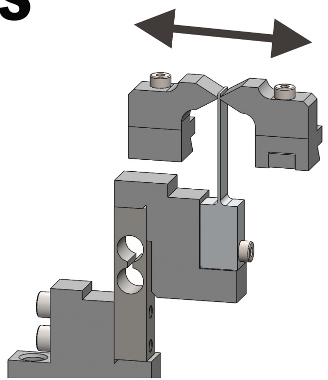
- Spring steels, carbon steels
  - e.g. X220CrVMo13-4 (K190)
  - X3NiCoMoTi18-9-5 (W720)
- Spring stainless-steels
  - e.g. X10CrNi18-8
  - N700 ESU (17-4 PH)
- Titanium alloys
  - e.g. Ti Al6 V4
- Copper-beryllium alloys
  - e.g. XHMS
- Bronze allows
  - e.g. Cu Ni 15 Sn 8
- Aluminum alloys
  - e.g. AlCuSiMn (Avional)
  - AlZnMgCu1.5 (Perunal)

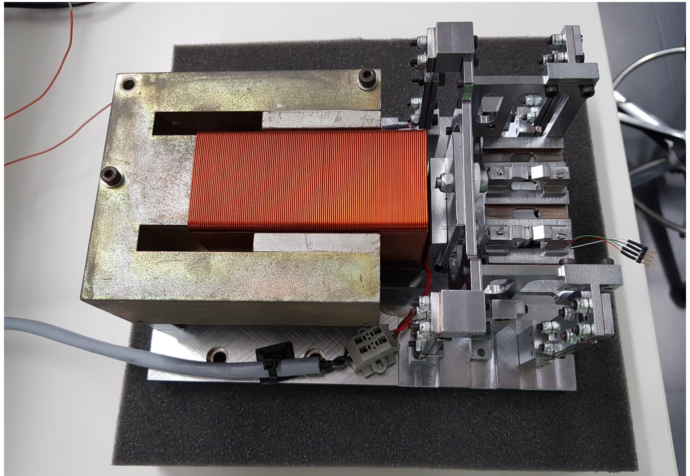
Fatigue and material aspects – tests

$R = \sigma_{min} / \sigma_{max}$ (R-1 is most common type for flexures)



J-L. González-Velázquez, Mechanical Behavior and Fracture of Engineering Materials, ISBN 978-3-030-29240-9



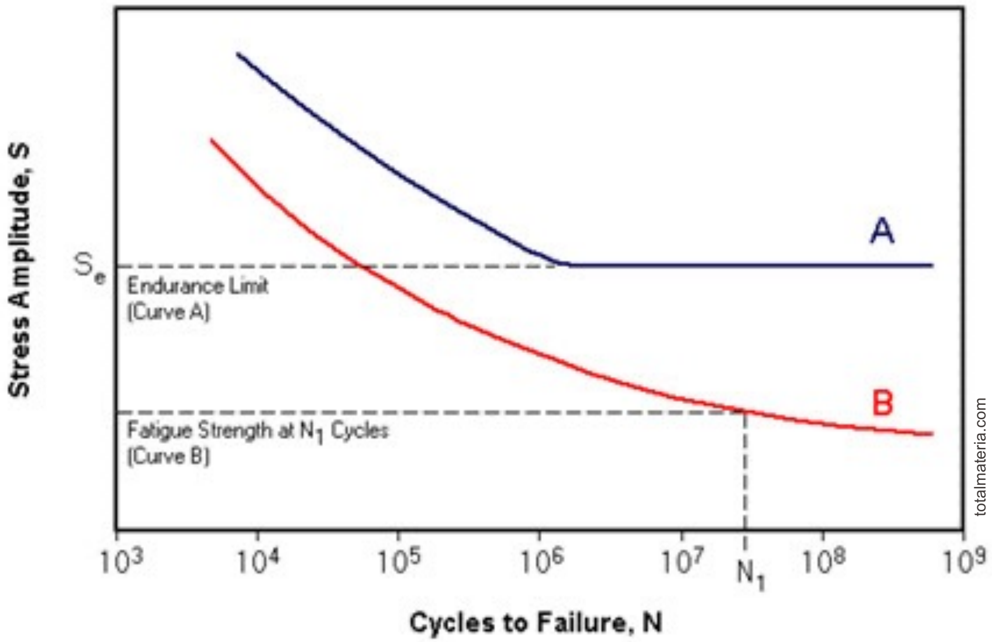


S. de Montmollin, Banc d'essai de fatigue en flexion alternée pour des applications spatiales

Fatigue and material aspects

- Examples of Wohler curves (S/N curve)
  - A has infinite lifetime (with fatigue plateau)
  - B has no infinite lifetime (without fatigue plateau)

☞ EXERCICE 7



Fatigue and material aspects - standards

- Failure probability and lifetime duration

ISO 3800
Threaded fasteners —
Axial load fatigue testing

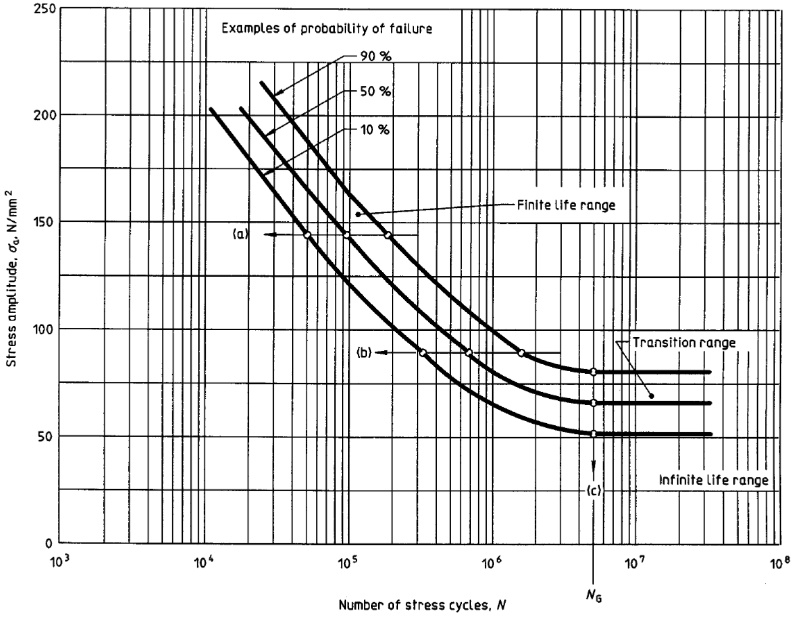
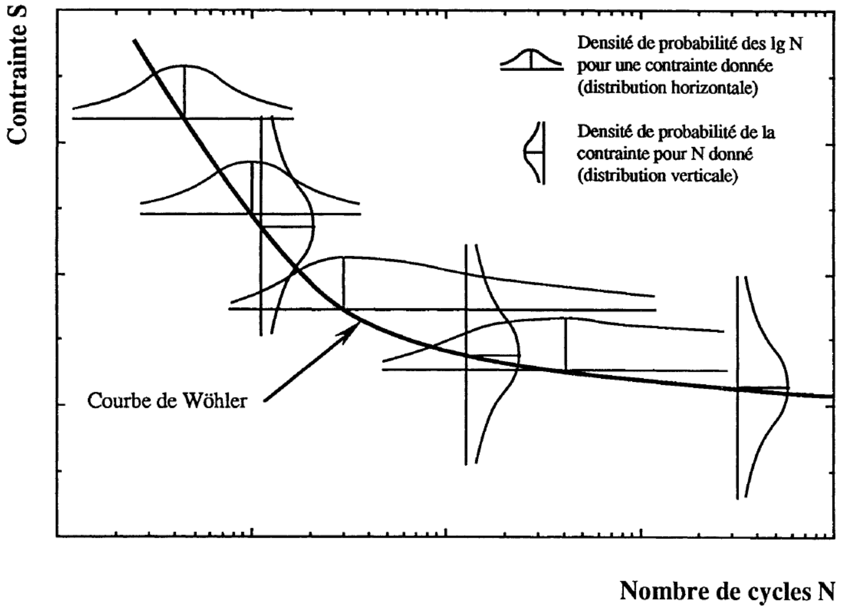


Figure 10 — Wöhler curve (S/N curve) $\sigma_a = f(N)$

AFNOR – A 03-405
Produits métalliques - Essais de fatigue -
Traitement statistique des données.



Week 6 exercises and homework

- Exercise
  - on MOODLE : EXO_6_Kinematic_analysis.pdf