Coupled MBS-FE Applications

A New Trend in Simulation

Michel Géradin
Emeritus Professor
University of Liège, Belgium

NAFEMS

Outline

- Introduction.
- General kinematics.
- Flexible MBS: a first series of examples.
- General formulation: variational approach, implicit solution methodology.
- Joint modeling: from functional to mesoscopic representation.
- Modal synthesis in multibody dynamics.
- Co-simulation.
- Conclusions.

Introduction

Coupled MBS – FE applications

A *new* trend in simulation

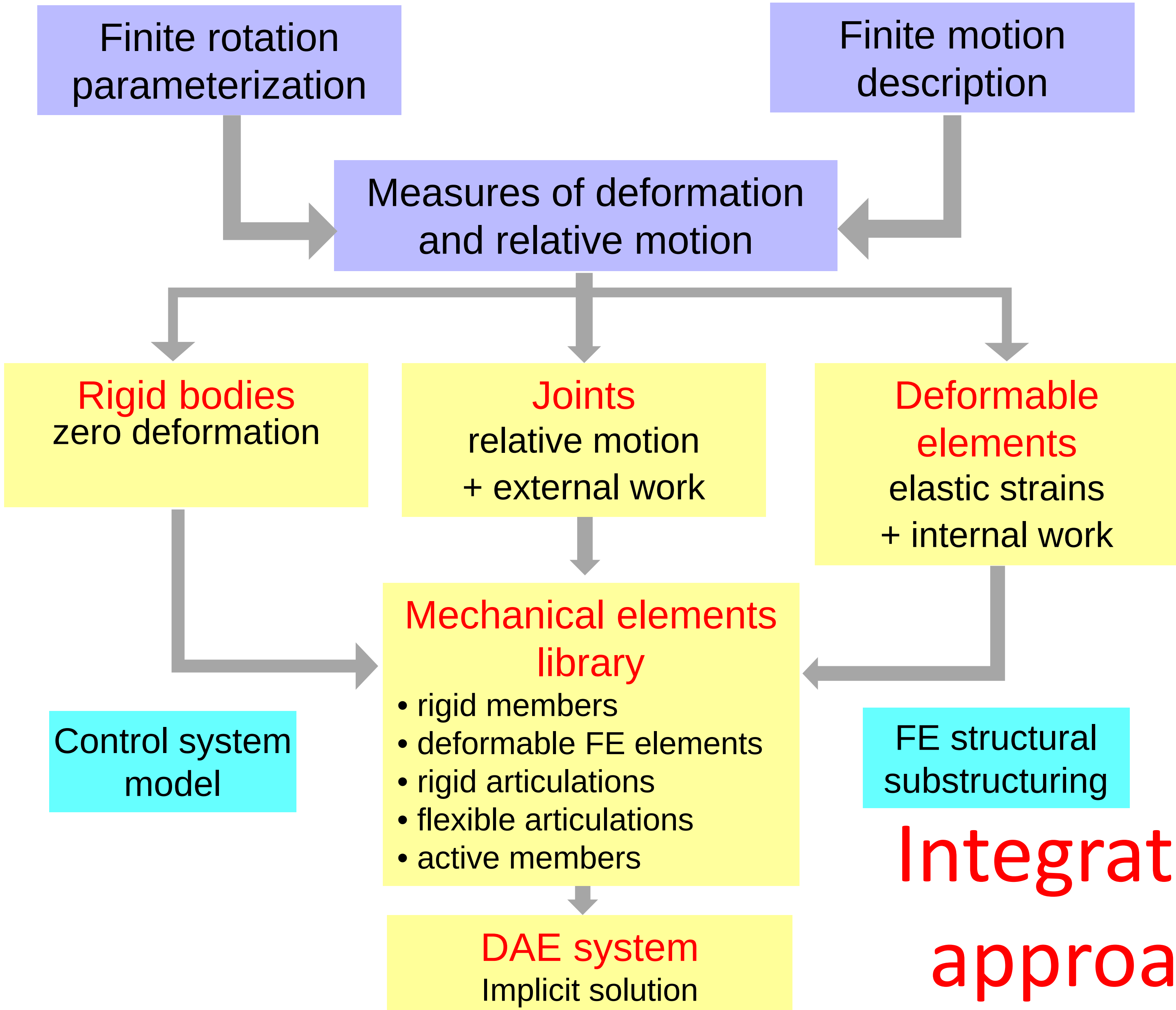
or

Integrated MBS – FE analysis

A *long-term* trend in simulation?

Some steps

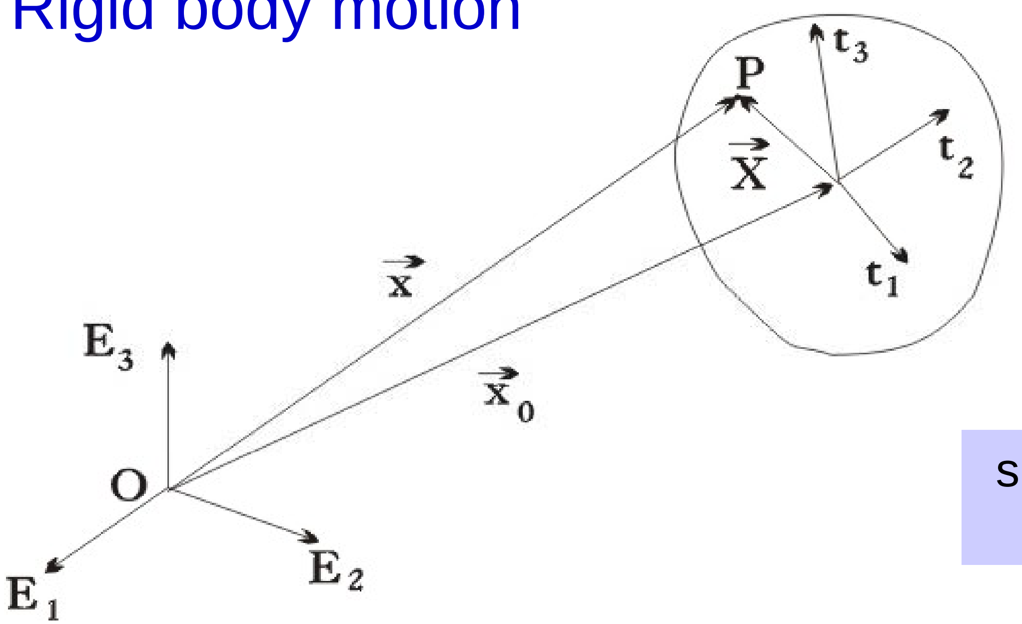
- 1983*: Computer Aided Analysis and Optimization of Mechanical System Dynamics.* NATO-ASI, University of Iowa. Organizer: Edward J. Haug.
- 1984: Géradin M. Finite element approach to kinematic and dynamic analysis of mechanisms using Euler parameters**.** In International Conference on Numerical Methods for Nonlinear Problems, Barcelona, Spain, 9-13 April 1984.
- 1991: A. Cardona. *An integrated approach to mechanism analysis*. PhD thesis. Université de Liège, Faculté des Sciences Appliquées.
- 2001: Géradin, M., & Cardona, A. *Flexible multibody dynamics: a finite element approach*. John Wiley.



Integrated approach

General kinematics

Rigid body motion



Combination of translation and rotation

$$\boldsymbol{x}_P = \boldsymbol{x}_0 + \boldsymbol{R}\boldsymbol{X}_P$$

spatial motion of origin

material coordinates of point P

Cartesian position and orientation of point P: are thus uniquely described by

$$(\boldsymbol{x}_P,\ \boldsymbol{R}_P)$$

Rotation parameterization

Orthonormality of $\boldsymbol{R}$ $\boxed{\boldsymbol{R}^T = \boldsymbol{R}^{-1}}$

6 constraints → 3 independent parameters

$\boxed{\boldsymbol{a}^T = \begin{bmatrix}\alpha_1 & \alpha_2 & \alpha_3\end{bmatrix}}$ ⟺ $\boxed{\boldsymbol{R} = \boldsymbol{R}(\boldsymbol{a})}$

Possible choices:

- ✓ Euler angles
- ✓ Bryant angles
- ✓ Rodrigues parameters
- ✓ Euler Parameters
- ✓ Cartesian rotation vector
- ✓ Conformal rotation vector
- ✓ …

Choice criteria:

- ✓ analytical complexity.
- ✓ singular configurations / their handling
- ✓ redundancy (e.g. Euler parameters: 4 → 1 constraint) .
- ✓ Composition rule (successive rotations).
- ✓ Differentiation (angular velocities, virtual displacements)
- ✓ Physical meaning

Infinitesimal motion velocity analysis

$$\boldsymbol{x}_P = \boldsymbol{x}_0 + \boldsymbol{R}\boldsymbol{X}_P$$

infinitesimal displacements

Infinitesimal rotations

$$\delta\boldsymbol{x}_P = \delta\boldsymbol{x}_0 + \delta\tilde{\boldsymbol{\theta}}(\boldsymbol{x}_P - \boldsymbol{x}_0)$$

$$\boldsymbol{v}_P = \boldsymbol{v}_0 + \tilde{\boldsymbol{\omega}}(\boldsymbol{x}_P - \boldsymbol{x}_0)$$

Linear velocities

Angular velocities

with

$$\delta\tilde{\boldsymbol{\theta}} = \delta\boldsymbol{R}\boldsymbol{R}^T$$

$$\tilde{\boldsymbol{\omega}} = \dot{\boldsymbol{R}}\boldsymbol{R}^T$$

In terms of Cartesian rotation vector:

$$\delta\boldsymbol{\theta} = \boldsymbol{T}^T(\boldsymbol{\Psi})\delta\boldsymbol{\Psi}$$

$$\boldsymbol{\omega} = \boldsymbol{T}^T(\boldsymbol{\Psi})\dot{\boldsymbol{\Psi}}$$

Tangent operator $\boldsymbol{T}(\psi)$: analytical expression similar to $\boldsymbol{R}(\psi)$

Kinematics summary

- General description of current point P motion made in Lagrangian form with the following parameters:
  - Absolute position vector $\boldsymbol{x}$
  - Cartesian rotation vector ψ
- Nonlinear mapping of angular quantities provides *geometrically exact* formalism:

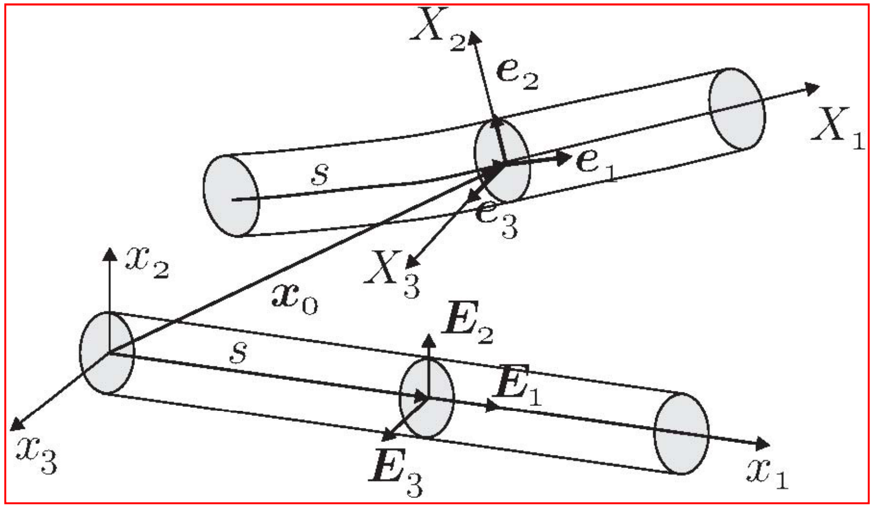
rotations:	$\boldsymbol{R} \leftrightarrow \boldsymbol{\psi}$
angular velocities:	$\boldsymbol{\omega} \leftrightarrow \boldsymbol{T}^T(\boldsymbol{\Psi})\dot{\boldsymbol{\Psi}}$
angular variations:	$\delta\boldsymbol{\theta} \leftrightarrow \boldsymbol{T}^T(\boldsymbol{\Psi})\delta\boldsymbol{\Psi}$

- Same formalism is applied to any MBS component (rigid bodies, joints, nonlinear 1D/2D/3D deformable members, superelements, subassemblies)
- Updated Lagrangian description of rotations is generally adopted.

Example: «geometrically exact» beam

Kinematic assumptions

- Beam is initially straight
- Cross sections remain plane and do not deform
- Shear deformation of neutral axis allowed
- Rotational energy of cross sections taken into account



Strains

Stress-strain relationships

Local equilibrium

Virtual work expression

Strain measures

Obtained from position gradients along s before and after deformation

$$\boldsymbol{D}(s, X_2, X_3) = \boldsymbol{R}^T \frac{d\boldsymbol{x}}{ds} - \frac{d\boldsymbol{X}}{ds} \Rightarrow \boxed{\boldsymbol{D}(s, X_2, X_3) = \underbrace{\boldsymbol{R}^T \left(\frac{d\boldsymbol{x}_0}{ds} - \boldsymbol{e}_1 \right)}_{\text{Neutral axis}} + \underbrace{\boldsymbol{R}^T \frac{d\boldsymbol{R}}{ds} \boldsymbol{Y}}_{\text{Cross section}}}$$

Deformation of neutral axis

$$\boxed{\boldsymbol{\Gamma}(s) = \boldsymbol{R}^T \left(\frac{d\boldsymbol{x}_0}{ds} - \boldsymbol{e}_1 \right)}$$

$$\Gamma_1 = \boldsymbol{e}_1^T \frac{d\boldsymbol{x}_0}{ds} - 1$$ Extension of neutral axis

$$\Gamma_i = \boldsymbol{e}_i^T \frac{d\boldsymbol{x}_0}{ds} \quad (i = 2, 3)$$ Shear strains

Material measures

$$\boxed{\boldsymbol{D} = \boldsymbol{\Gamma} - \tilde{\boldsymbol{Y}} \boldsymbol{K}}$$

Measure of curvature

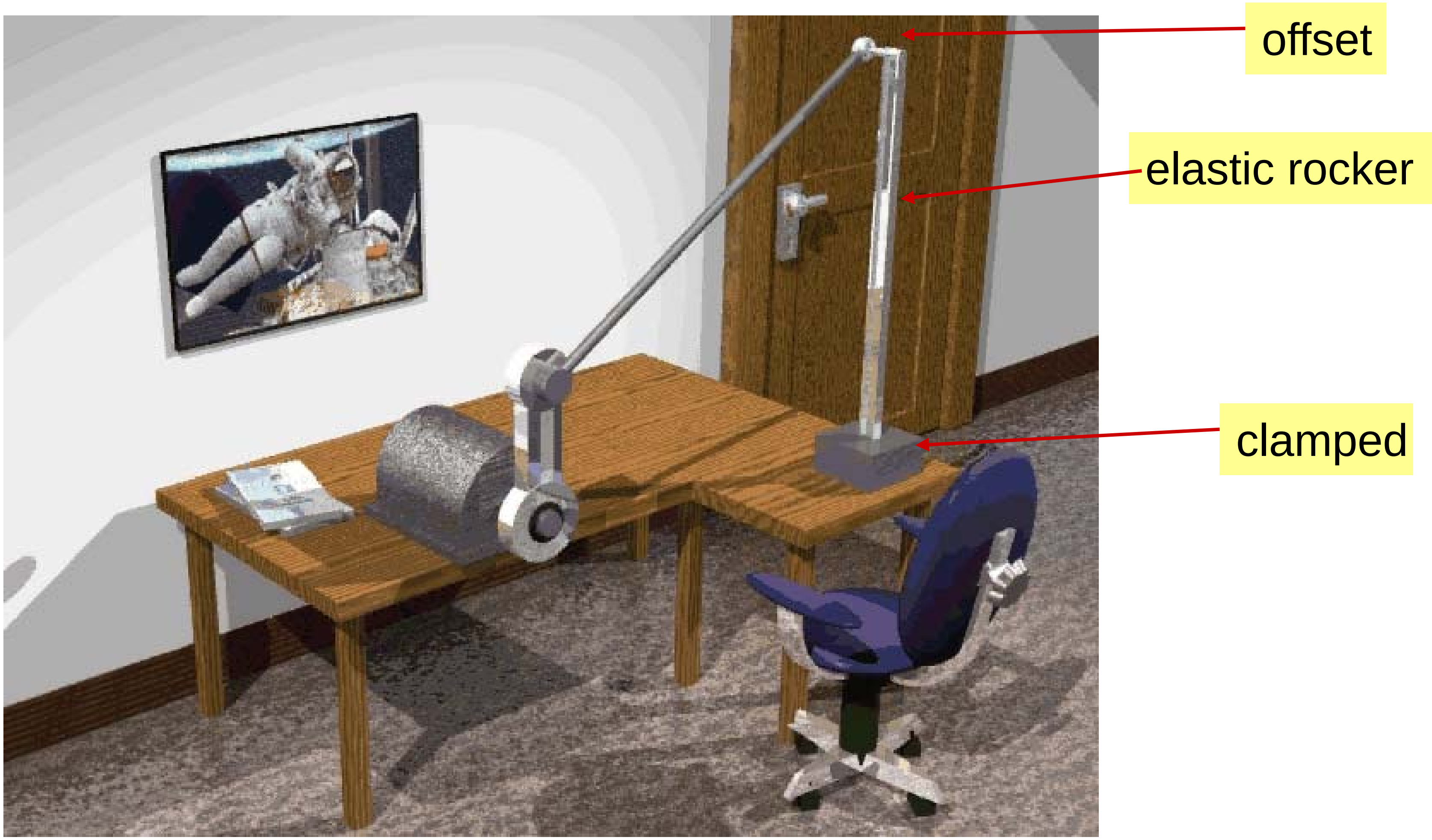
$$\boxed{\tilde{\boldsymbol{K}} = \boldsymbol{R}^T \frac{d\boldsymbol{R}}{ds}}$$ Curvature matrix

$$\boxed{\boldsymbol{K} = \boldsymbol{T}(\boldsymbol{\Psi}) \frac{d\boldsymbol{\Psi}}{ds}}$$ Curvature vector

K_1 torsion of neutral axis

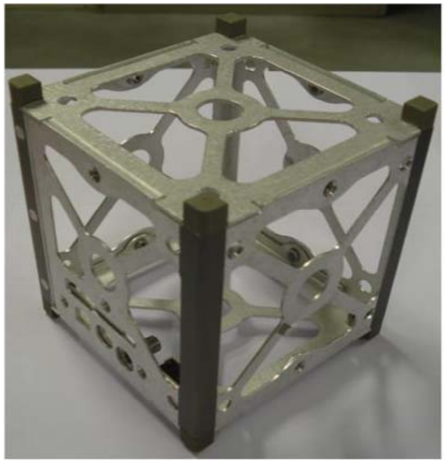
K_2, K_3 bending curvatures

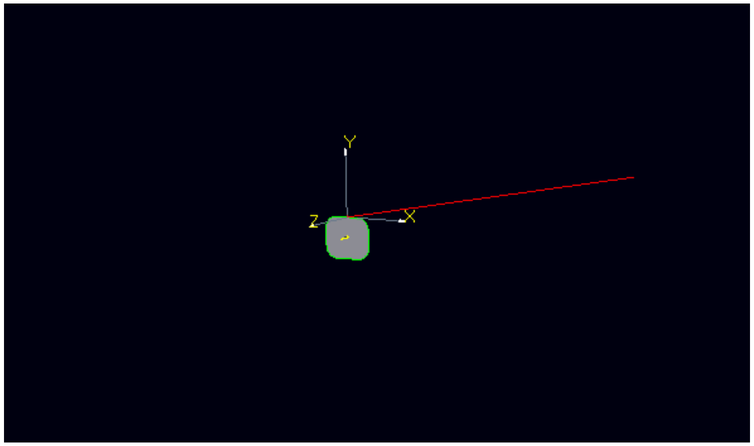
Elastic 4-bar mechanism

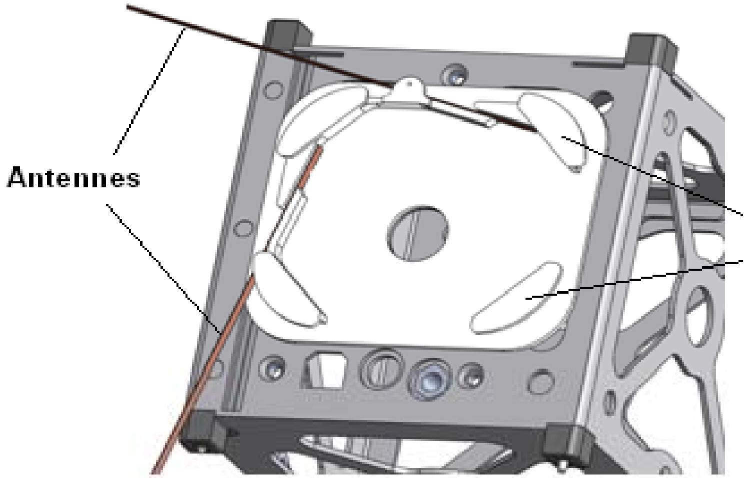


OUFTI-1 satellite

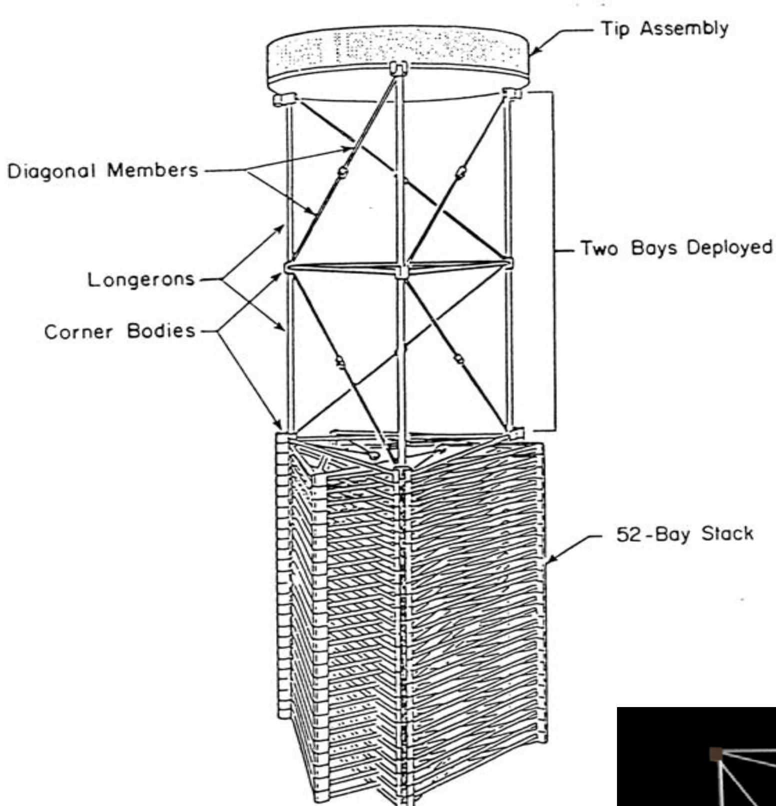
- Student CubeSat designed in Liège
- 10x10x10 cm
- 1 kg
- D-STAR amateur-radio
- digital-communication protocol
- Two deployable antennas



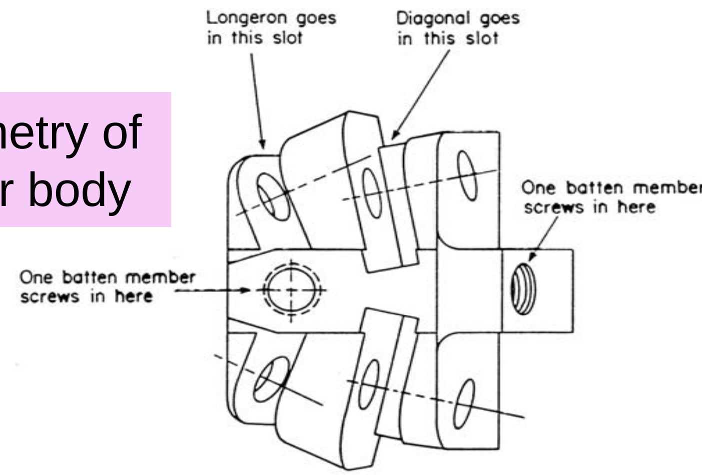




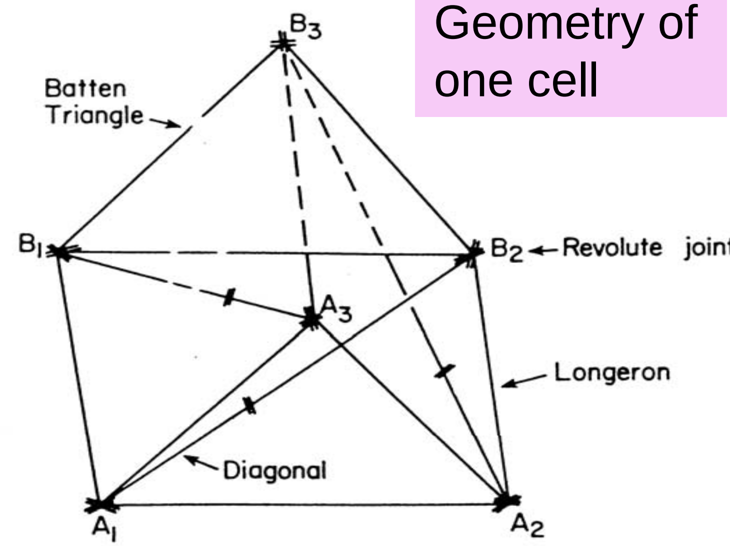
Deployment of astromast cell



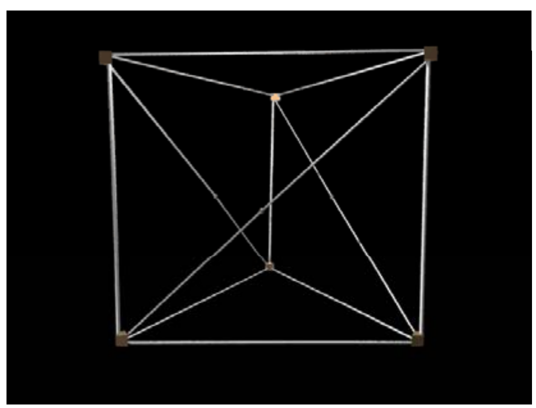
Geometry of corner body



Geometry of one cell



Stacking method: 2 bays deployed

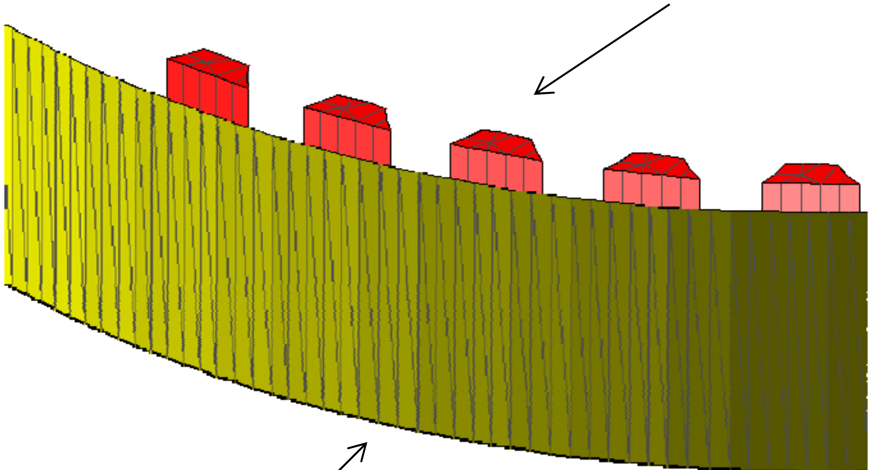


Modeling of transmission belt

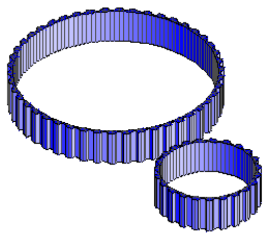
Structural hypotheses

Belt: rubber

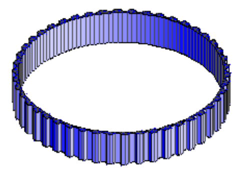
Volume elements

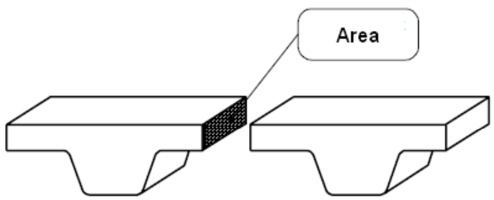


Membrane elements



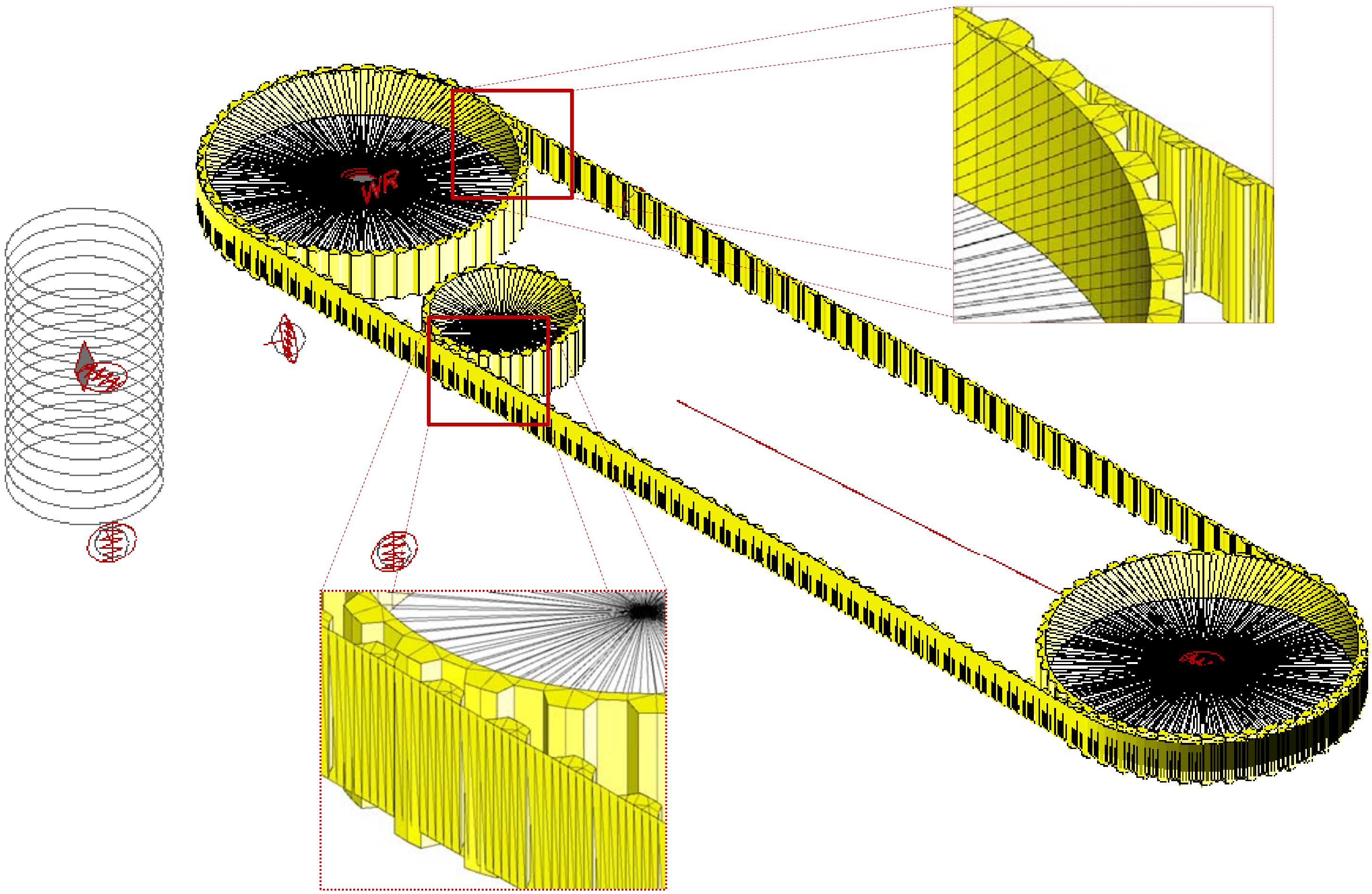
Pulleys and
sprocket: steel



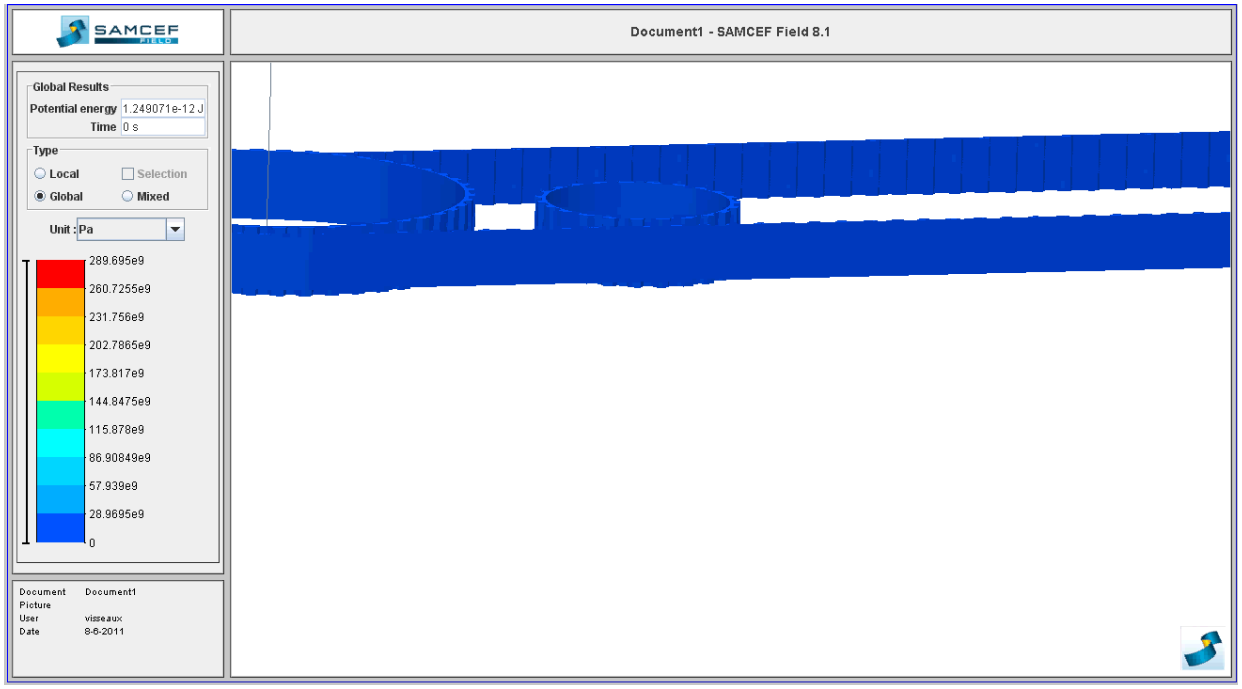


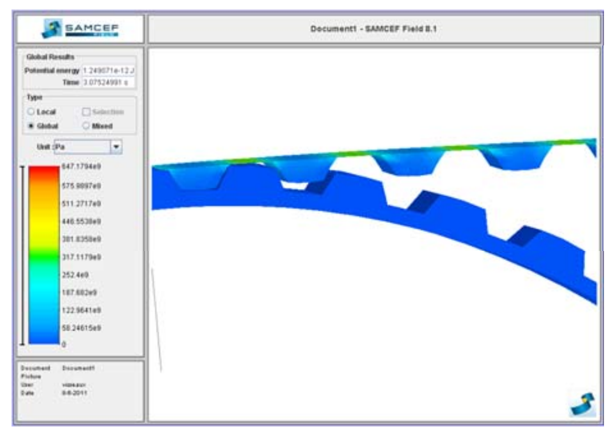
Modelling of transmission belt

- Finite element mesh: 7500 nodes, 14600 elements.



Dynamics of transmission belt





General formulation
Variational approach

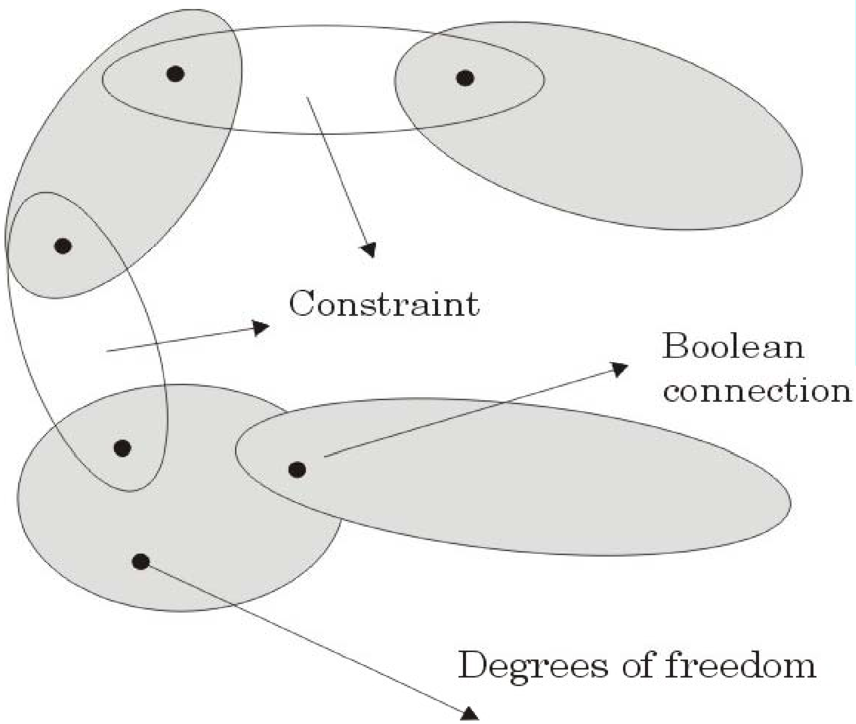
$$\delta \int_{t_1}^{t_2} \left(\mathcal{K} - \mathcal{W} - \mathcal{P} - k\boldsymbol{\lambda}^T \boldsymbol{\Phi} - \frac{p}{2} \boldsymbol{\Phi}^T \boldsymbol{\Phi} \right) dt = \boldsymbol{f}_{nc}^T \delta \boldsymbol{q}$$

- $\boldsymbol{q}$ Generalized coordinates.
- $\mathcal{K}(\boldsymbol{q}, \dot{\boldsymbol{q}})$ Kinetic energy.
- $\mathcal{W}(\boldsymbol{q})$ Strain energy.
- $\mathcal{P}(\boldsymbol{q})$ Potential energy of external loads.
- $\boldsymbol{\Phi}(\boldsymbol{q}, t)$ Kinematic constraints.
- $\boldsymbol{\lambda}$ Lagrange multipliers.
- $\boldsymbol{f}_{nc}$ Non-conservative forces.
- p Penalty coefficient.
- k Scaling coefficient.

Summation over elements

Augmented Lagrangian method with scaling $\rightarrow$ improved robustness of linear solution

System assembly



$\boldsymbol{q}$ = DOF at structural level

$\boldsymbol{q}_e$ = DOF at element level

$\boldsymbol{L}_e$ = DOF localization operator(boolean)

$\boldsymbol{\lambda}$ = lagrangian multipliers

- Boolean constraints: localization of DOF
- Implicit constraints: algebraic treatment

$$\boldsymbol{q_e} = \boldsymbol{L_e q}$$

$$\boldsymbol{\Phi}(\boldsymbol{q_e}, \dot{\boldsymbol{q}}_{\boldsymbol{e}}, t) = 0$$

System topology results implicitly from Boolean assembly and constraint description

Dynamic equilibrium

- Formulation using the Augmented Lagrangian method

$$\begin{cases} \boldsymbol{M}\ddot{\boldsymbol{q}} + \boldsymbol{B}^T(k\boldsymbol{\lambda} + p\boldsymbol{\Phi}) = \boldsymbol{g}(\boldsymbol{q}, \dot{\boldsymbol{q}}, t) \\ p\boldsymbol{\Phi}(\boldsymbol{q}, t) = 0 \end{cases}$$

  - M constant with respect to translation DOF.
  - Exact treatment of constraints
  - Well-conditioned iteration matrix

- Implicit / iterative Solution → recast in residual form

$$\begin{cases} \boldsymbol{r}(\boldsymbol{q}, \dot{\boldsymbol{q}}, \ddot{\boldsymbol{q}}, \boldsymbol{\lambda}) = \boldsymbol{g}(\boldsymbol{q}, \dot{\boldsymbol{q}}, t) - \boldsymbol{M}\ddot{\boldsymbol{q}} - \boldsymbol{B}^T(k\boldsymbol{\lambda} + p\boldsymbol{\Phi}) = 0 \\ p\boldsymbol{\Phi}(\boldsymbol{q}, t) = 0 \end{cases}$$

Implicit time integration
Newmark type methods

- One-step methods $\rightarrow$ obey to the general form

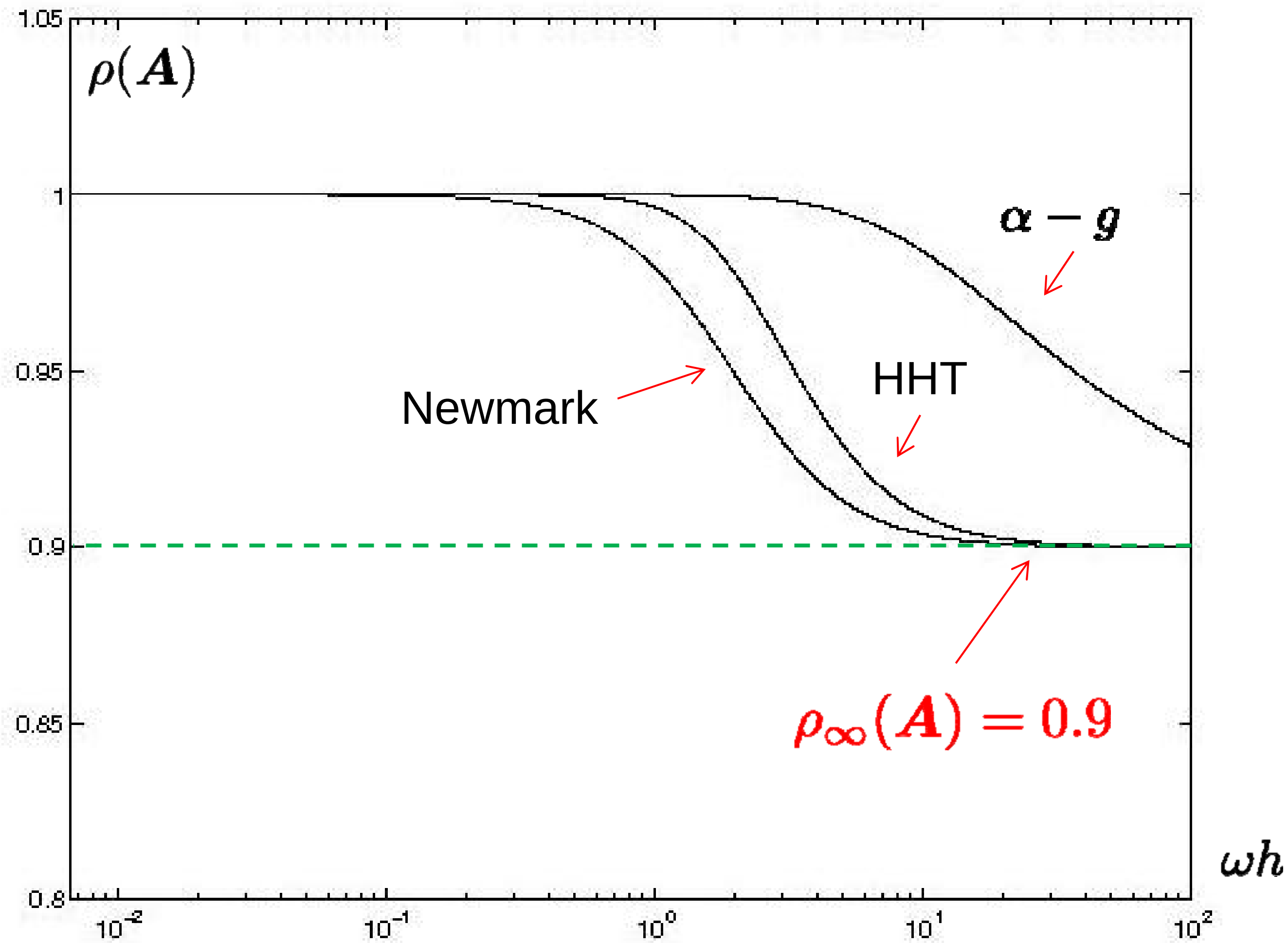
$$\boxed{\begin{bmatrix} \boldsymbol{q}_{n+1} \\ \dot{\boldsymbol{q}}_{n+1} \\ \dot{\boldsymbol{q}}_{n+1} \end{bmatrix} = \boldsymbol{A} \begin{bmatrix} \boldsymbol{q}_{n} \\ \dot{\boldsymbol{q}}_{n} \\ \dot{\boldsymbol{q}}_{n} \end{bmatrix}}$$

- Can provide *unconditional stability over entire frequency range*.
- Accuracy /stability properties adjusted through free parameters.
    - Second-order accuracy can be achieved.
    - Stability: governed by *spectral radius* = highest eigenvalue of amplification matrix $\boldsymbol{A}$.

$$\boxed{\rho(\boldsymbol{A}) \leq 1}$$

(Guarantees stability of index-3 DAE system).

Spectral radius of Newmark type methods



Modified α–g method

- Newmark interpolation of displacements and velocities

$$\dot{\boldsymbol{q}}_{n+1} \simeq \dot{\boldsymbol{q}}_n + (1-\gamma)h\boldsymbol{a}_n + \gamma h \boldsymbol{a}_{n+1}$$
$$\boldsymbol{q}_{n+1} \simeq \boldsymbol{q}_n + h\dot{\boldsymbol{q}}_n + \left(\frac{1}{2}-\beta\right)h^2\boldsymbol{a}_n + \beta h^2 \boldsymbol{a}_{n+1}$$

- Pseudo-accelerations $\boldsymbol{a}_n$ defined through collocation

$$(1-\alpha_m)\boldsymbol{a}_{n+1} + \alpha_m \boldsymbol{a}_n = (1-\alpha_f)\ddot{\boldsymbol{q}}_{n+1} + \alpha_f \ddot{\boldsymbol{q}}_n$$

- Residual form of equilibrium

$$\begin{cases} \boldsymbol{r}(\boldsymbol{q}_{n+1}, \dot{\boldsymbol{q}}_{n+1}, \ddot{\boldsymbol{q}}_{n+1}, \boldsymbol{\lambda}) = 0 \\ \boldsymbol{\Phi}(\boldsymbol{q}_{n+1}, t_{n+1}) = 0 \end{cases} \quad \rightarrow \quad \begin{cases} \boldsymbol{r}(\boldsymbol{q}_{n+1}, \boldsymbol{\lambda}) = 0 \\ \boldsymbol{\Phi}(\boldsymbol{q}_{n+1}) = 0 \end{cases}$$

- 4 parameters $(\beta,\ \gamma,\ \alpha_f,\ \alpha_m)$

Nonlinear system to be solved iteratively

Equilibrium iteration

- Linear solution step

$$\boxed{\begin{bmatrix} \boldsymbol{S} & k\boldsymbol{B}^T \\ k\boldsymbol{B} & 0 \end{bmatrix} \begin{bmatrix} d\boldsymbol{q} \\ d\boldsymbol{\lambda} \end{bmatrix} = \begin{bmatrix} \boldsymbol{r}(\boldsymbol{q}^\star, \boldsymbol{\lambda}^\star) \\ -k\boldsymbol{\Phi}(\boldsymbol{q}^\star) \end{bmatrix}}$$

with matrix $\boldsymbol{S}$ linear combination (coefficients depending upon integration scheme) of matrices

$$\boldsymbol{M},\ \frac{\partial \boldsymbol{g}}{\partial \boldsymbol{q}},\ \frac{\partial \boldsymbol{g}}{\partial \dot{\boldsymbol{q}}},\ p\boldsymbol{B}\boldsymbol{B}^T + \frac{\partial \boldsymbol{B}^T}{\partial \boldsymbol{q}}(k\boldsymbol{\lambda} + p\boldsymbol{\Phi})$$

- *Sparsity linked to FE mesh topology.*
- Can be solved using standard linear solver for large sparse systems.
- The $\boldsymbol{\lambda}$ can be assimilated to ordinary DOF

$$\boxed{\boldsymbol{S}d\boldsymbol{q} = \boldsymbol{r}(\boldsymbol{q}^\star)}$$

Time step size control

- Drawbacks due to use of predefined, fixed time step size:
  - Time step too large: higher frequencies filtered out → loss of detail in the response.
  - Time step too small: increase of computer time, waste of resources.
- Time step adjstment during numerical integration can be based on a measure of integration error computed from

$$\boxed{e = \left(\beta - \frac{1}{6}\right) h^2 \Delta \ddot{\boldsymbol{q}}}$$

- Adequate strategy should:
  - Avoid exceeding prescribed tolerance.
  - Keep time step fixed for sufficiently long periods.

Joint modeling

Level 1*: functional description*

- purely kinematic model
- description by algebraic constraints
- rigid behavior
- « perfect » interaction

Level 2*: macroscopic engineering description*

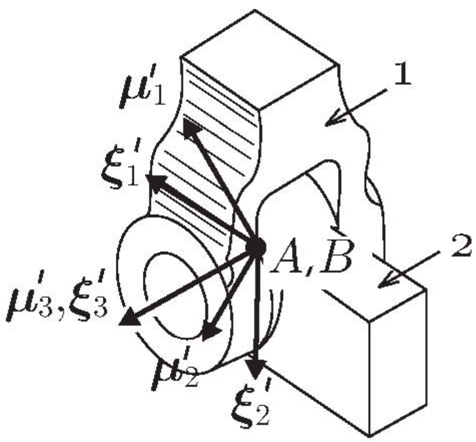
- semi-rigid behavior
- partial description by algebraic constraints.
- Addition of global constitutive laws (e.g. compliance, friction).

Level 3*: detailed / mesoscopic description*

- 3D geometric representation
- Account of structural detailing
- Finite element modeling

Revolute joint: functional description

Parts A and B



2 rotation constraints

$$\boldsymbol{\mu}_1'^T \boldsymbol{\xi}_3' = 0 \text{ and } \boldsymbol{\mu}_2'^T \boldsymbol{\xi}_3' = 0$$

$$\Phi_1 = \boldsymbol{\mu}_1^T \boldsymbol{R}_A^T \boldsymbol{R}_B \boldsymbol{\xi}_3 = 0$$
$$\Phi_2 = \boldsymbol{\mu}_2^T \boldsymbol{R}_A^T \boldsymbol{R}_B \boldsymbol{\xi}_3 = 0$$

$$\delta \boldsymbol{\Phi} = \boldsymbol{B} \delta \boldsymbol{q}$$

3 position constraints

$$\boldsymbol{x}_B = \boldsymbol{x}_A$$

Boolean

$$\boldsymbol{B} = \left[\frac{\partial \Phi_i}{\partial q_j} \right]$$

$\{\boldsymbol{\mu}_1', \boldsymbol{\mu}_2', \boldsymbol{\mu}_3'\}$ $\rightarrow$ Base vectors of body A - reference configuration

$\{\boldsymbol{\xi}_1', \boldsymbol{\xi}_2', \boldsymbol{\xi}_3'\}$ $\rightarrow$ Base vectors of body B - reference configuration

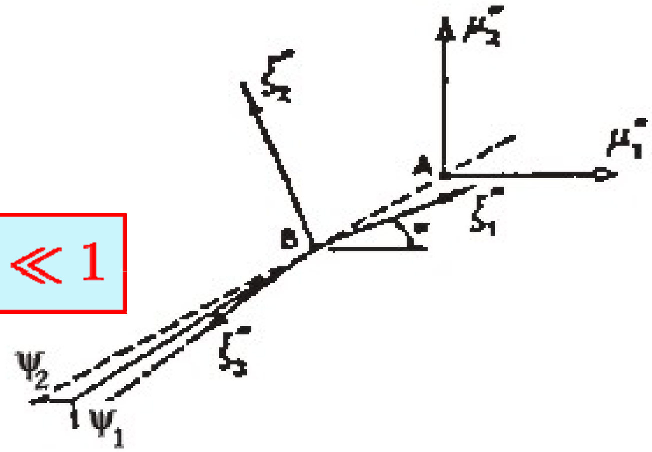
$\boldsymbol{\mu}_i' = \boldsymbol{R}_A \boldsymbol{\mu}_i$ $\rightarrow$ Base vectors of body A - current configuration

$\boldsymbol{\xi}_i' = \boldsymbol{R}_B \boldsymbol{\xi}_i$ $\rightarrow$ Base vectors of body B - current configuration

Revolute joint: level 2 description

angular deformations

$$\mu_1'^T \xi_3' = \sin\psi_2 \simeq \psi_2$$
$$\mu_2'^T \xi_3' = -\sin\psi_1 \simeq -\psi_1$$



translation deformations

$$\boldsymbol{u} = \boldsymbol{R}_A^T(\boldsymbol{x}_B - \boldsymbol{x}_A) \qquad \|\boldsymbol{u}\| \ll 1$$

Strain energy

$$\mathcal{W} = \frac{1}{2}\boldsymbol{u}^T \boldsymbol{H}_u \boldsymbol{u} + \frac{1}{2}\boldsymbol{\psi}^T \boldsymbol{H}_\psi \boldsymbol{\psi}$$

elastic coefficients

Internal forces

Tangent stiffness

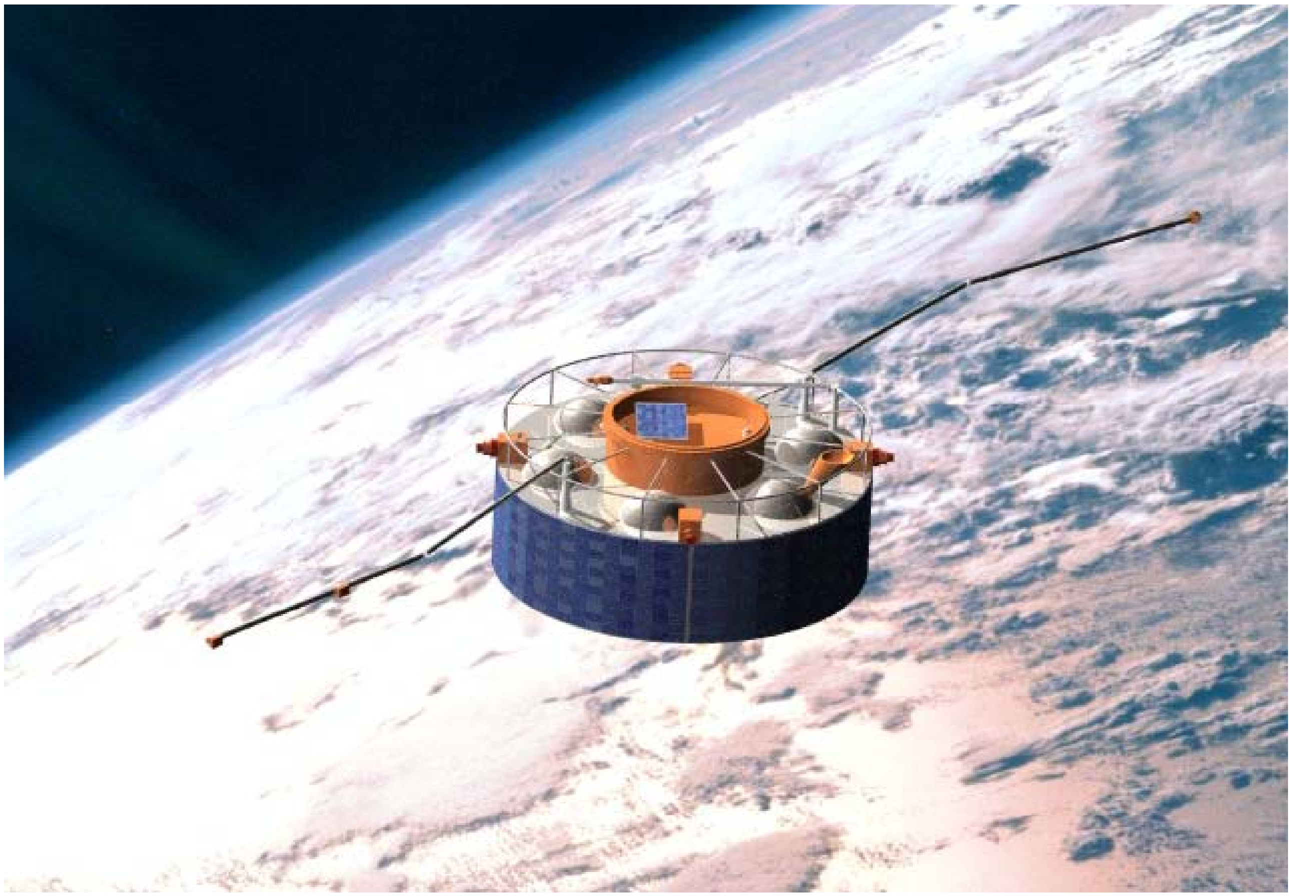
$$\boldsymbol{g}_{int} = -\frac{\partial \mathcal{W}}{\partial q_j}$$

Specialized hinges for deployment of space structures

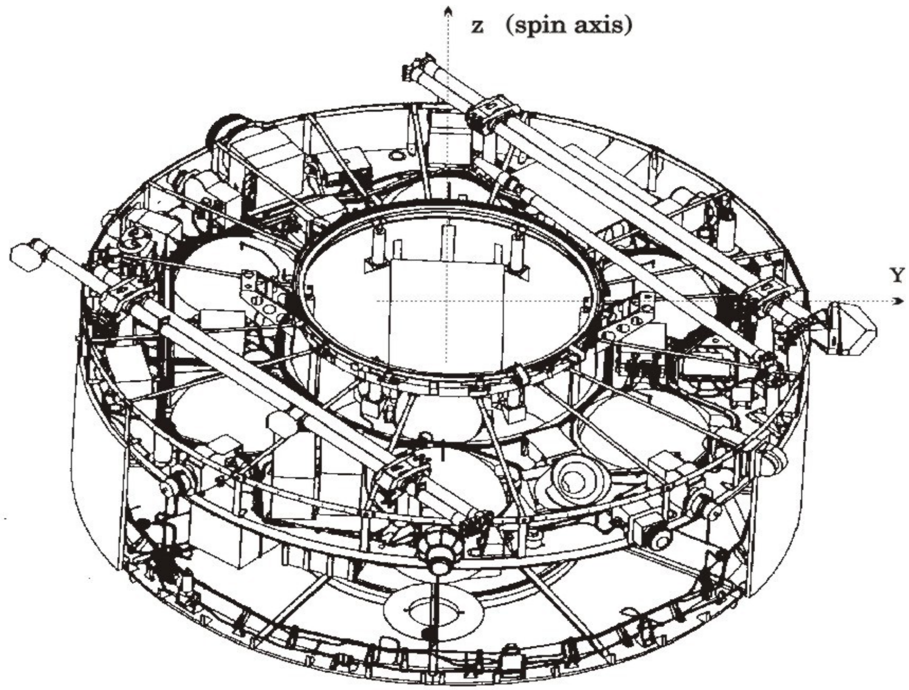
Desirable characteristics

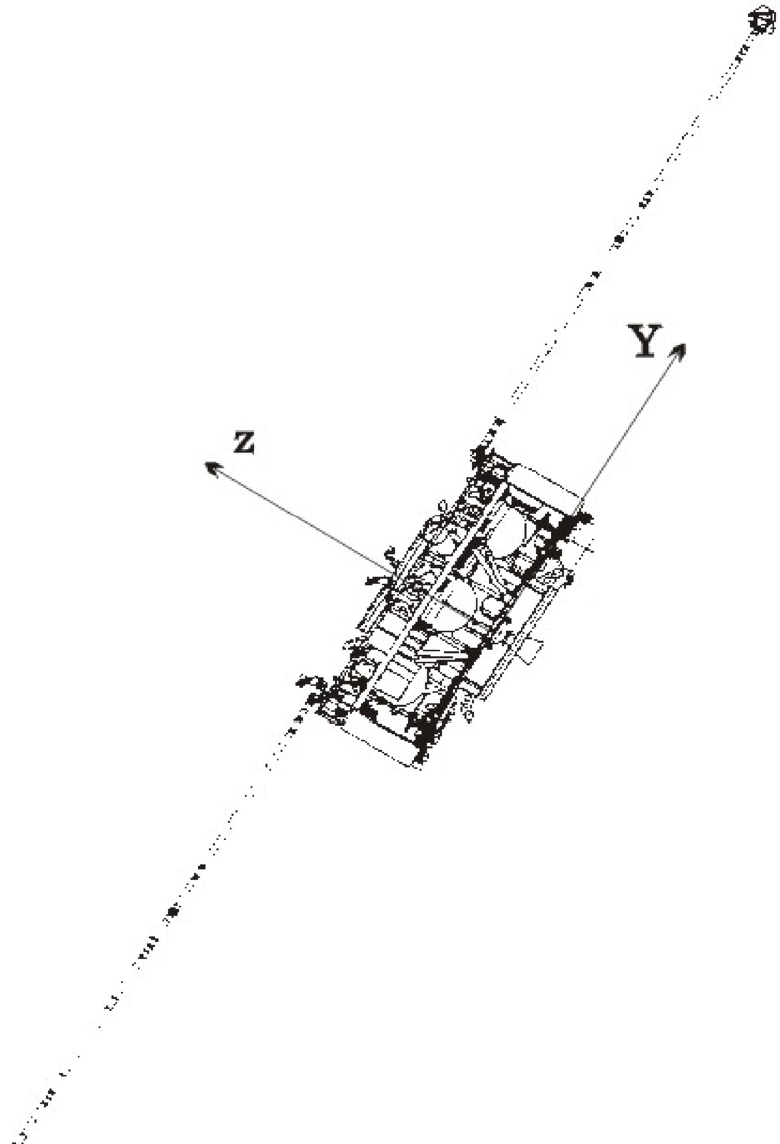
- Passive motorization (energy release)
  - Adequate torque / rotation relationship.
  - Auto-locking.
- High transverse stiffness
- High precision motion → minimum backlash and friction
- High stiffness in folded and locked configurations.
- Minimum size.
- Well-characterized stiffness and damping properties.

Simulation of Cluster satellite boom deployment



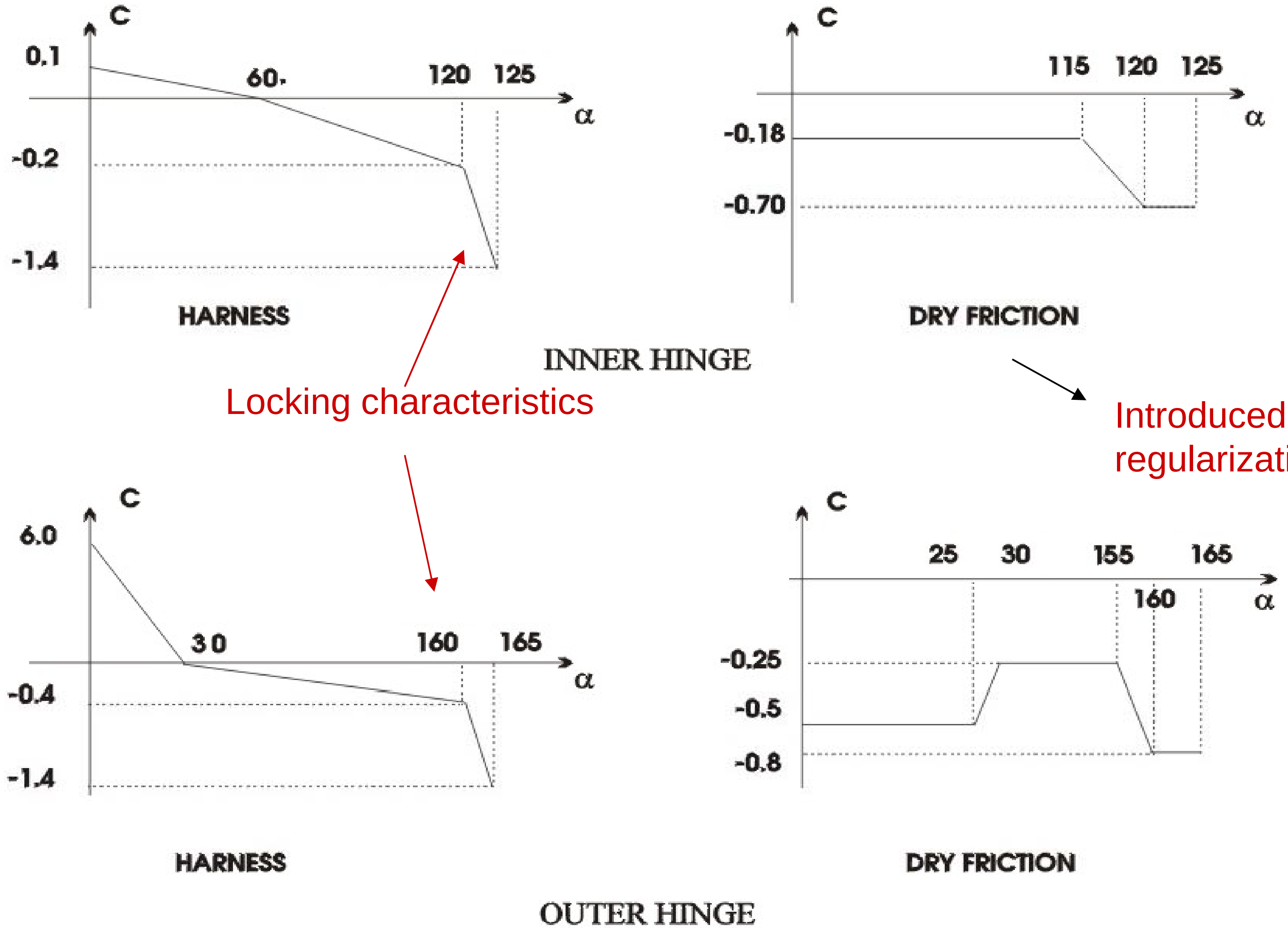
Boom design





- 2 folded booms
- spring actuators
- harnesses

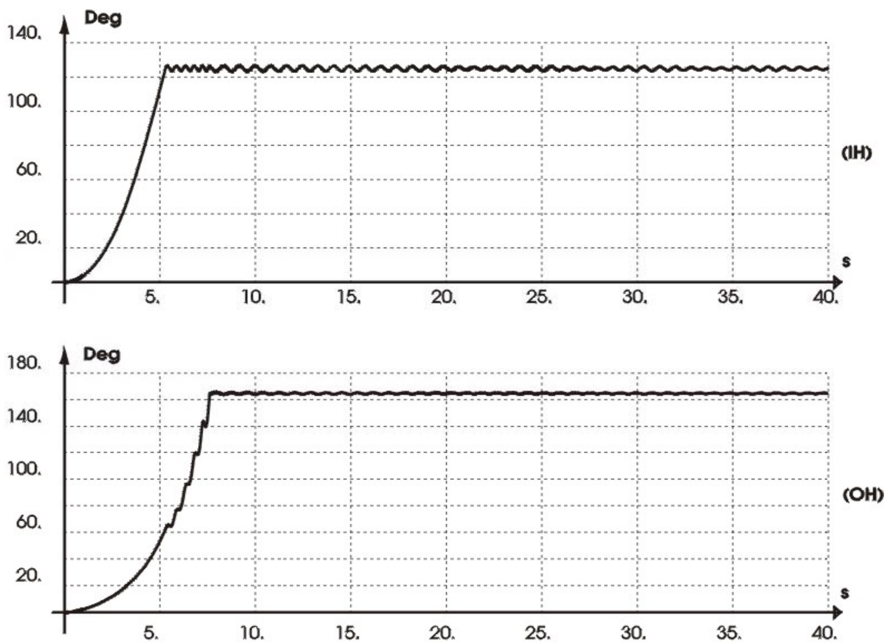
Motorized hinge characteristics



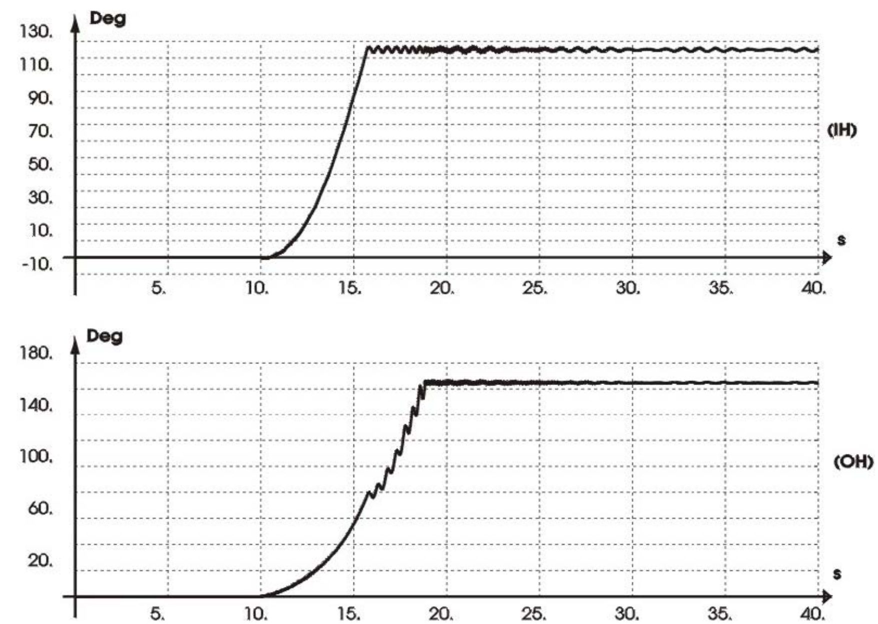
Locking characteristics

Introduced through regularization

Deployment with flexible booms



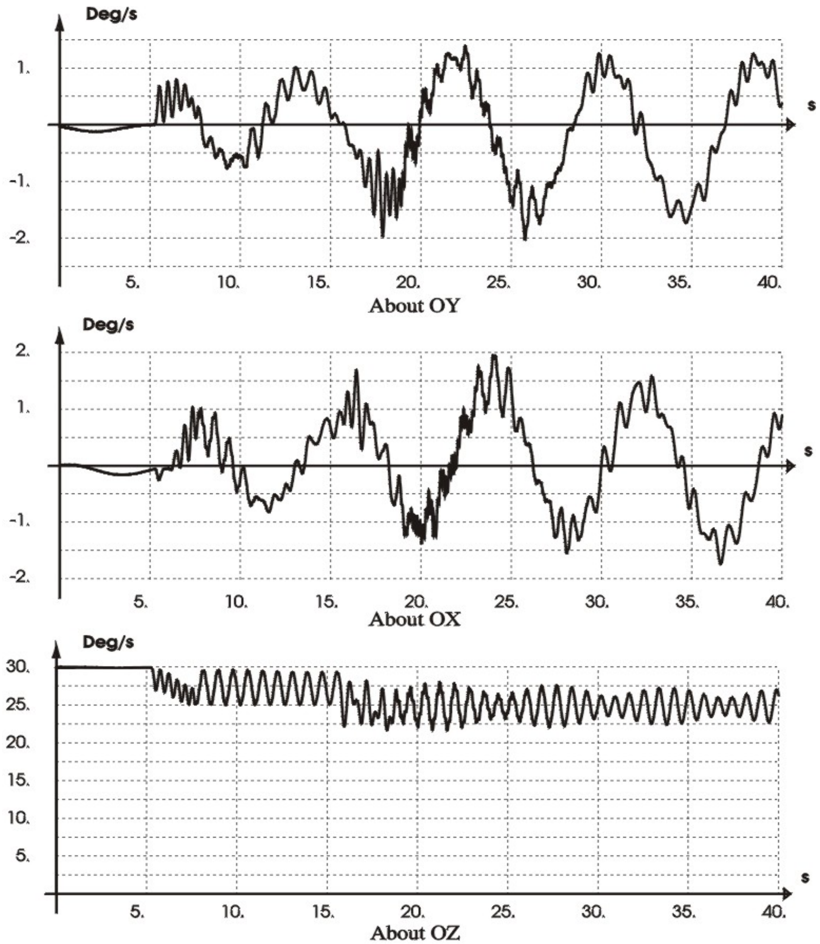
Boom -Y



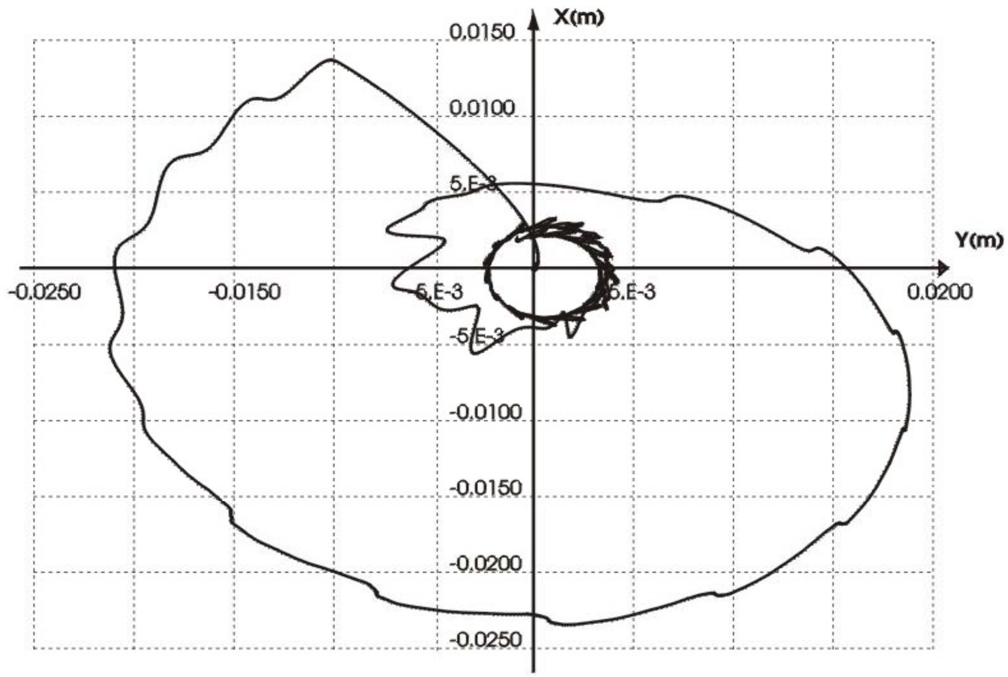
Boom +Y

Angular displacements

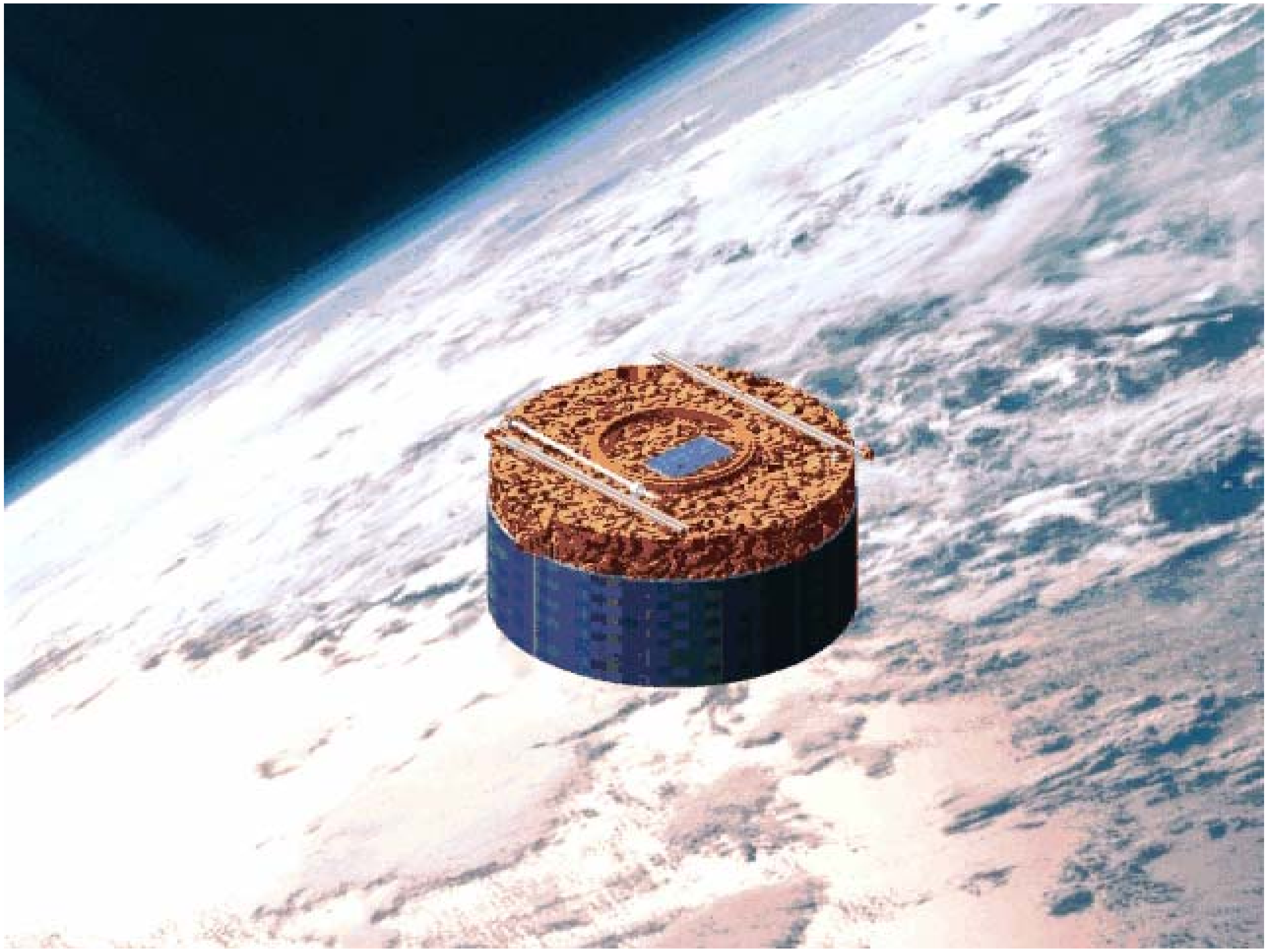
Deployment with flexible booms



Angular velocity of CM



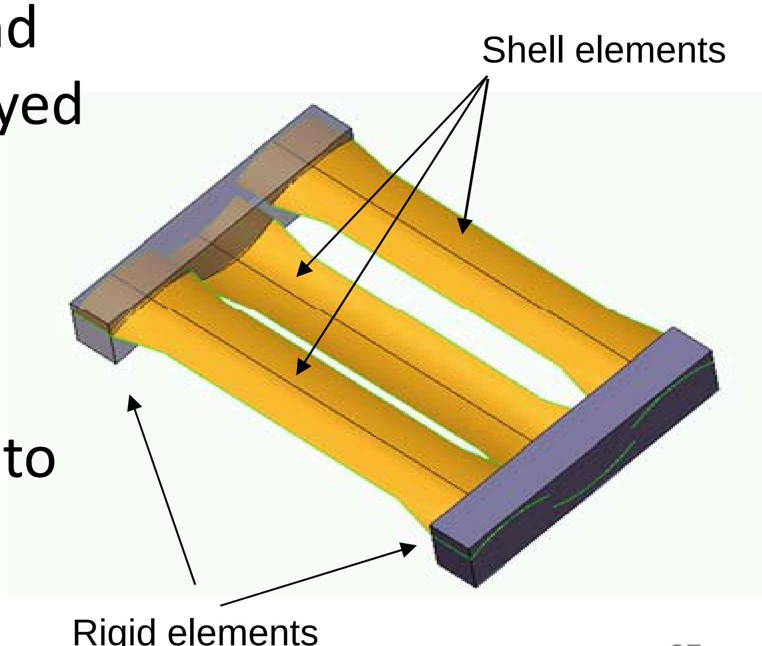
Trajectory of CM



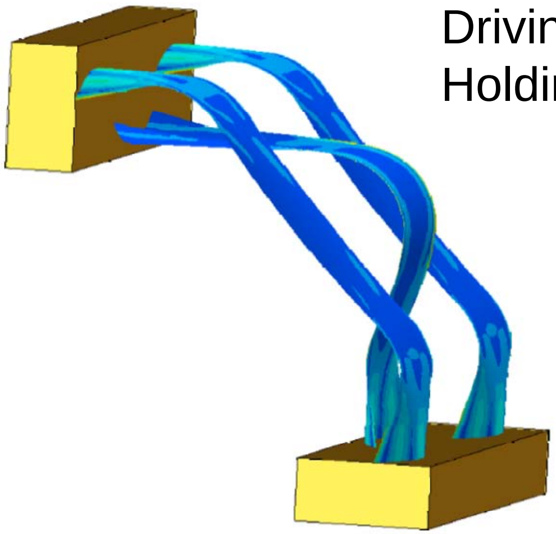
Mesocopic joint modeling: MAEVA hinge

- Improved version of Carpentier joint (patented design).
- Consists of three steel strips with curved cross section.
- Combines guidance (due to anistropic stiffness), actuation (spring effect) and locking (high holding torque in deployed configuration).
- Minimization of mechanical parts.
- No sliding / moving interfaces.
- But: detailed 3D modeling essential to represent non-linear behavior and buckling phenomena.

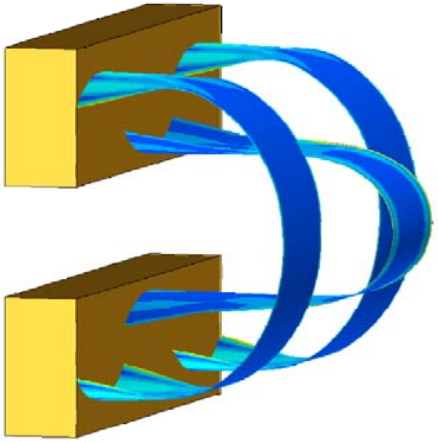
- Sicre, J.et al. Application of MAEVA hinge to Myriade microsatellite deployment needs. *ESMATS* 2005*.*
- Lathuiliere M. et al. Deployment of appendices using MAEVA hinges: progress in dynamic models. *ESMATS* 2009.



Static behavior



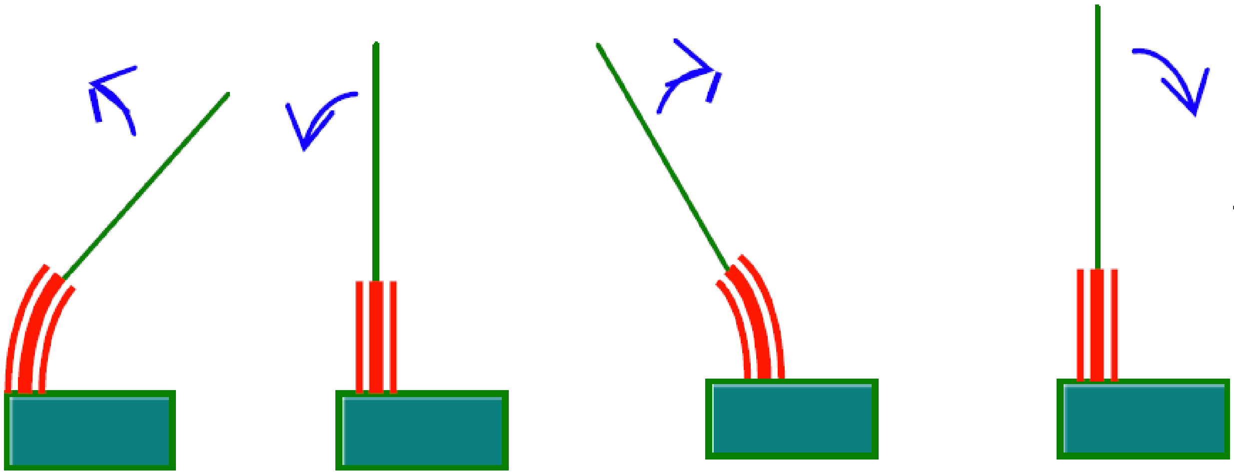
Driving torque : 0.152 Nm
Holding torque : 6.67 Nm



Driving torque : 0.194 Nm
Holding torque : 6.67 Nm

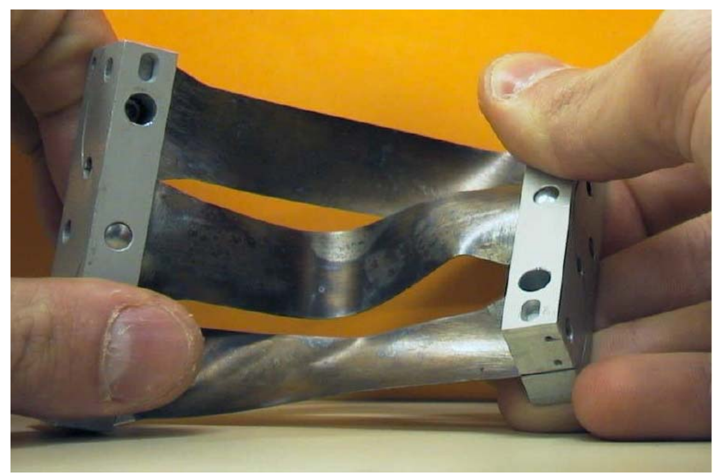
Deployed configuration Stowed configurations

Features of dynamic behavior

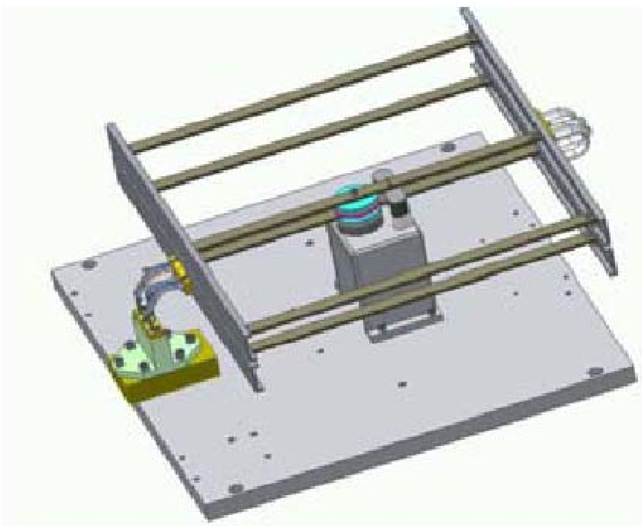


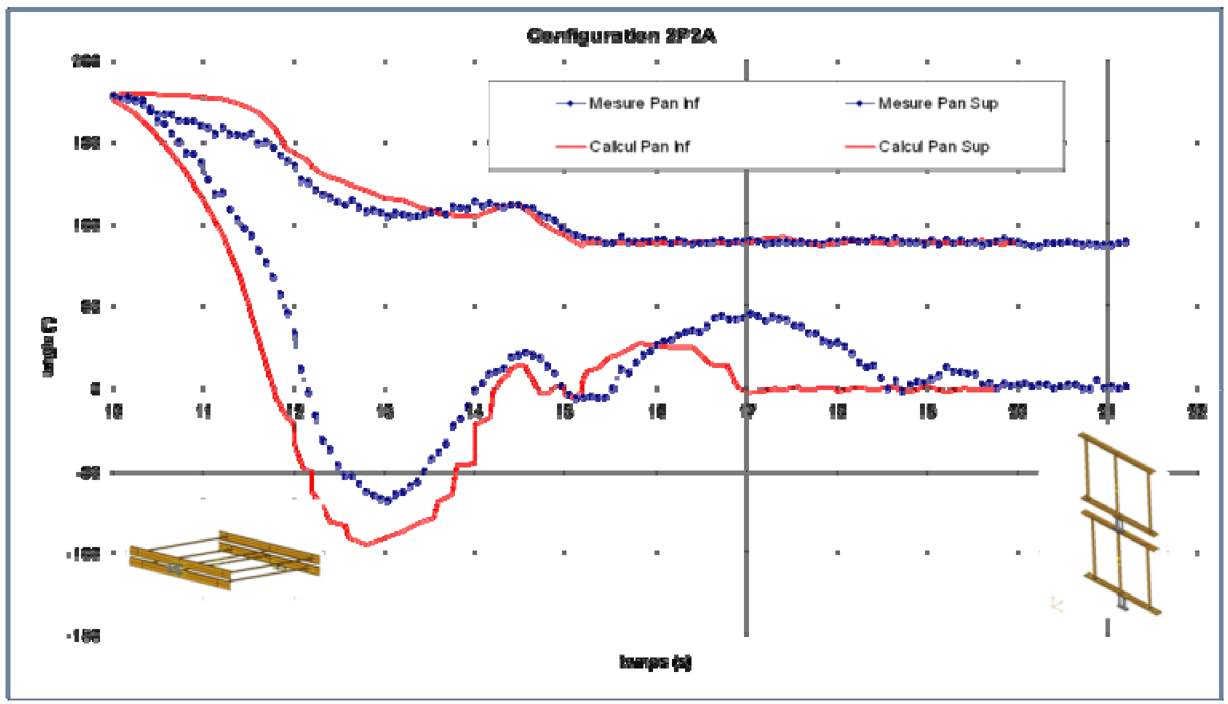
1. Oscillations around equilibrium position

2. Low torsional stiffness



2-panel deployment





Modal synthesis in multibody dynamics

Goal

• To represent complex structural members by standard, linear FE models

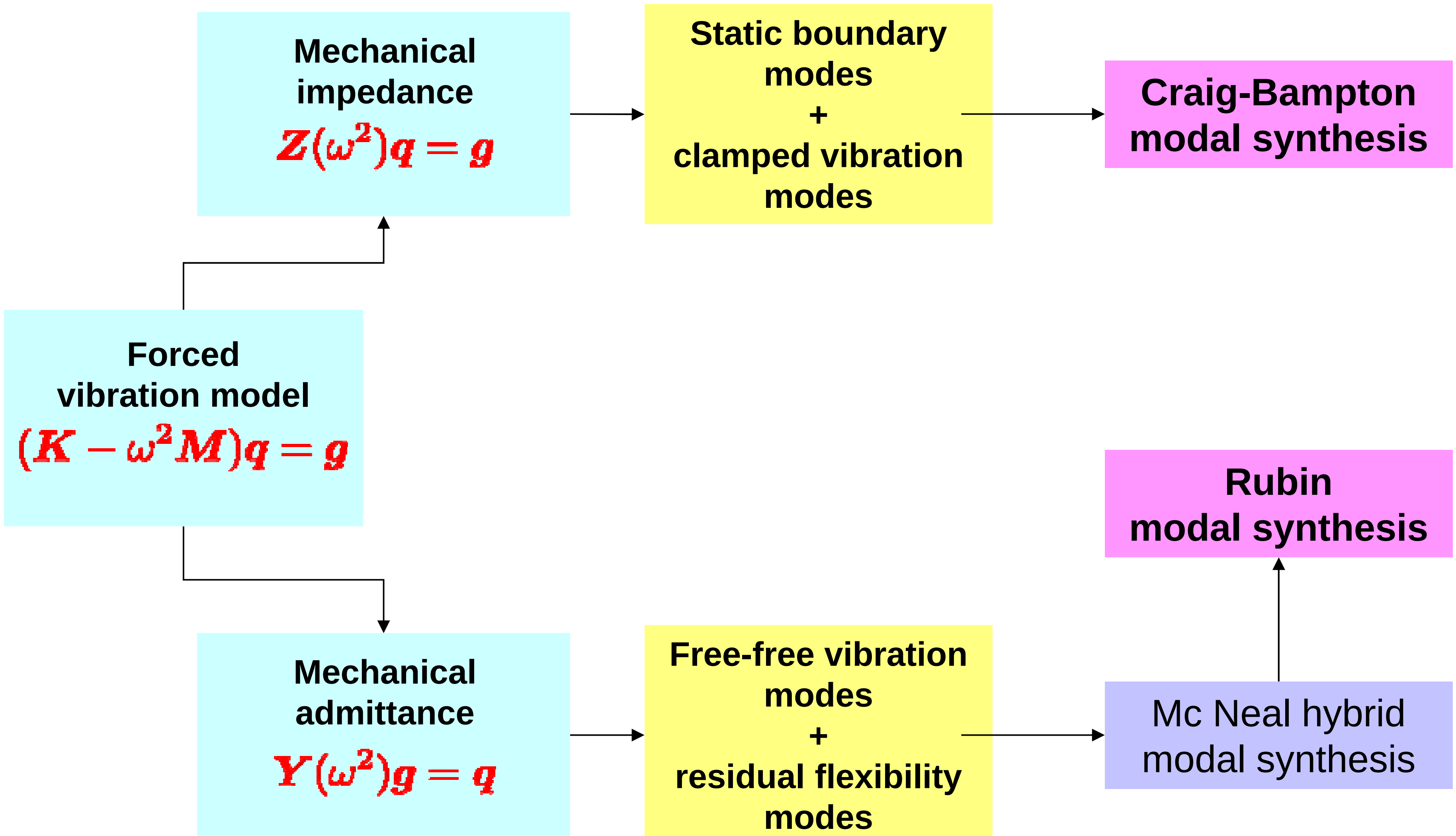
Method

• Assume linear behavior in local (co-rotational) frame
• Produce reduced model with
  - attachment modes to external system
  - reduced set of internal modes to represent local behavior

Limitations

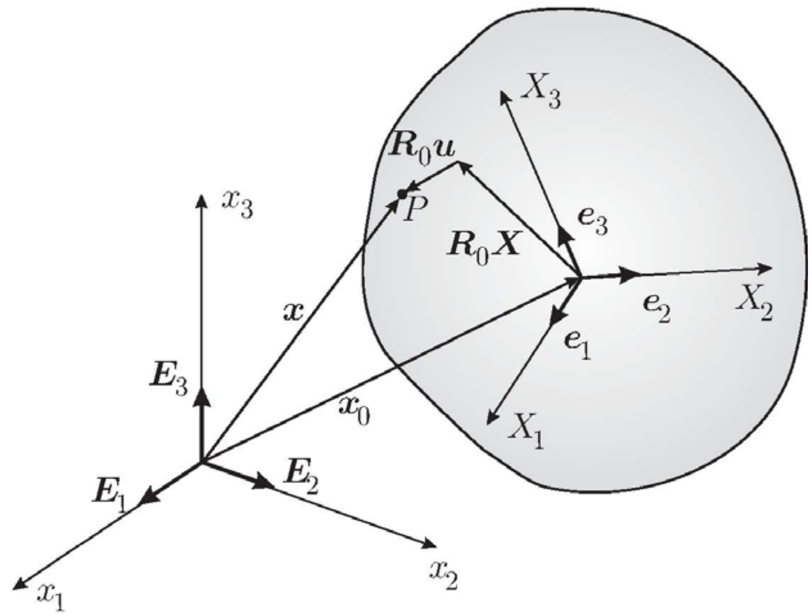
• Due to linearity assumption
• Recuperation of model components (superelement) from standard linear analysis $\rightarrow$ missing information on inertia.

Modal synthesis: dual approaches



Underlying kinematic assumption

$$\boldsymbol{x} = \boldsymbol{x}_0 + \boldsymbol{R}_0(\boldsymbol{X} + \boldsymbol{u})$$
$$\boldsymbol{\Psi} = \boldsymbol{\Psi}_0 \circ \boldsymbol{\psi}$$



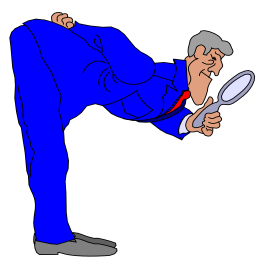
- Solve with respect to elastic displacements

$$\begin{bmatrix} \boldsymbol{u} \\ \boldsymbol{\psi} \end{bmatrix} = \begin{bmatrix} \boldsymbol{R}_0^T(\boldsymbol{x} - \boldsymbol{x}_0) - \boldsymbol{X} \\ (-\boldsymbol{\Psi}_0) \circ \boldsymbol{\Psi} \end{bmatrix}$$

- Modal representation of elastic displacements

$$\begin{bmatrix} \boldsymbol{u} \\ \boldsymbol{\psi} \end{bmatrix} = \overline{\boldsymbol{\Phi}} \boldsymbol{y}$$

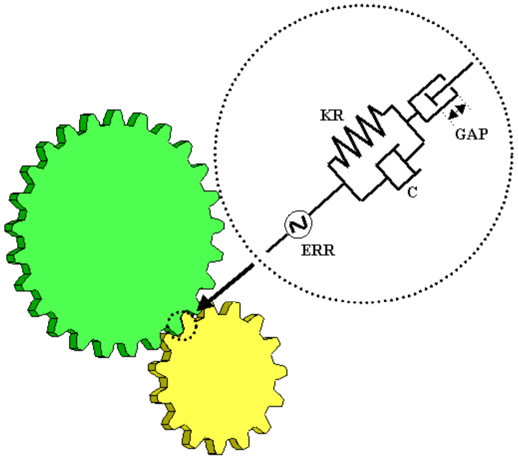
- Small displacement assumption in local frame



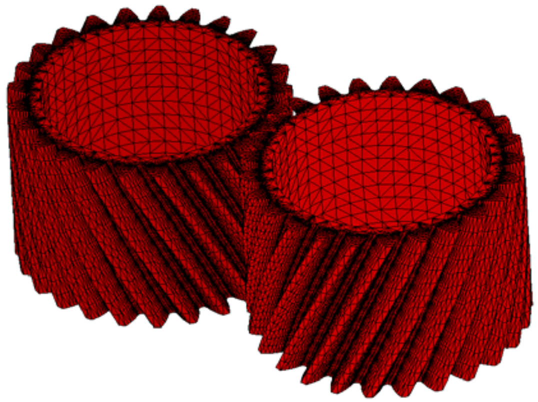
$$\frac{\|\boldsymbol{u}\|}{\|\boldsymbol{X}\|}, \ \|\boldsymbol{\psi}\| \ll 1$$

Application example: Gear modeling

- ❑ **Macroscopic gear models** [Cardona, 1995]
  - Kinematic constraints between wheel centers.
  - Gear wheels = rigid bodies.
  - Spring-damper along normal pressure line.
  - Crude /empirical modeling of meshing defaults: (e.g. backlash, friction, load transmission error).



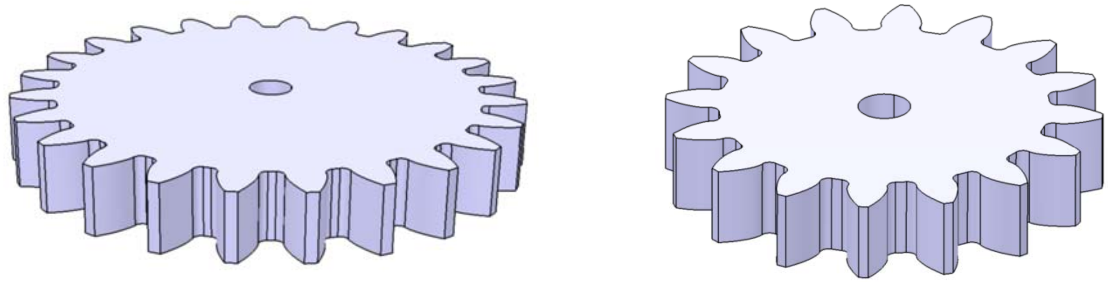
- ❑ **Contact condition between full FE models**
  - Gear teeth deformation and gear web accurately taken into account.
  - Meshing defaults naturally modeled.
  - Long simulation times $\rightarrow$ limited use.



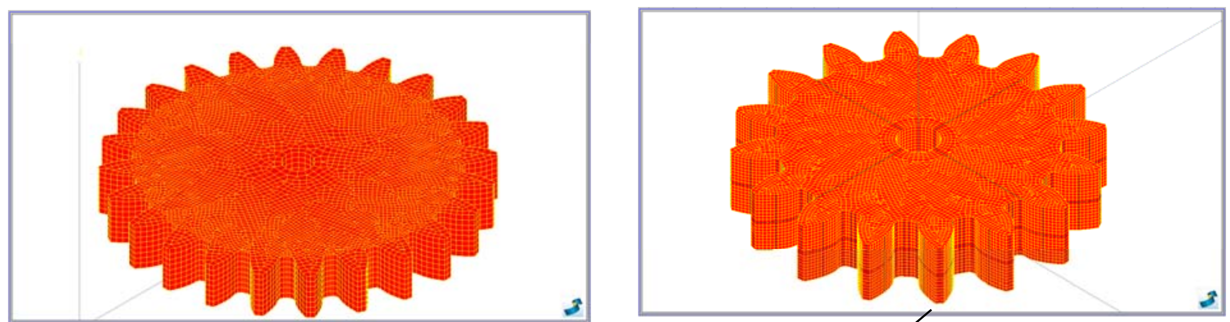
- ❑ **Contact condition between superelements**
  - Gear wheel flexible behavior globally accounted for.
  - Determination of actual contact points by means of 3D gear wheel geometry.
  - Study of misalignment, backlash, gear hammering,…

Simple gear system

1) CAD modeling

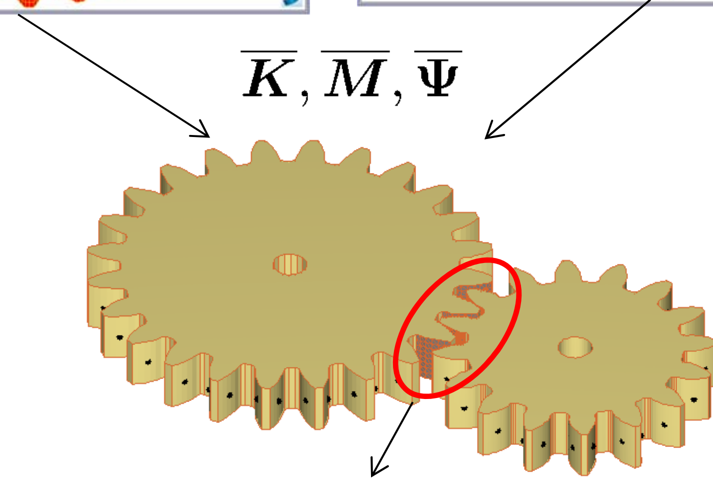


2) FE modeling and model reduction



$$\overline{\boldsymbol{K}},\ \overline{\boldsymbol{M}},\ \overline{\boldsymbol{\Psi}}$$

3) Simulation of unilateral contact between superelements (MATLAB)

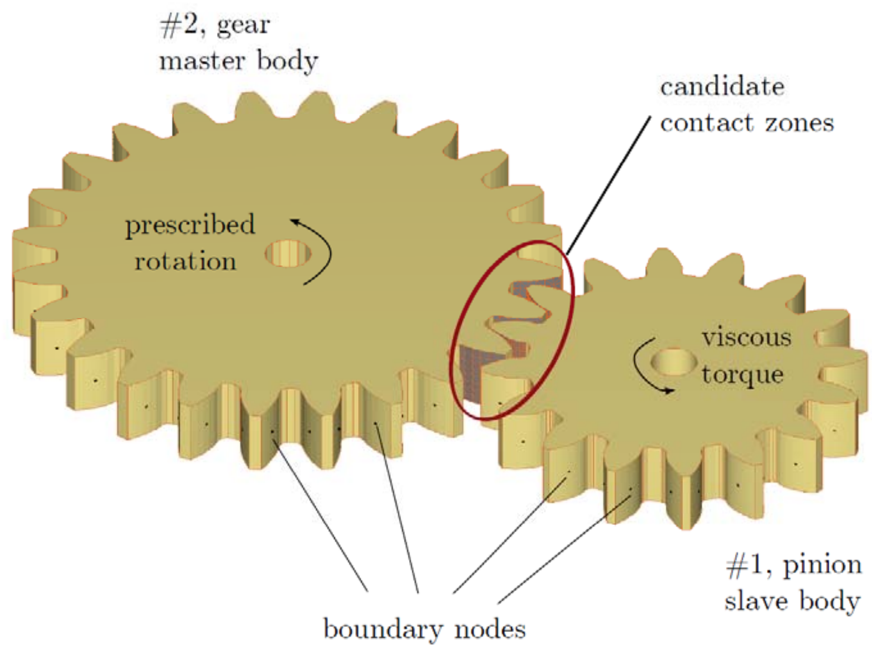


Selection strategy of slave-master flank pairs

Simple gear system simulation Craig & Bampton modal synthesis

	pinion	gear
Number of teeth [-]	16	24
Pitch diameter [mm]	73,2	109,8
Outside diameter [mm]	82,64	118,64
Root diameter [mm]	62,5	98,37
Addendum coef. [-]	0,196	0,125
Tooth width [mm]	15	
Pressure angle [deg]	20	
Module [mm]	4,5	

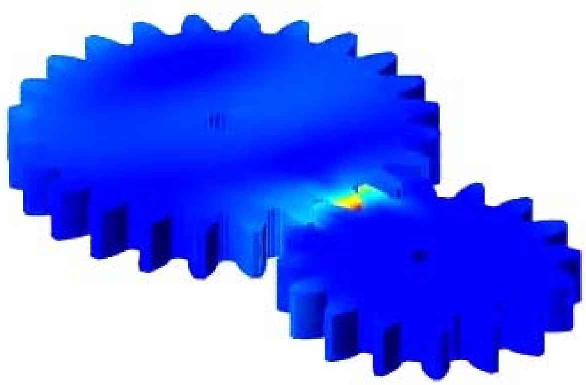
(Lundvall, Strömberg, Klarbring, 2004)

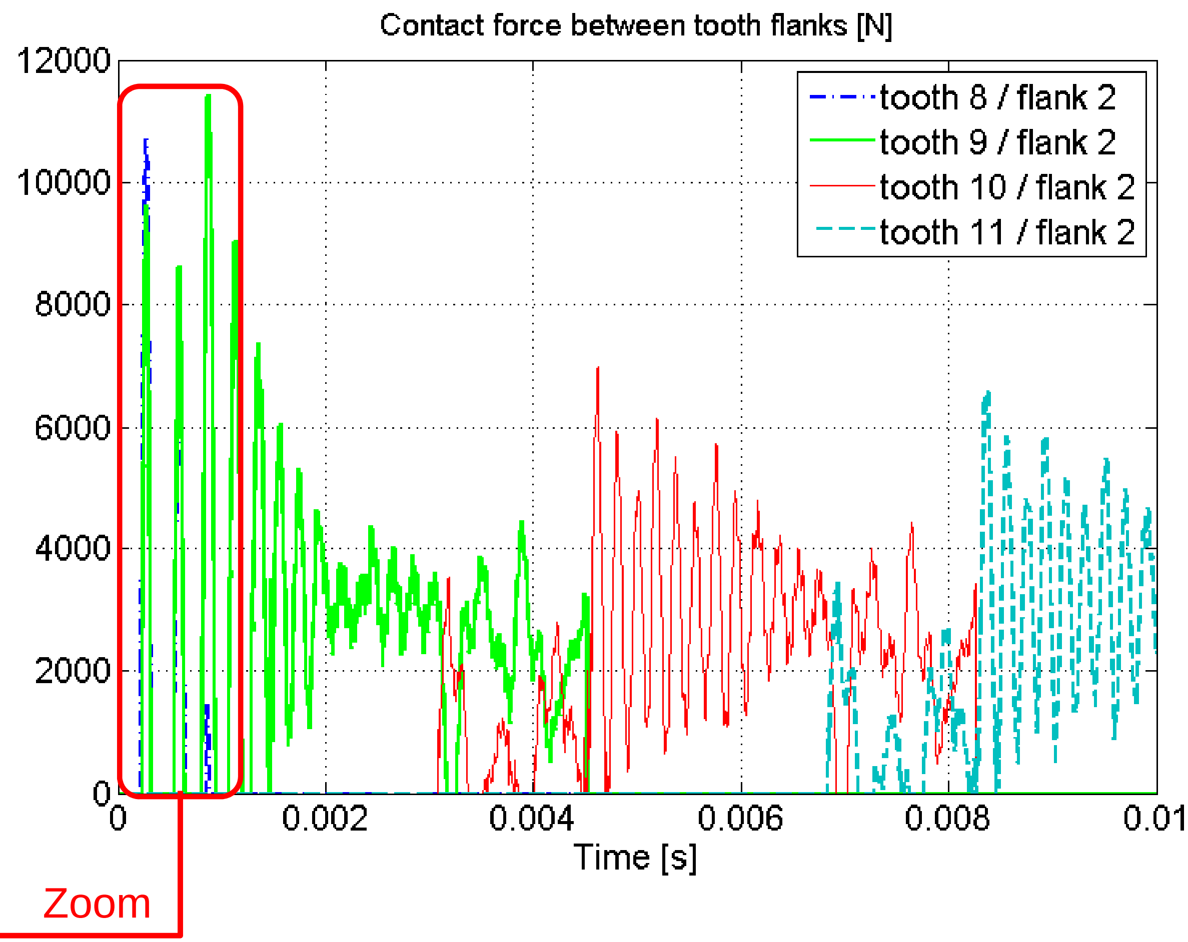


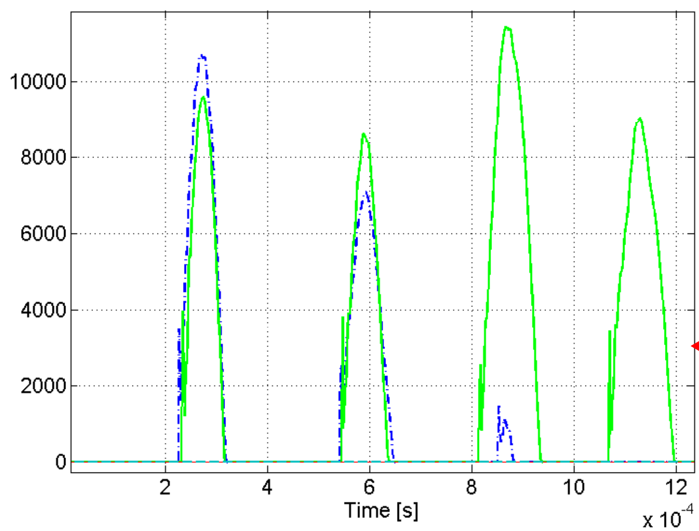
- 1 boundary node per tooth flank
  100 internal vibrations modes
  $\rightarrow$ 695 DOFS << 480171 for FEM
- Parallel rotation axis $\rightarrow$ no misalignment
- Large center distance $\rightarrow$ significant backlash
- At t=0s, ω_1 = -1000 rpm , ω_2= 667 rpm
- For t > 0s : Viscous torque: $T_1 = -1\ \omega_1$; ω_2= 667 rpm
- Time step: h=1E.-6s

Eigenfrequencies of internal vibration modes (Hz)		
	pinion	gear
f_1	19520	10402
f_{100}	146068	115469

Gear system simulation







Co-simulation

Objectives

- Coupling between standard MBD and FEA softwares.
- Co-simulation of mechatronic systems.
- « Domain decomposition » of very large problems for parallel processing → multi-model solution.

Methodology

- Generally based on the master / slave concept
- Different coupling levels can be envisaged, depending upon the nature of coupled systems and degree of accuracy of final solution.

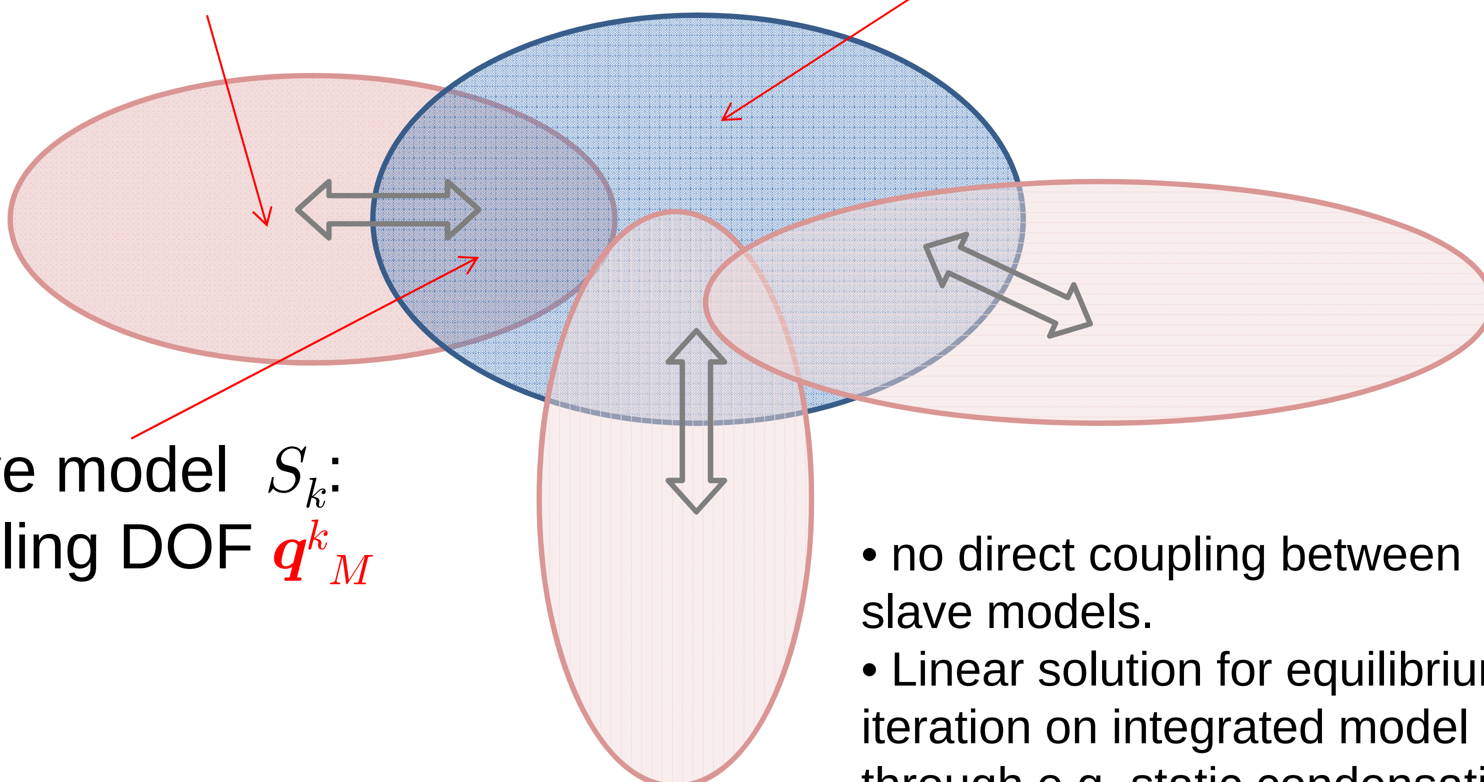
Example of multi-model solution

- vehicle dynamics simulation in higher frequency range → requires detailed modeling of critical components (e.g. coil springs, torsion bar, tires).
- Necessarily implies fully coupled solution.

Principle of multi-model solution

slave model S_k:
Internal DOF $\boldsymbol{q}^k{}_i$

Master model M.
DOF $\boldsymbol{q}^M$

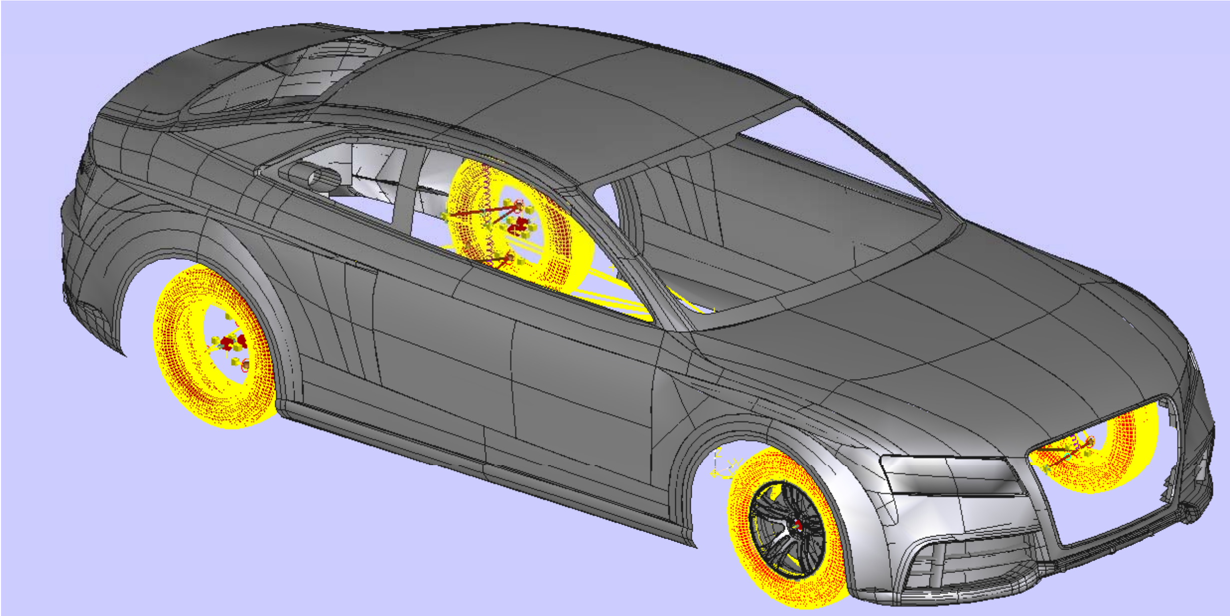


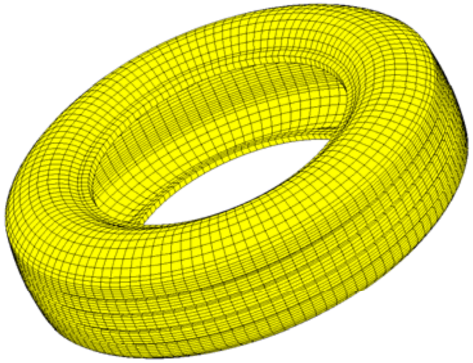
slave model S_k:
coupling DOF $\boldsymbol{q}^k{}_M$

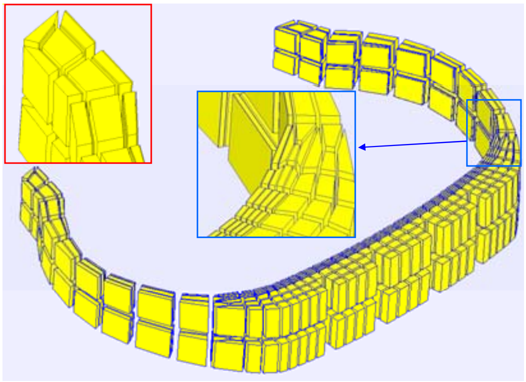
• no direct coupling between slave models.
• Linear solution for equilibrium iteration on integrated model through e.g. static condensation

Example: MBD car model with FE tire models

- Car body: rigid.
- Tires and twist beam: FE models.
- ~ 2.10^6 DOF.



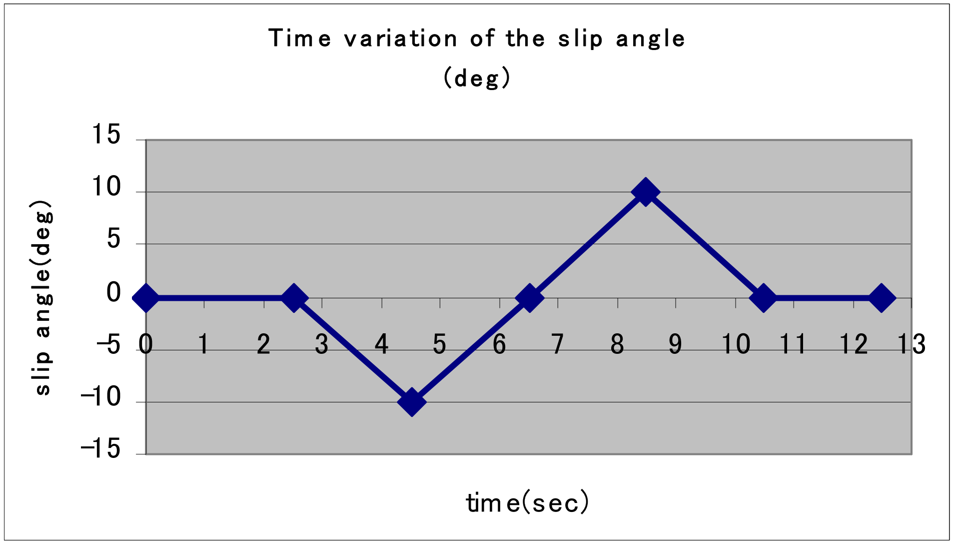


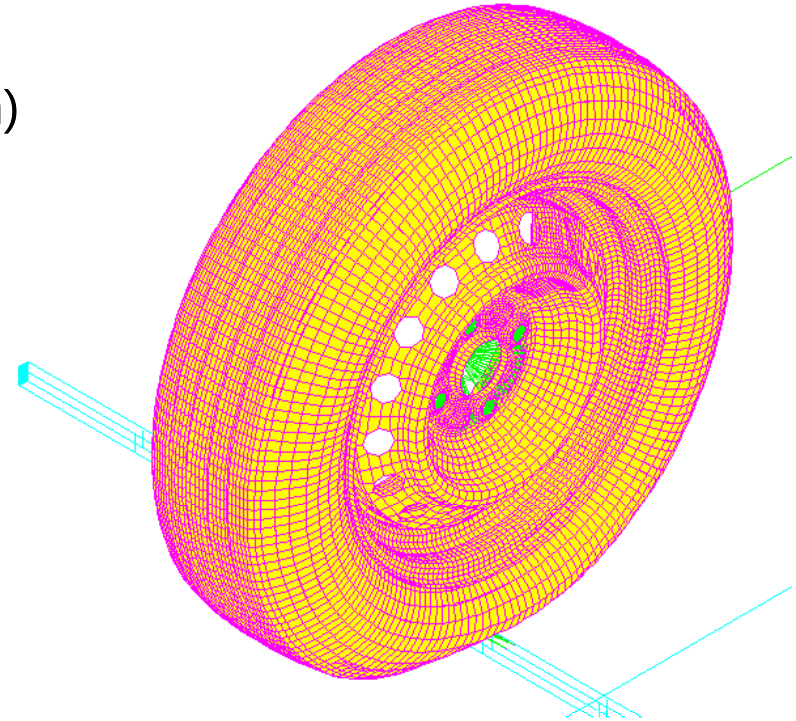


Objective: to capture high frequency response to road roughness.

FEA tire equipped car model

- ❑ 2 validation rides at 60 km/h
  - Straight ride with imposed variations of the slip angle
  - Straight ride over a comfort obstacle
- ❑ Simulation done in 3 phases
  - Static equilibrium (gravity + tire inflation)
  - Kinematics (initial velocity field)
  - Dynamic ride





Straight ride with slip angle variation

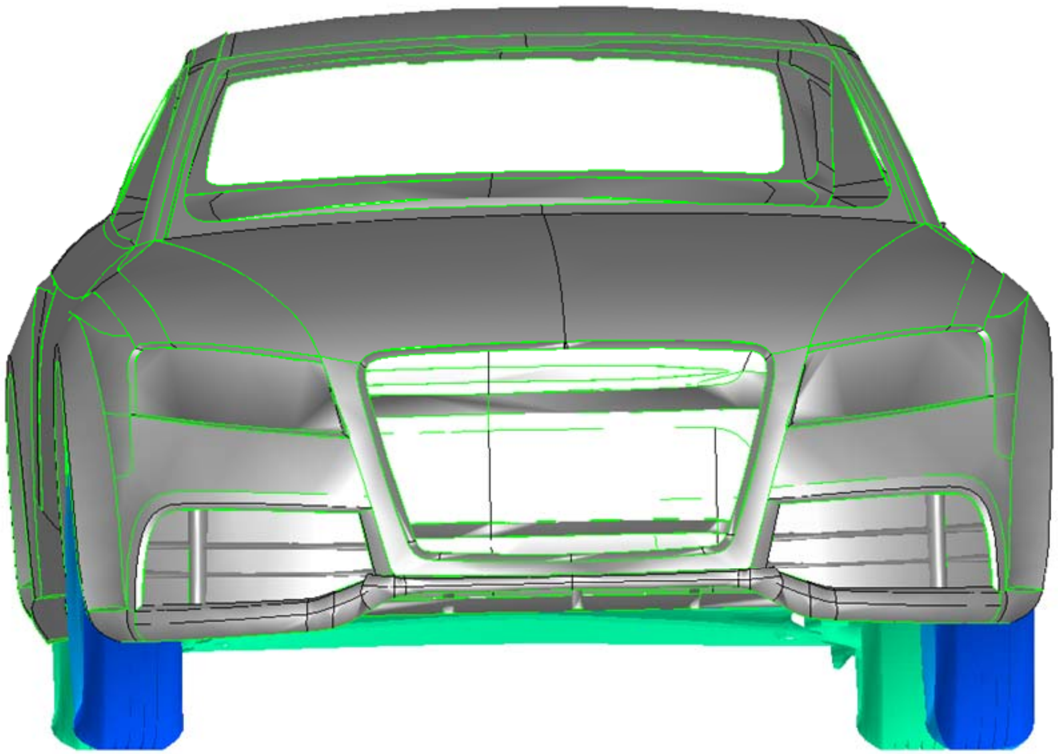


Figure 6. lateral displacement during slalom ride (front view)

Solution / multi-level solution accuracy

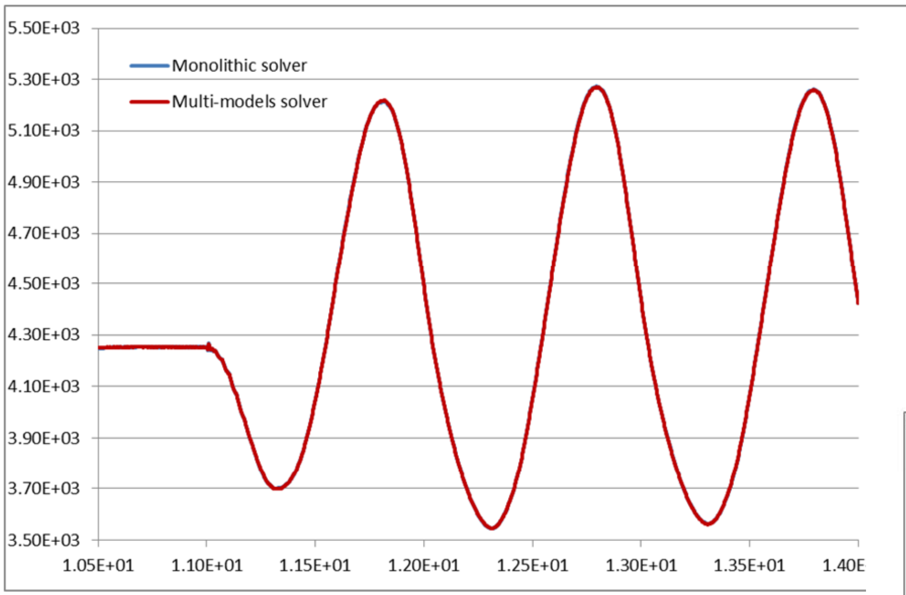


Figure 7. Magnitude of the force transmitted by the front left wheel

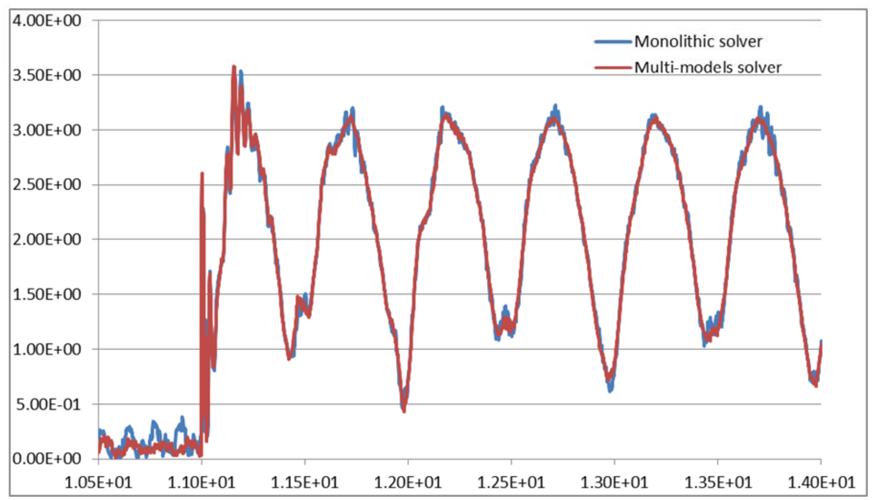


Figure 8. Global acceleration at front left wheel center

Computation performance

models	*Nr of iterations (linear system solutions)*	*Iteration cost ratio*	*Total Cost ratio*
1 model (6 Proc)	9087	1	1
1 model (12 Proc)	9087	0.62	0.66
1 model (24 proc)	9087	0.69	0.79
Multi-model (4 slave models, 25 proc)	7787	0.29	0.25
Multi-model (5 slave models, 26 proc)	7800	0.23	0.22

Conclusions

- FEA-MBD integration: not a dream, but a reality.
- Complexity of problems being addressed continuously increasing due to progress both in physical model quality, computer technology and numerical methods.
- Numerous topics still open for possible progress (e.g. contact/impact & friction, post-processing, computing performance).

Thank you !