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Planetary gear modelling/optimization; Experiments on coated gears

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Planetary gear modelling/ optimization

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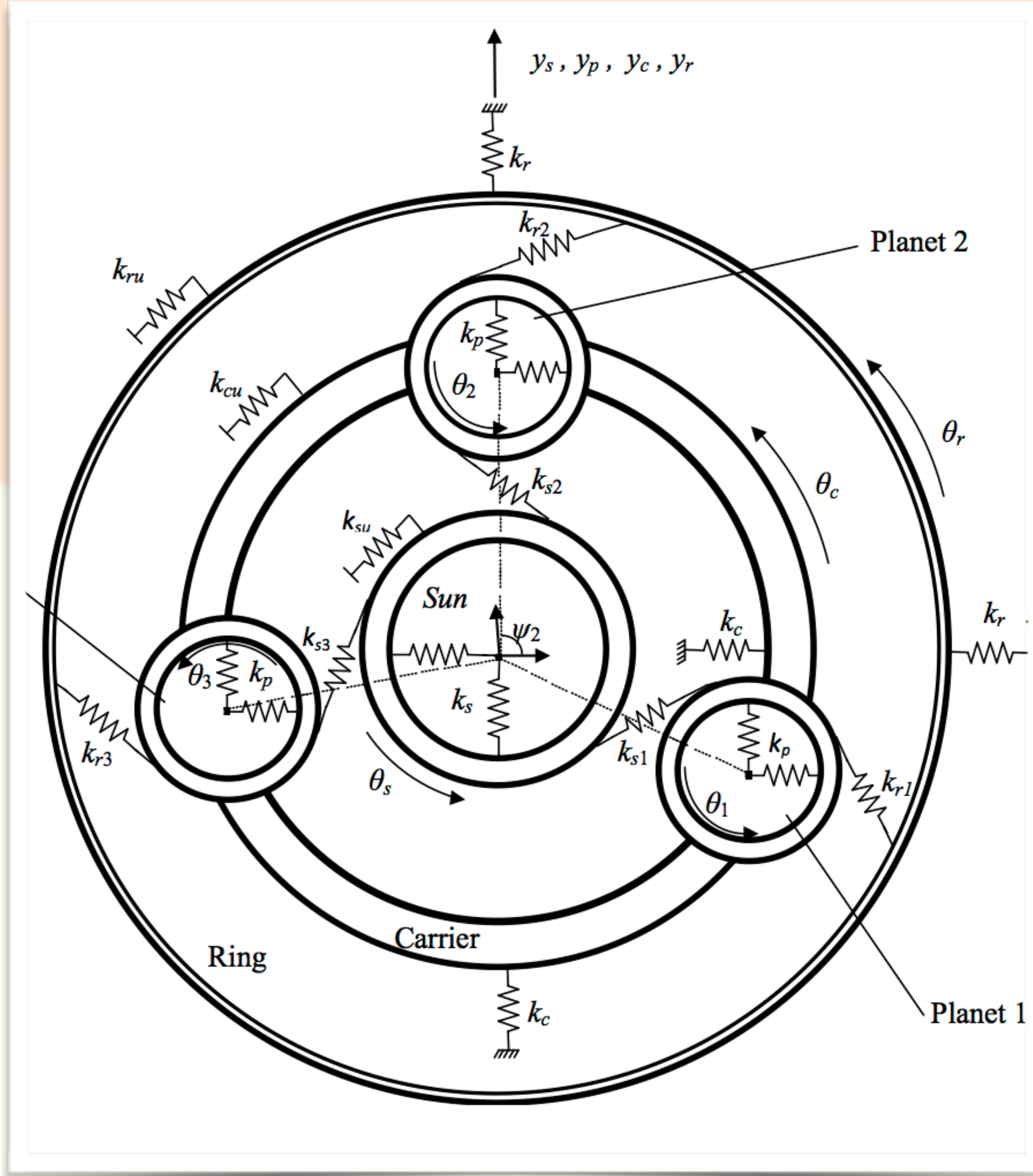
Forever Forward

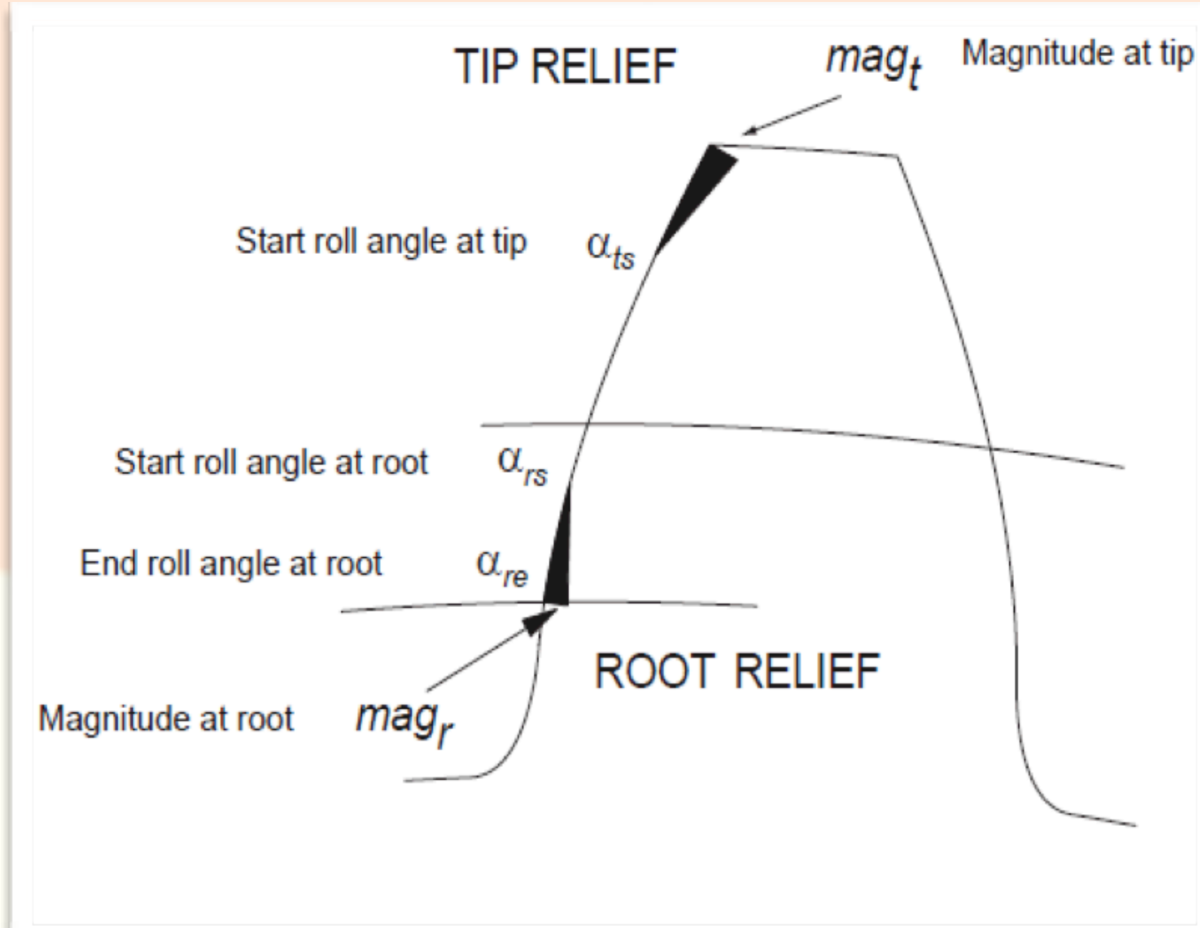
Planetary gear modelling

- The goal was to find optimal profile modifications in order to reduce the overall vibration (Gear day 2017)
- Dynamic behavior of planetary gear can be modeled using lumped parameter model
- Tip and root relief on sun and planets are considered (no modifications on the ring)
- The model includes:
  - FE evaluated stiffnesses of gear pairs
  - Backlash in the contacts
  - Geometric errors due top profile modifications
  - Bearing stiffness



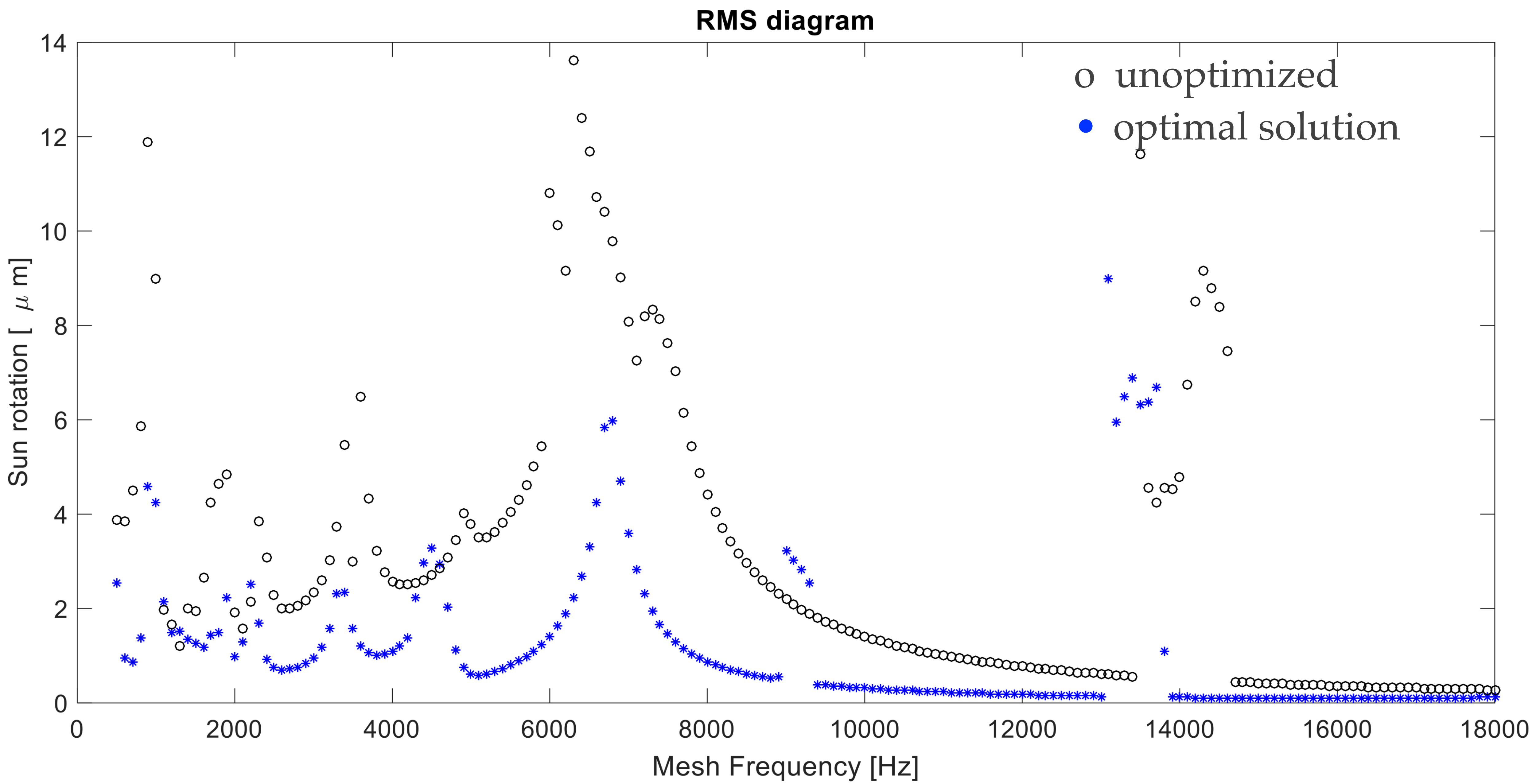
Optimization of profile reliefs

- Optimization parameters:
  - Sun profile reliefs (tip and root)
  - Planet profile reliefs (tip and root)
- Objective function:
  - Peak to peak of the STE of the planetary gear
- No modifications on ring



Pinion				Gear			
α_{ts}	mag_t	α_{rs}	mag_r	α_{ts}	mag_t	α_{rs}	mag_r
11 bits	6 bits	11 bits	6 bits	11 bits	6 bits	11 bits	6 bits
0110...	01..						...01

Planetary gear optimisation



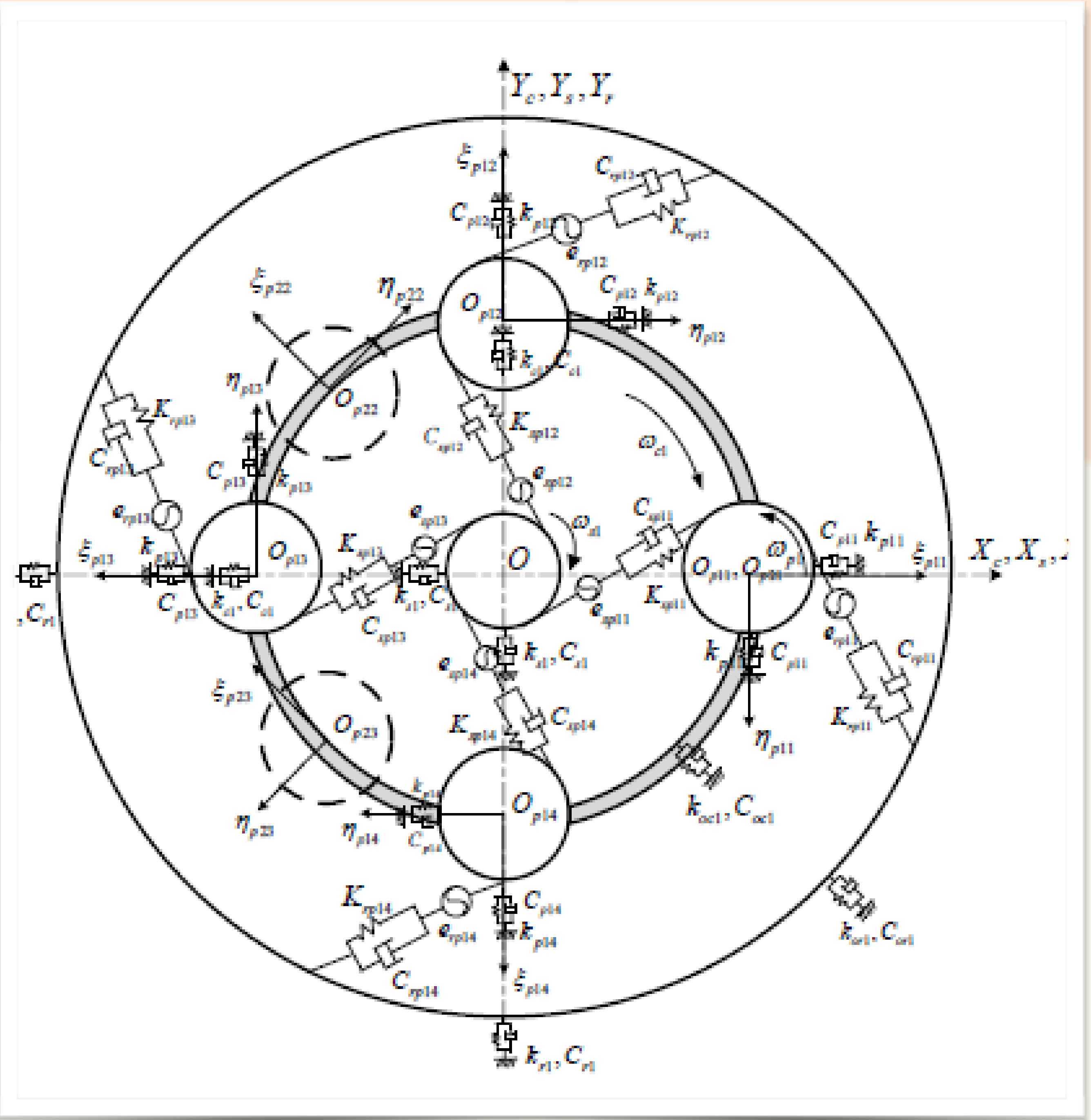
- Optimization (Genetic Algorithms) of the overall Static Transmission Error STE allows for reducing the Dynamic Transmission Error DTE at each mesh frequency

Evaluating Load Sharing Coefficient

- In a system having four or more planets, static load imbalance occurs
- Load sharing coefficient is a measure of the average force on the i-th planet

$$L_i = \frac{\frac{1}{T_m}\int_0^{T_m} F_i(t)\,dt}{M_1 / R_1}$$

- The dynamic model can be adapted in order to evaluate the load sharing coefficient



Evaluating Load Sharing Coefficient

Numerical analysis

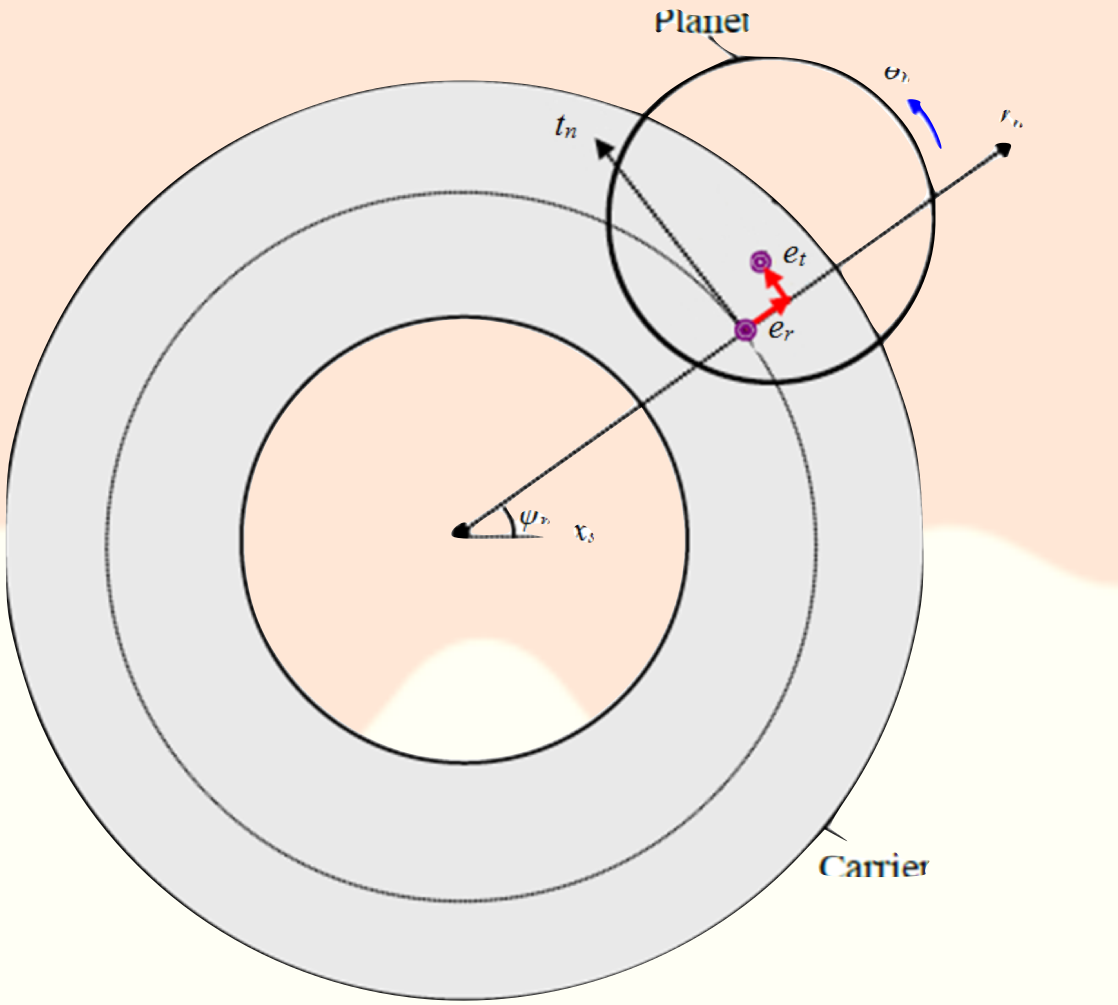
- Dynamic load-sharing sensitivity to errors which could provide references for designing planetary gears and load balancing.

The main sources of imbalance are:

- Carrier pinhole position error (the most significant)
- Planet size variation
- Run-out errors of the gears

Case study:

- 2D spur planetary gearbox with N equally spaced planets
- Nonlinear model with time varying mesh stiffnesses and backlash
- 200 micron positioning error of 2nd planet in tangential direction



Evaluating Load Sharing Coefficient

Angular position of the n-th planet

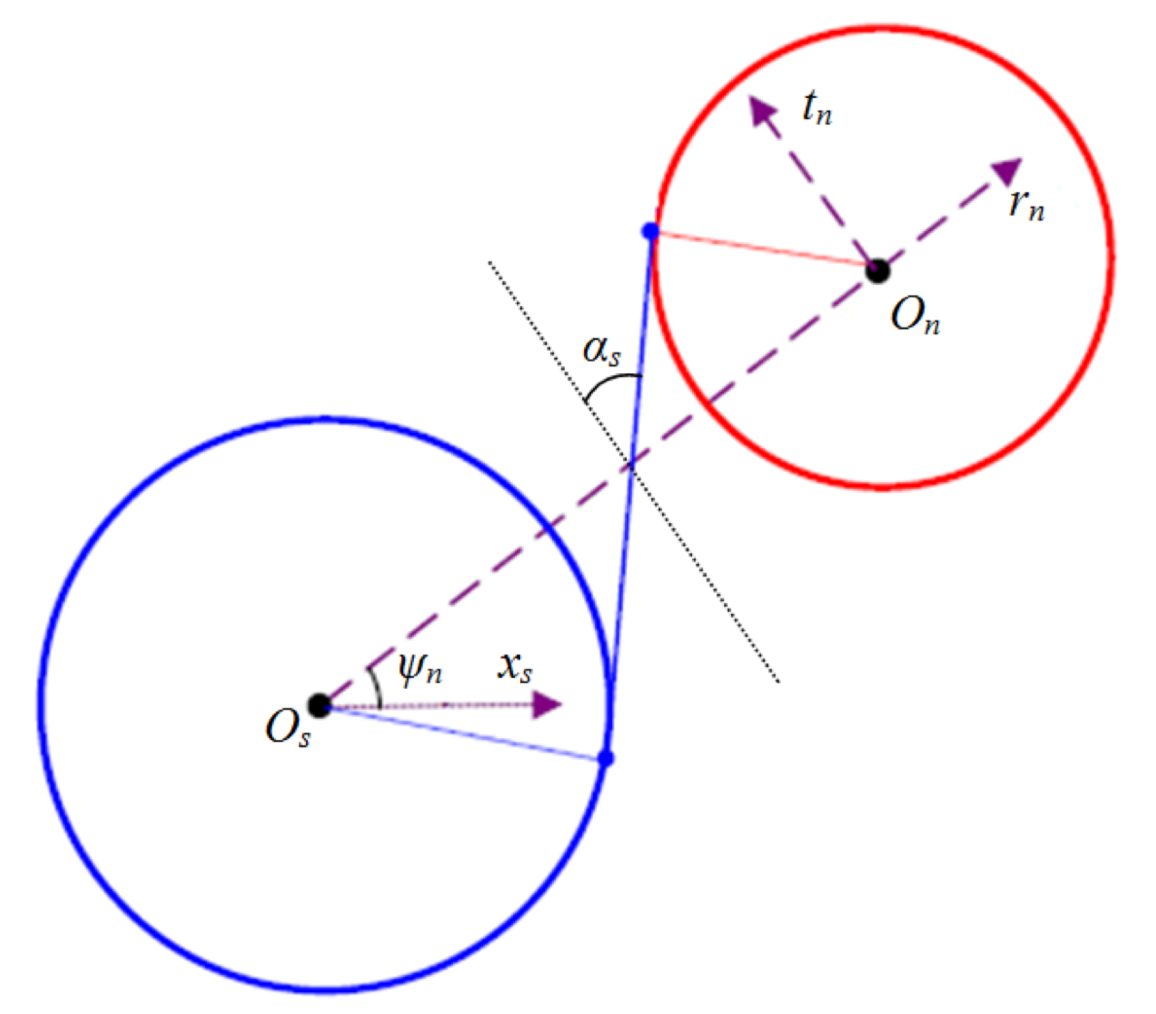
$$\hat{\psi}_n = \psi_n + a\tan\left(\frac{e_t}{e_r + cd}\right) \quad \psi_n + \frac{e_t}{e_r + cd}$$

Center distance

$$\hat{cd} = \sqrt{(cd + e_r)^2 + e_t^2}$$

Pressure angle

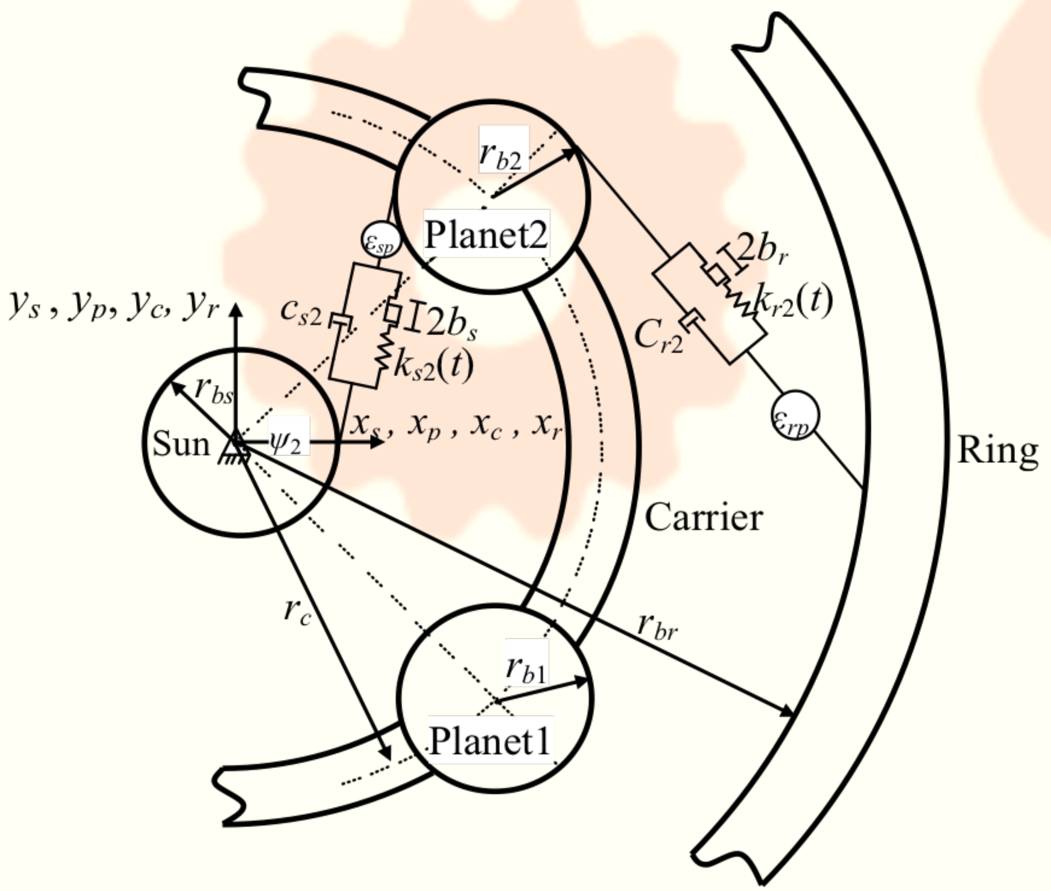
$$\hat{\alpha}_{s/r} = a\cos\left(\frac{r_{bs} \pm r_{bn}}{\hat{cd}}\right)$$

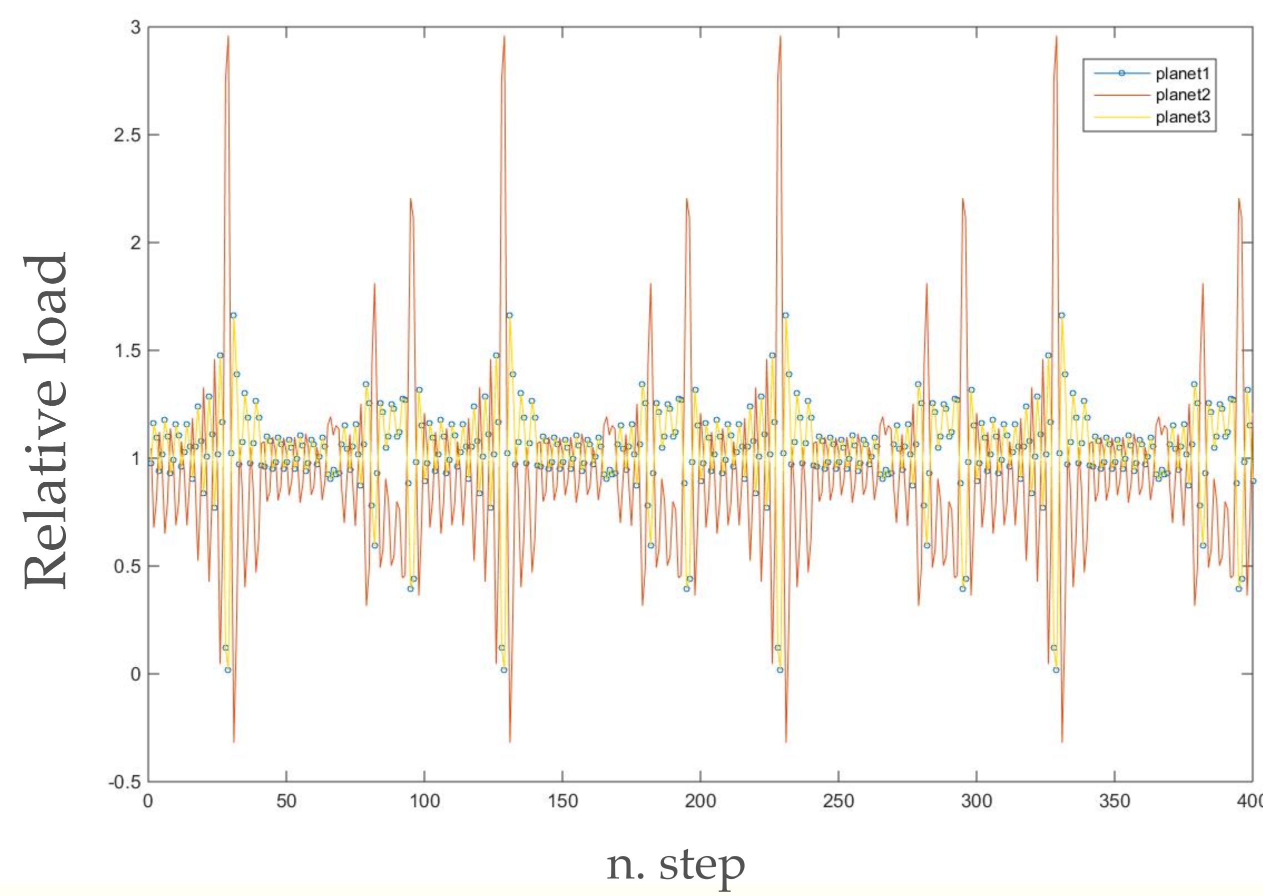


$$\begin{pmatrix} r_n \\ t_n \end{pmatrix} = \begin{pmatrix} \cos\psi_n & \sin\psi_n \\ -\sin\psi_n & \cos\psi_n \end{pmatrix} \begin{pmatrix} x_n \\ y_n \end{pmatrix}$$

- $\hat{\psi}_n, \hat{cd}, \hat{\alpha}_{s/r}$ are to be modified accordingly to the presence of errors
- Planet pin positioning errors are the more relevant

Evaluating Load Sharing Coefficient





- 200 µm error planet 2
- LS evaluation is performed for system with more planets

3 equally spaced planets

Load sharing factor	Planet1	Planet2	Planet3	
Load-sharing (sun-planet)	0.3436	0.3128	0.3436	

4 equally spaced planets

Load sharing factor	Planet1	Planet2	Planet3	Planet4
Load-sharing (sun-planet)	0.2634	0.1476	0.2682	0.3208

Load Sharing: next steps

- Further developments include:
  - An experimental validation of the proposed method, which can be performed by means of strain-gauges measurements
  - Optimization of planet phasing in order to reduce the AGMA mesh load factor $K\gamma$

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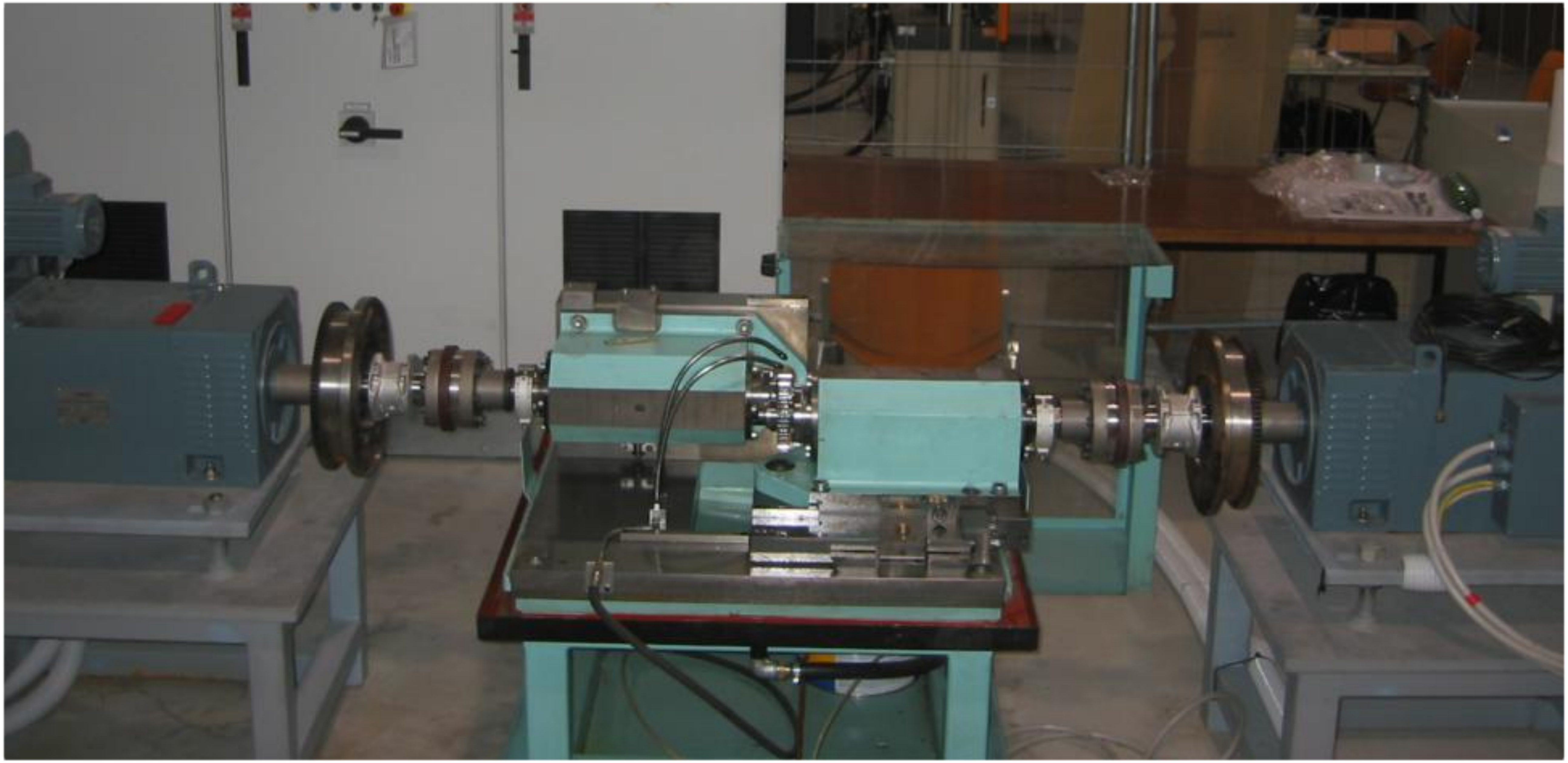
Experiments on coated gears

Test rig for coatings/treatments on gears



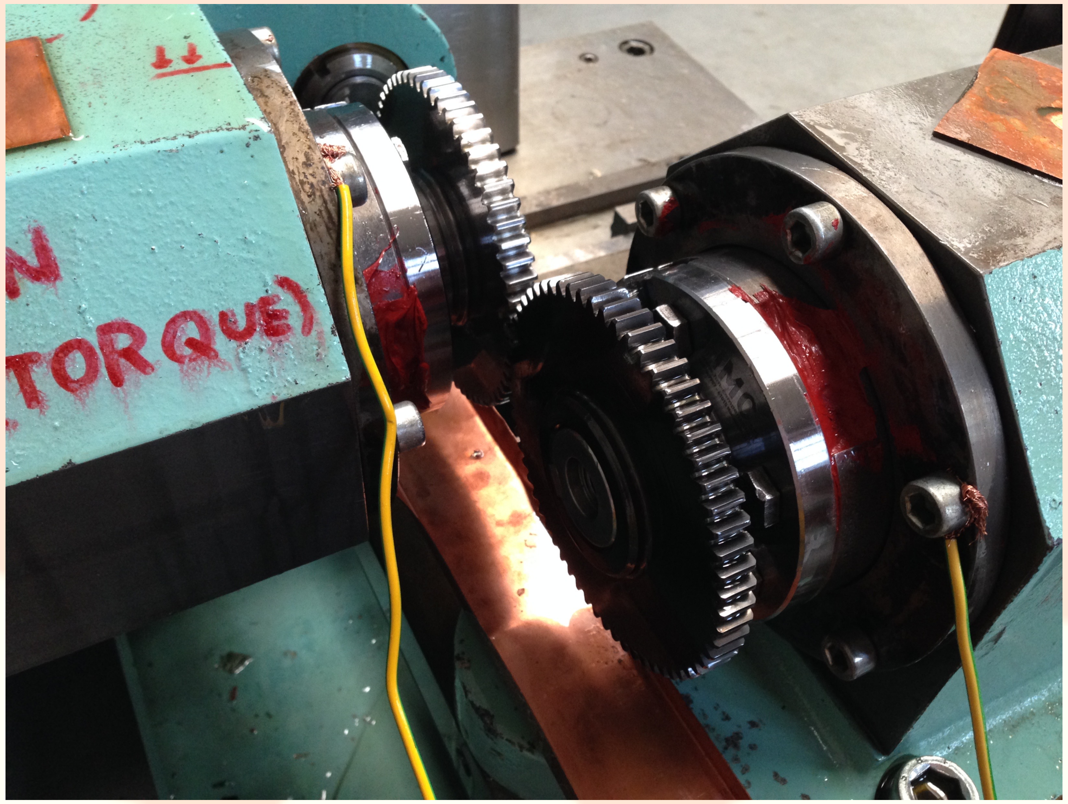
- A test rig for gear pair dynamic measurement (courtesy of CNH) has been used for an experimental campaign

Test planning

1-3	tempered	/	/
4-6	tempered	coating 1	/
7-9	tempered	coating 2	/
10-11	tempered	/	/
12-14	carburized	/	/
15-17	carburized	coating 1	/
18-20	carburized	coating 2	/
21-23	carburized	/	texturing 1
24-26	carburized	/	texturing 2
27-29	nitrided	/	/
30-32	nitrided	coating 1	/
33-35	nitrided	coating 2	/

- Gears with three kinds of thermal treatment are considered
- For each thermal treatment, two different kinds of Tungsten Carbide/Carbon (WC/C) coatings and the uncoated case are investigated
- Some gears have been left for surface texturing (to be tested)
- An equivalent set of tribometer disks have been produced (using the same steel)

Test planning

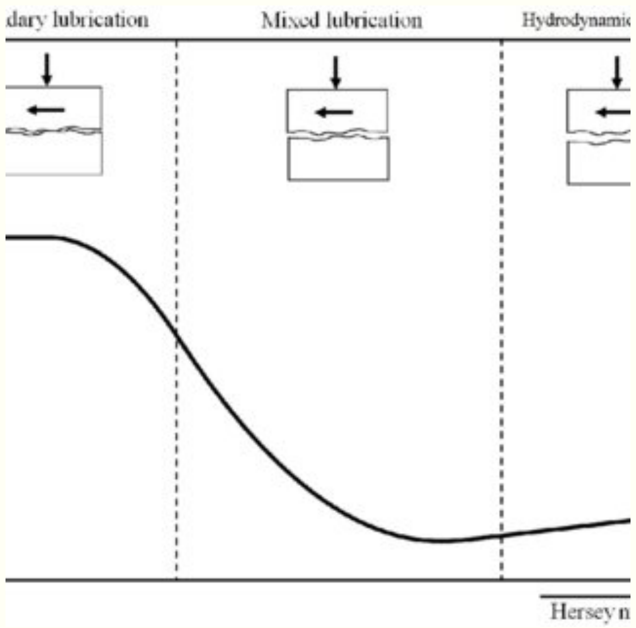


<u>Gear test</u>

- Rolling/sliding contact
- Max pressure: 1.5 GPa (pure rolling)
- Max sliding speed: 2 m/s @ 0.8 Gpa
- Oil jet lubrication: Shell OMALA VG 68 (mineral)
- Efficiency curves will be measured

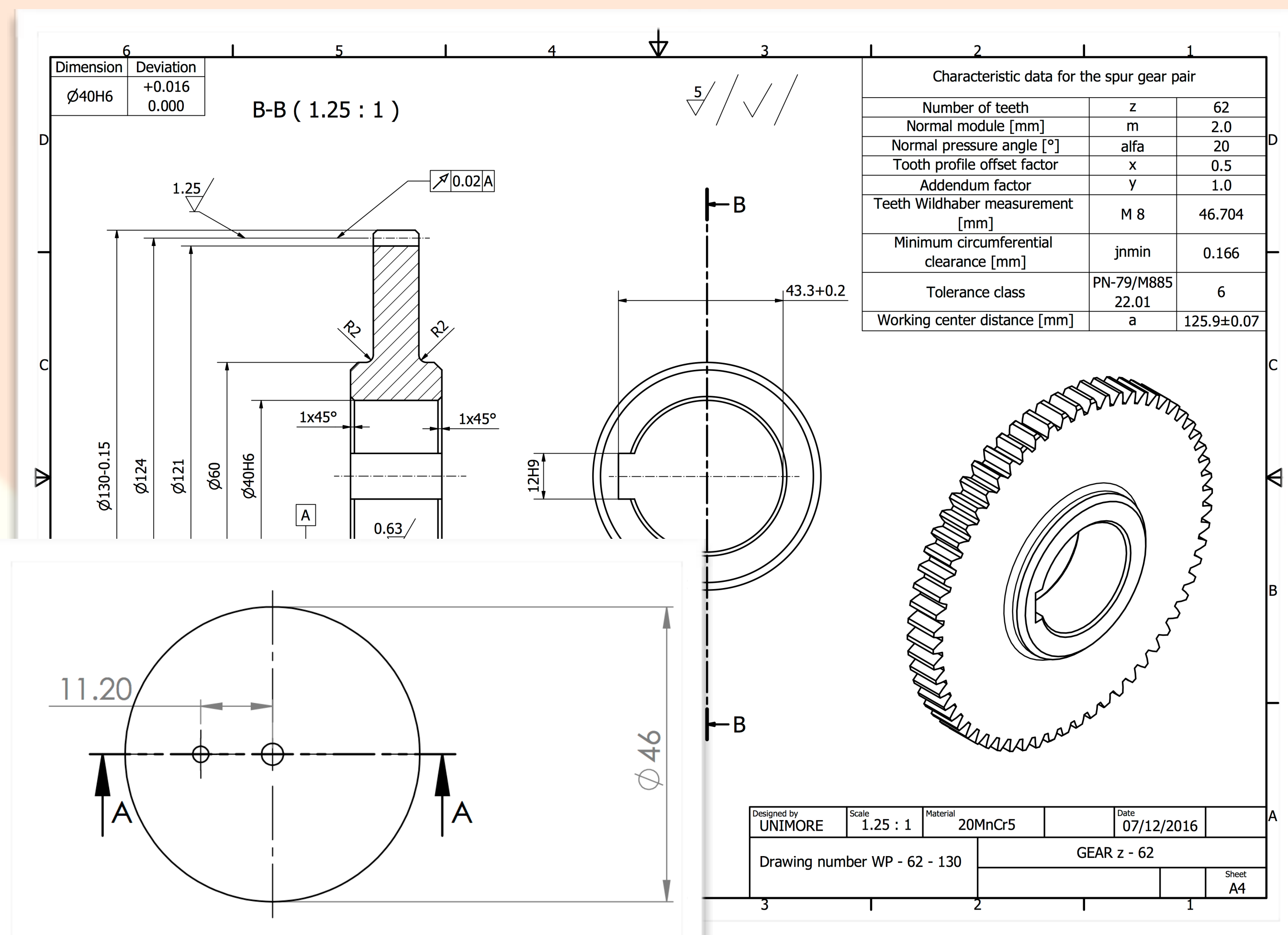
<u>Ball on disk tribometer</u>

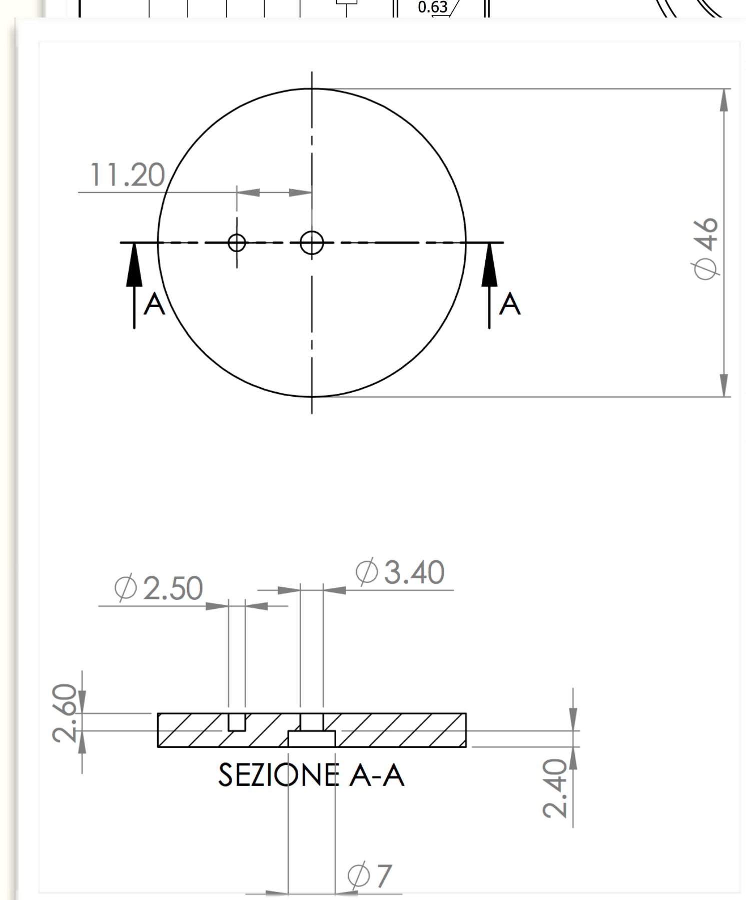
- Sliding contact
- Max pressure: 0.8 GPa
- Max sliding speed: 1.5 m/s
- Stribeck curves will be measured
- Dip lubrication: VG 68 LT or VG 150 HT
- Pure sliding will result in less hydrodynamic lift



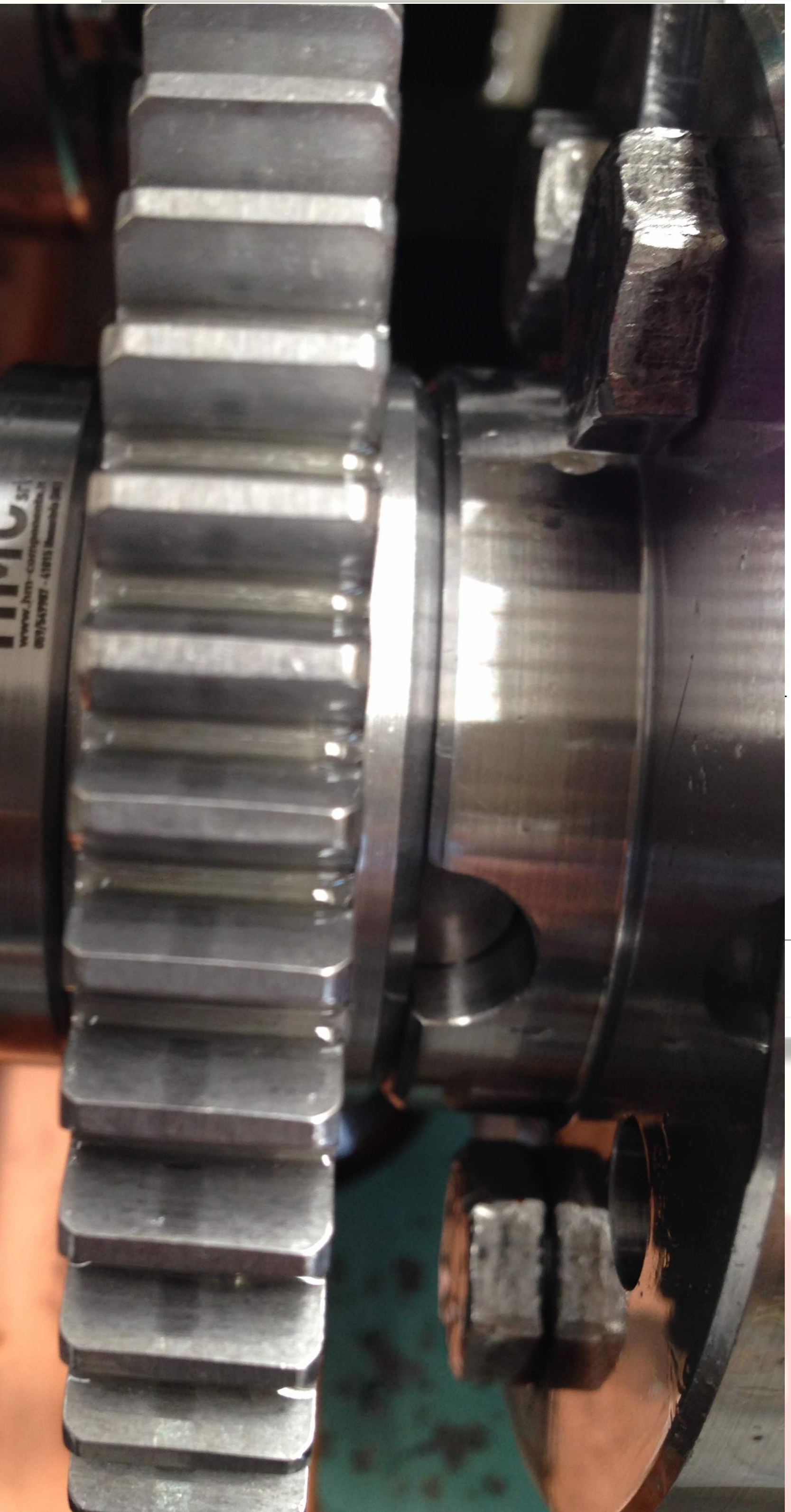
Gear sample design

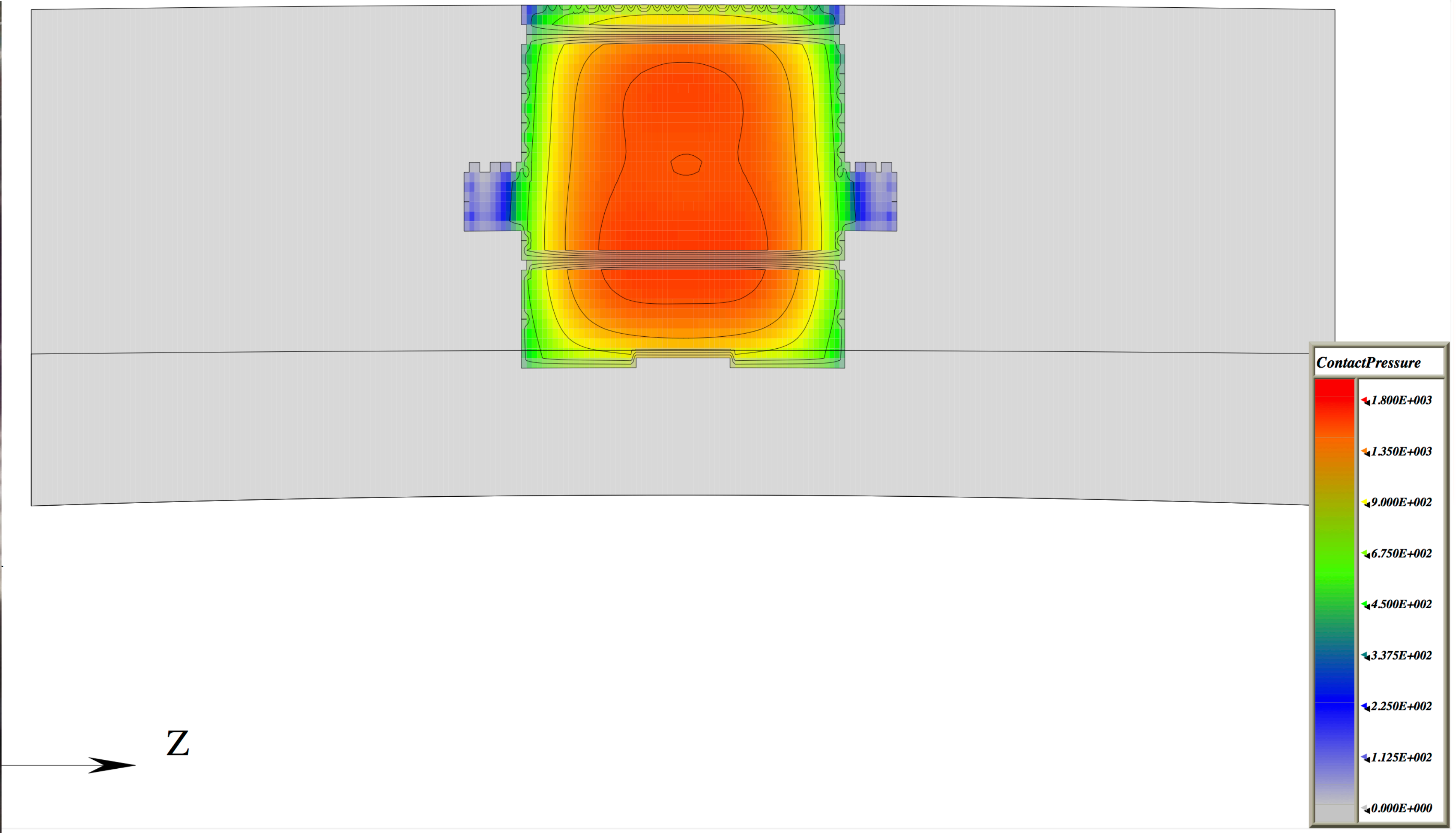
- Spur gear z=62, m=2mm, 1:1 ratio
- Gears and tribometer disks are made using 20MnCr5 steel (suitable both for carburizing and nitriding)
- Thermal treatments are applied
- Both gears and disks are grinded (large crowning, 100µm)
- Tumble finish is applied to both (Ra=0.2µm)
- WCC coatings are performed





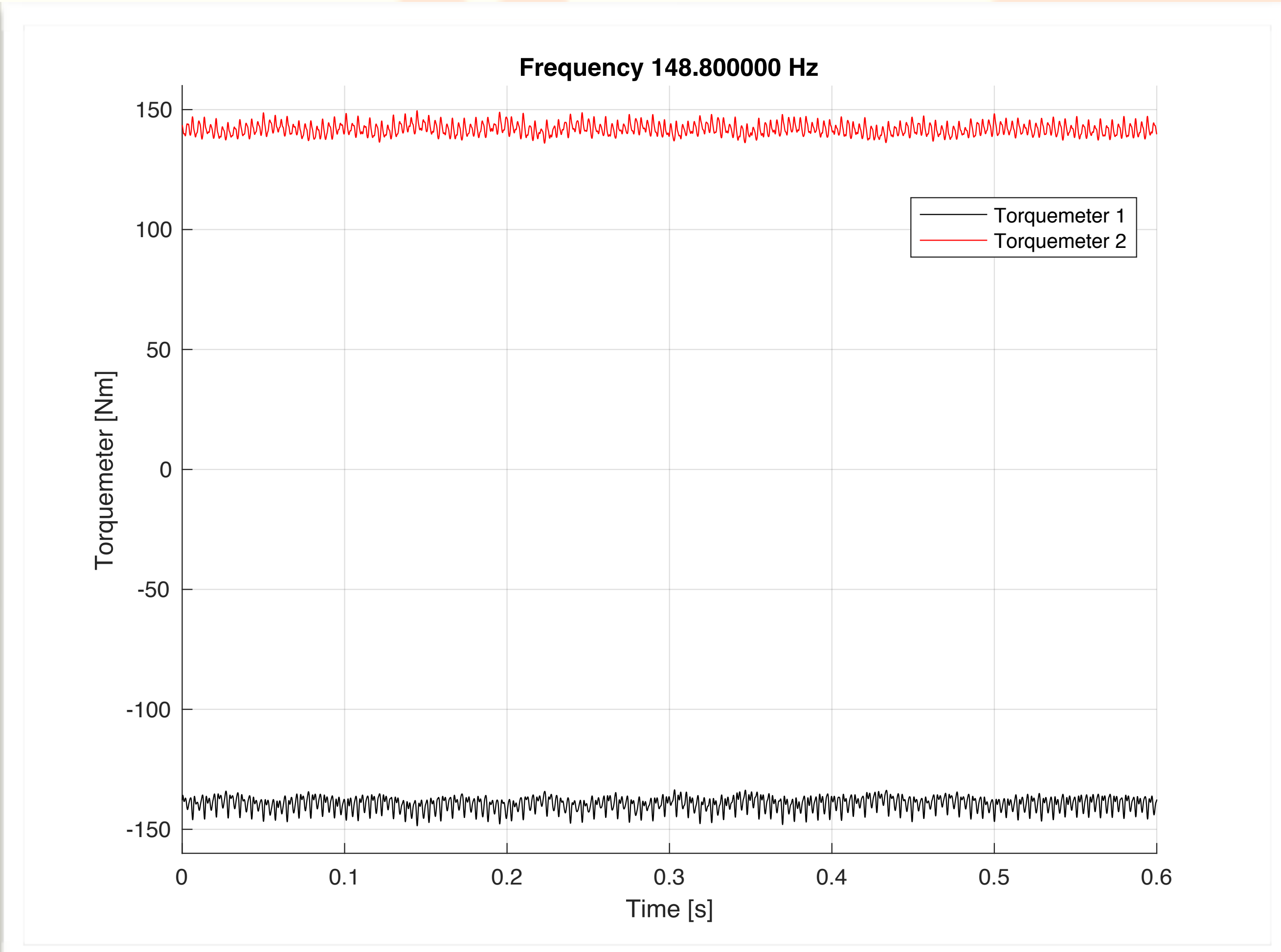
Load conditions

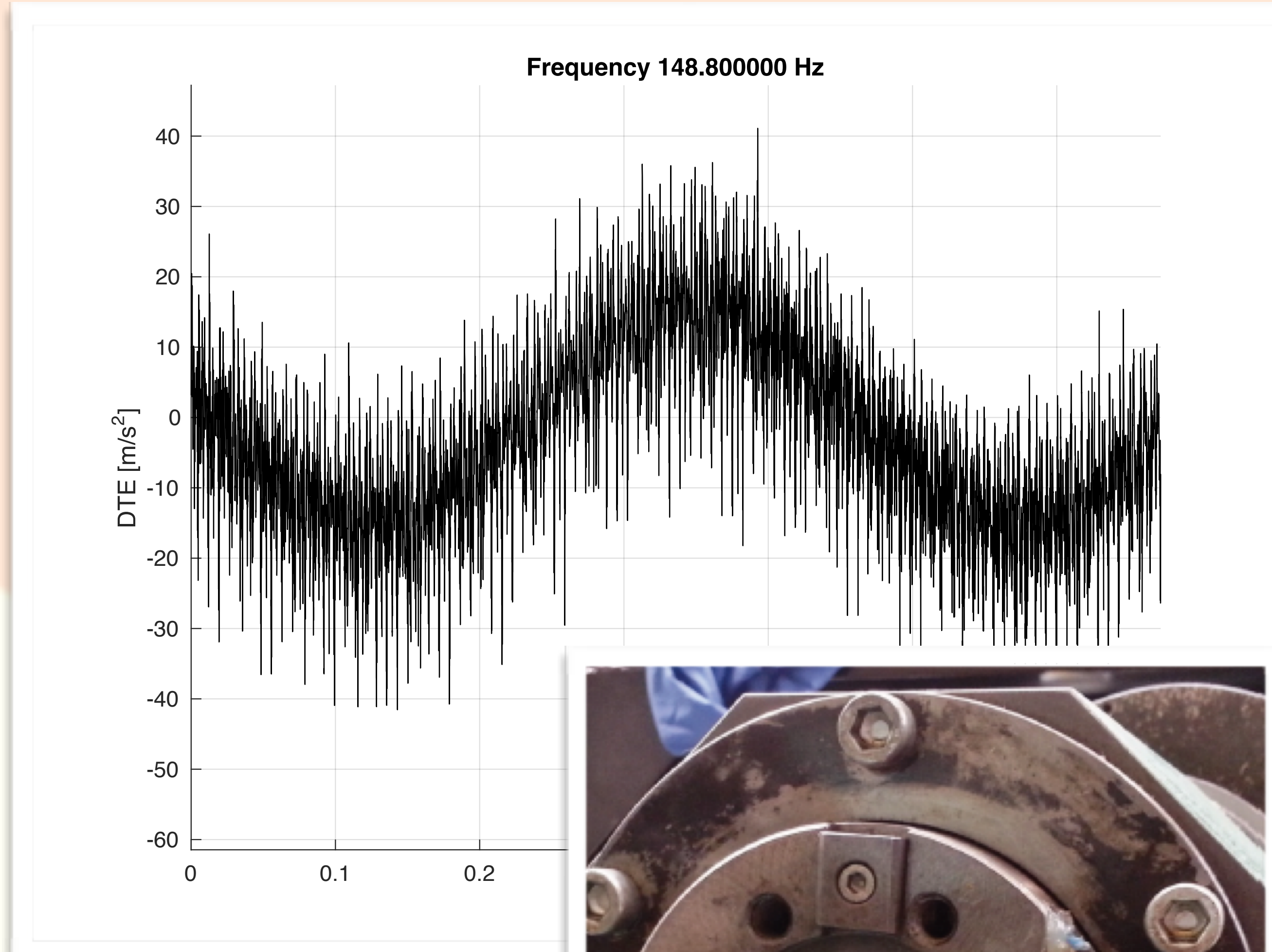




- 62-62 gear pairs are tested
- Maximum allowable load 240 Nm
- Nominal load 150 Nm, p_h=1.5GPa

Dynamic measurements

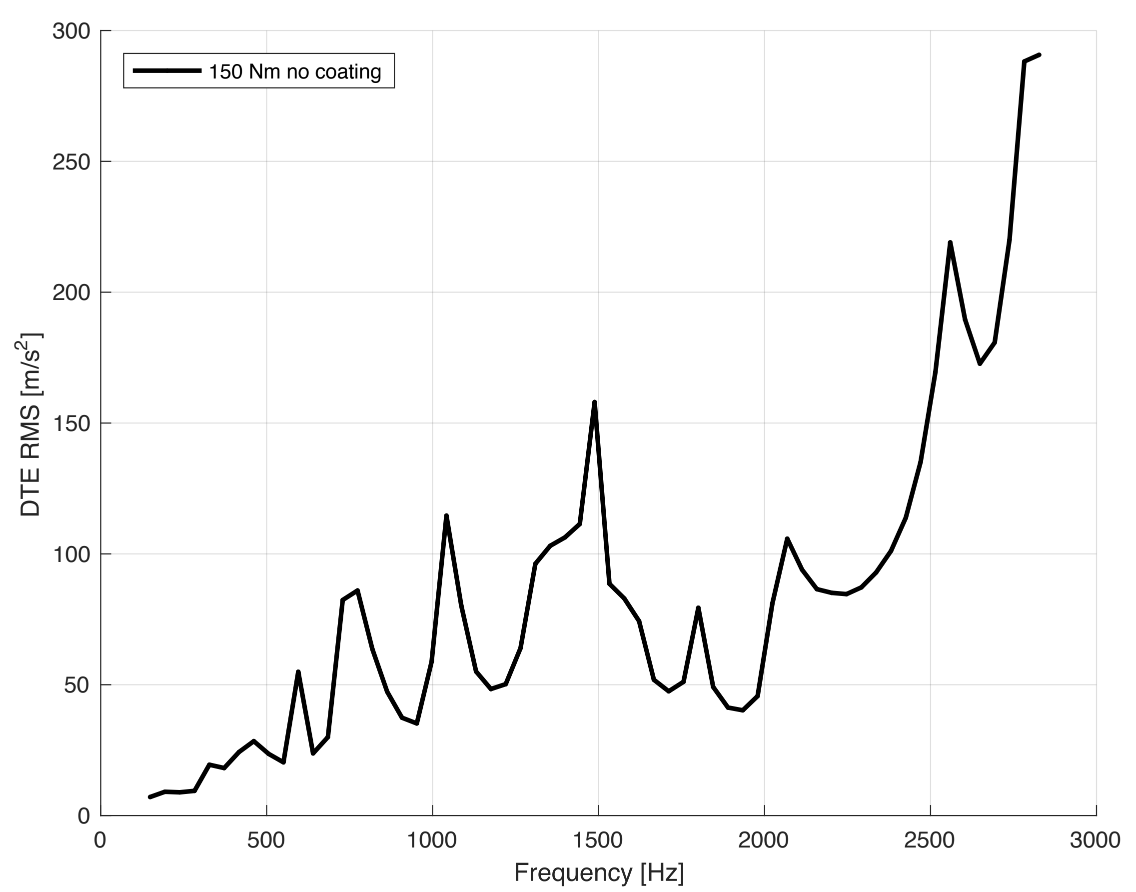


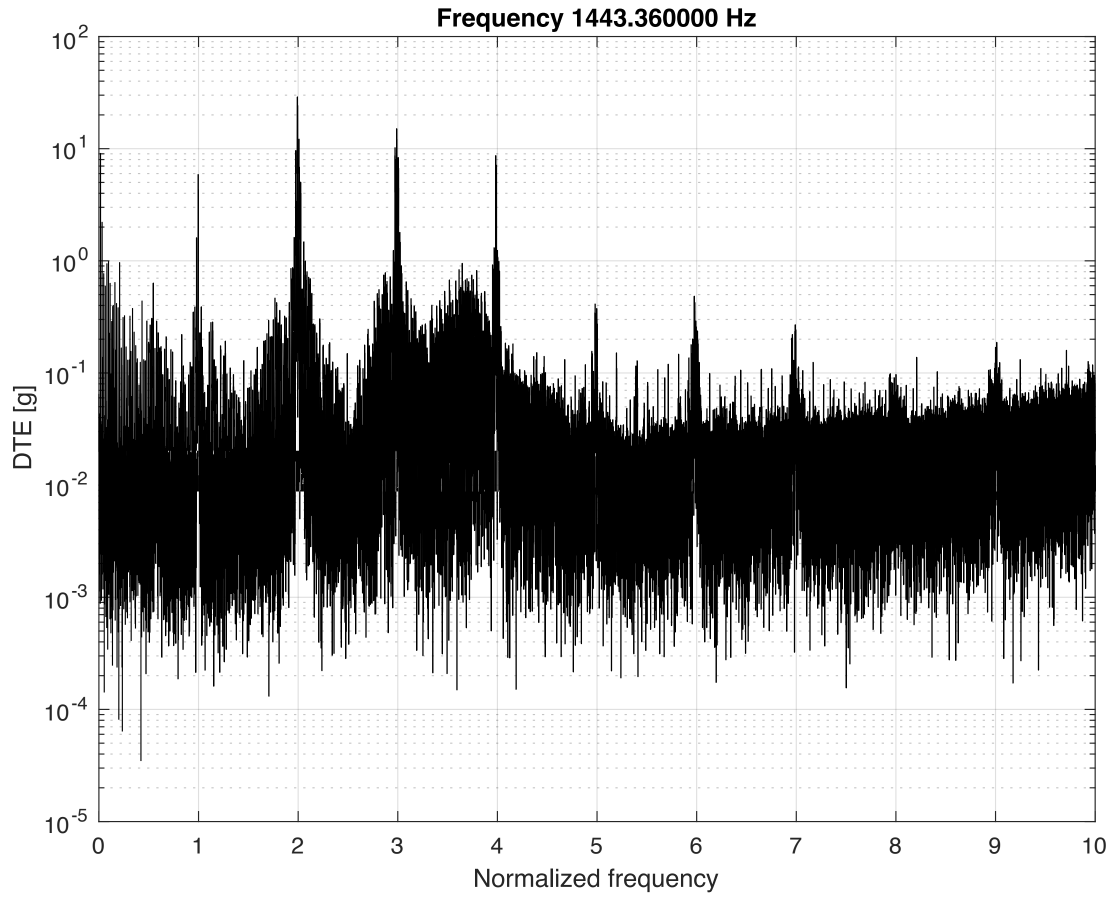


- Acceleration and torque signals are acquired simultaneously
- DTE is computed combining 4 acceleration signals

$$DTE = \frac{\ddot{x}_{11} + \ddot{x}_{12}}{2} \frac{R_{b1}}{R_{c1}} - \frac{\ddot{x}_{21} + \ddot{x}_{22}}{2} \frac{R_{b2}}{R_{c2}}$$

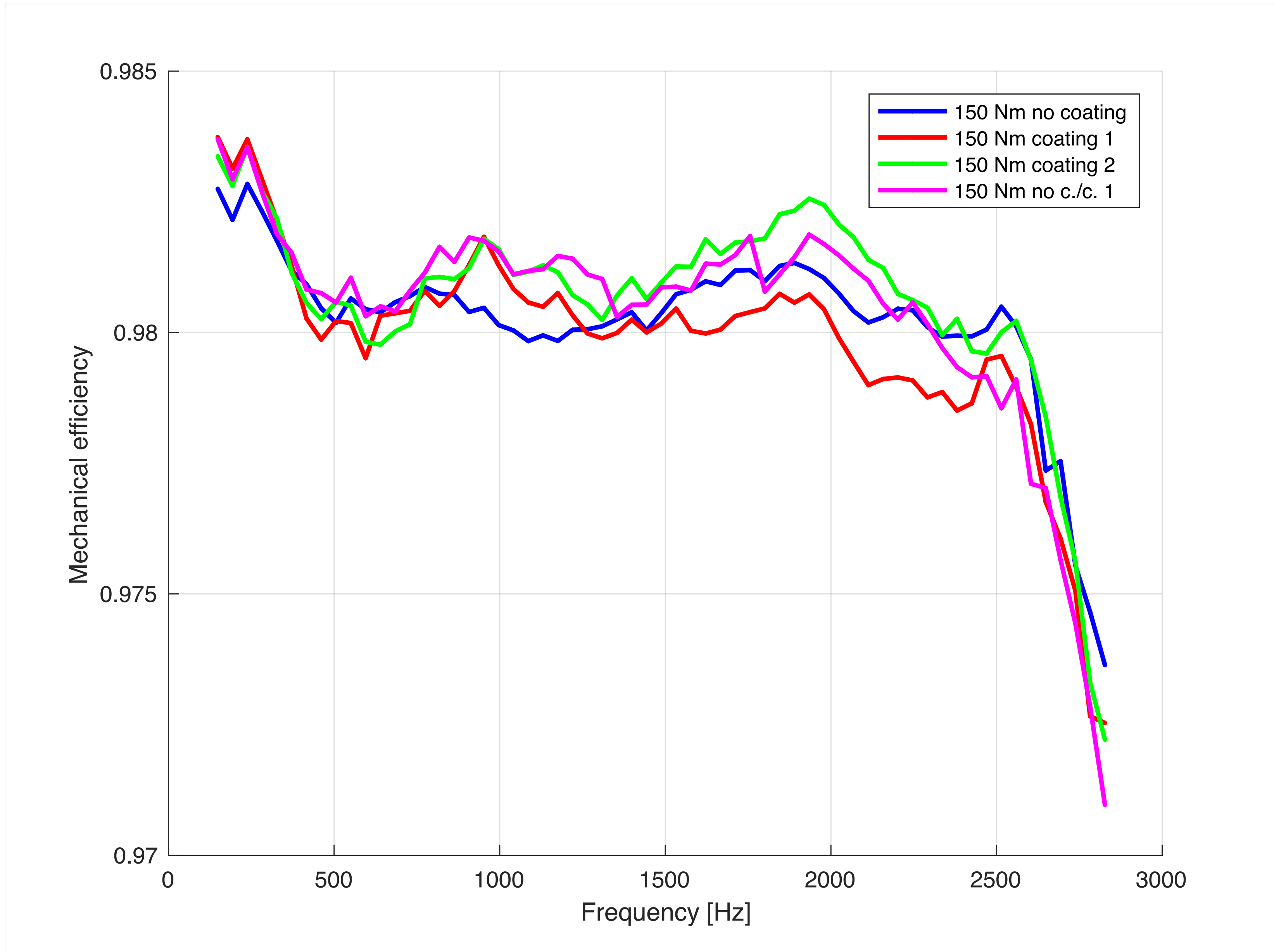
Dynamic measurements





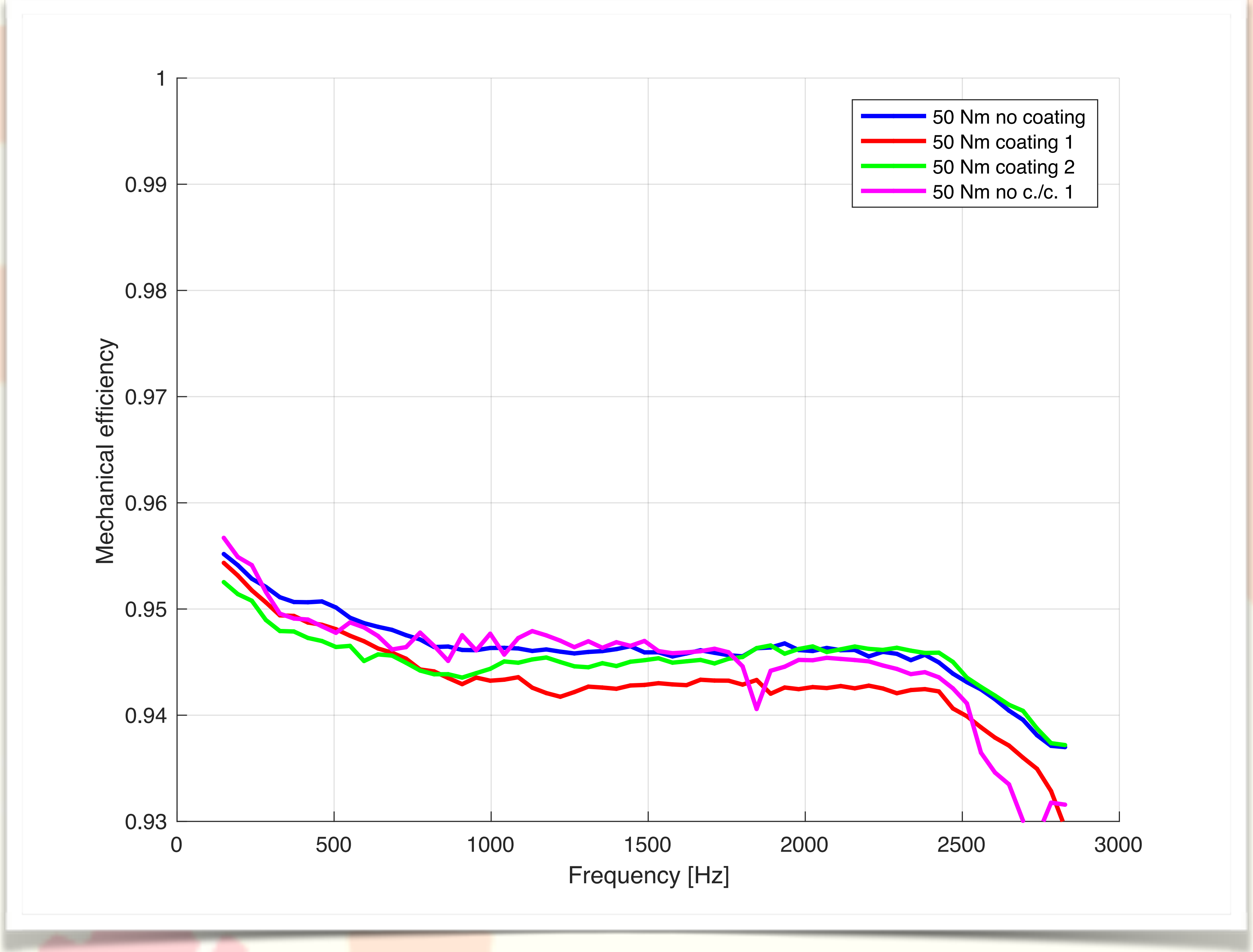
- 61 mesh frequencies (144-2736 RPM at pinion)
- Each speed lasts for 40 s (10 s acquisition)
- Efficiency is evaluated thanks to the torquemeters
- All gears are carburized

Efficiency 150 Nm



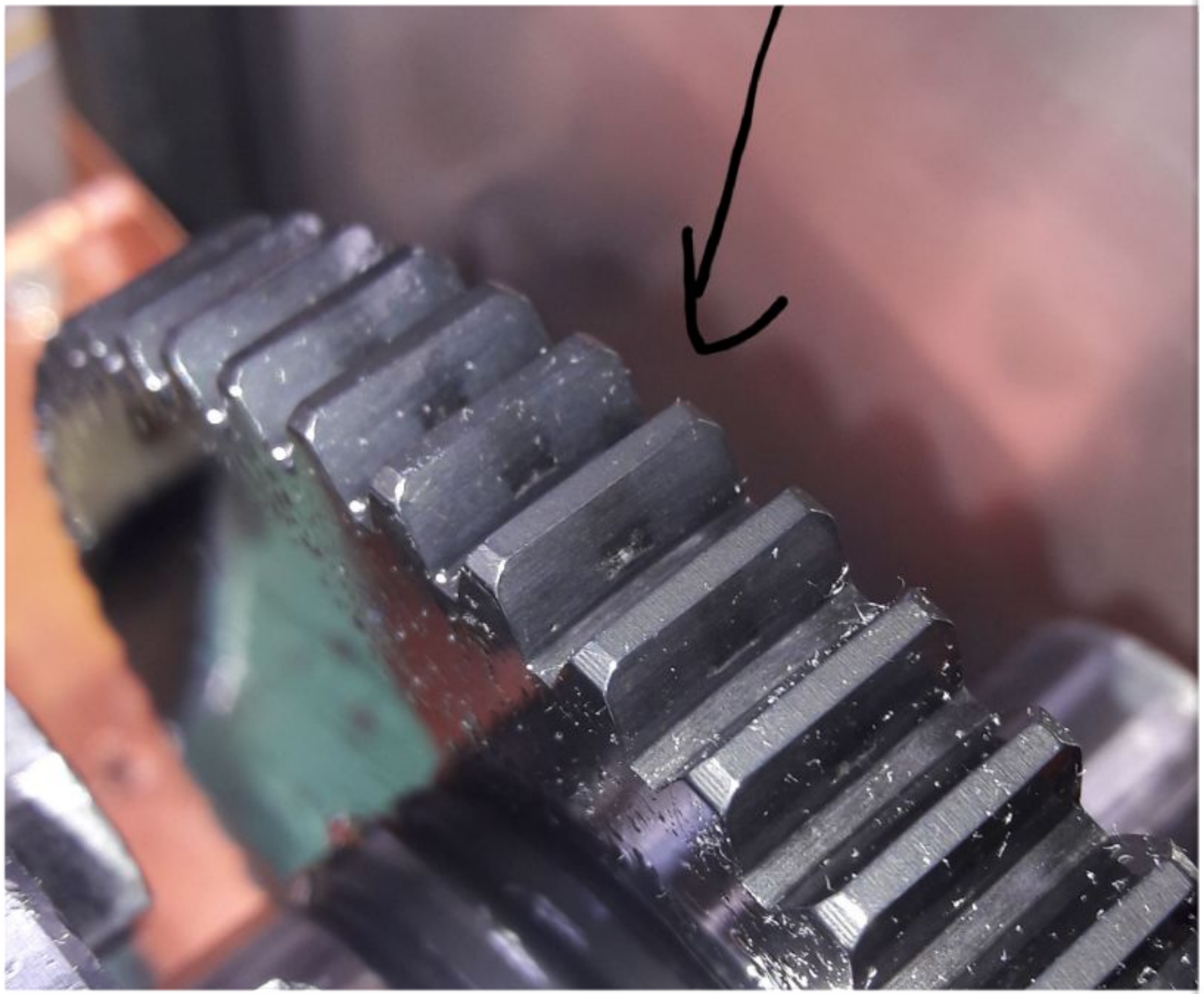
- Coating 1 and Coating 2 + Carburized are more efficient at the nominal torque
- At low speed Coating 2 and Coating 2 + Carburized are the most efficient

Efficiency 50 Nm



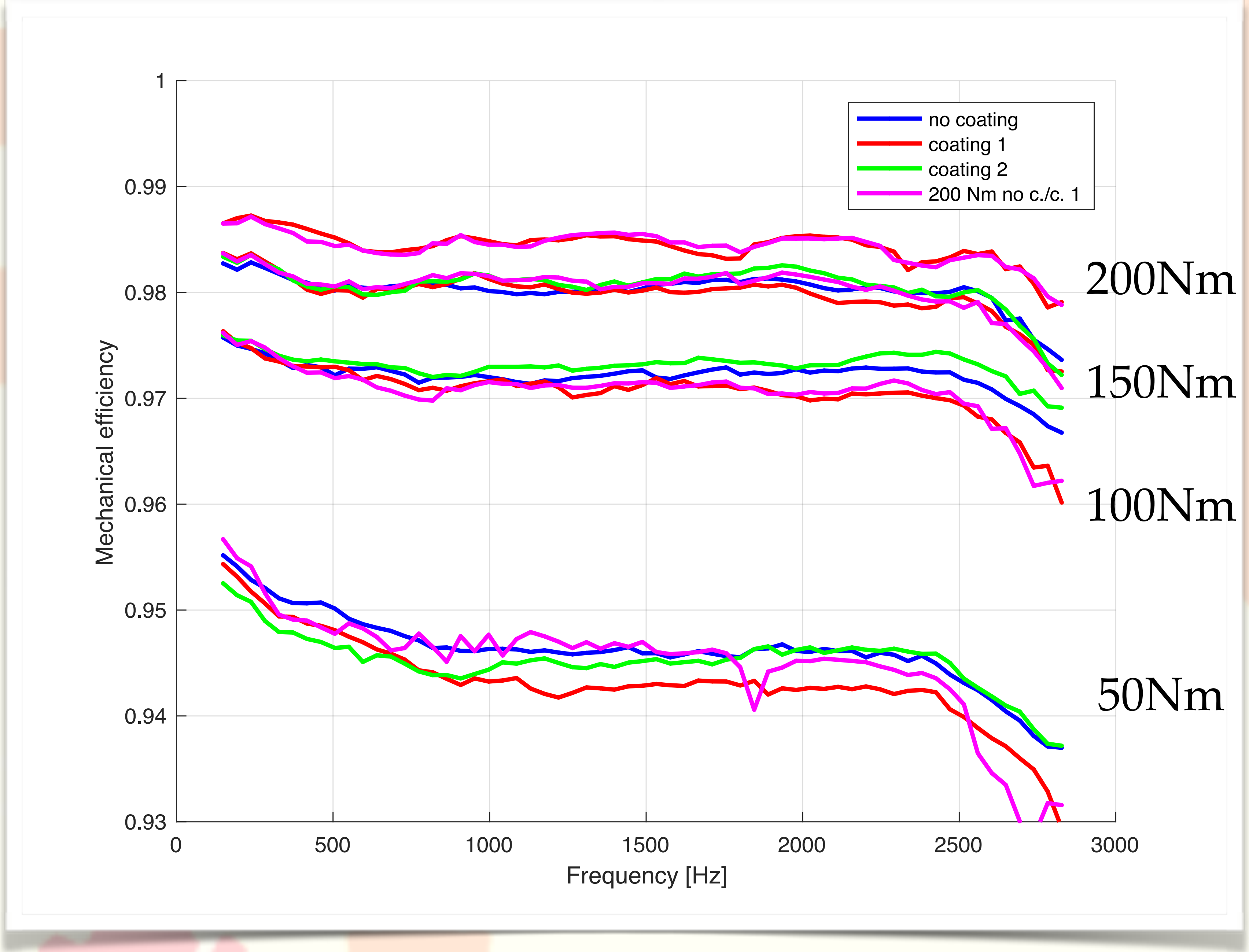
- Coating 1 and Coating 2 + Carburized are as much efficient as uncoated

Coating 2 failure



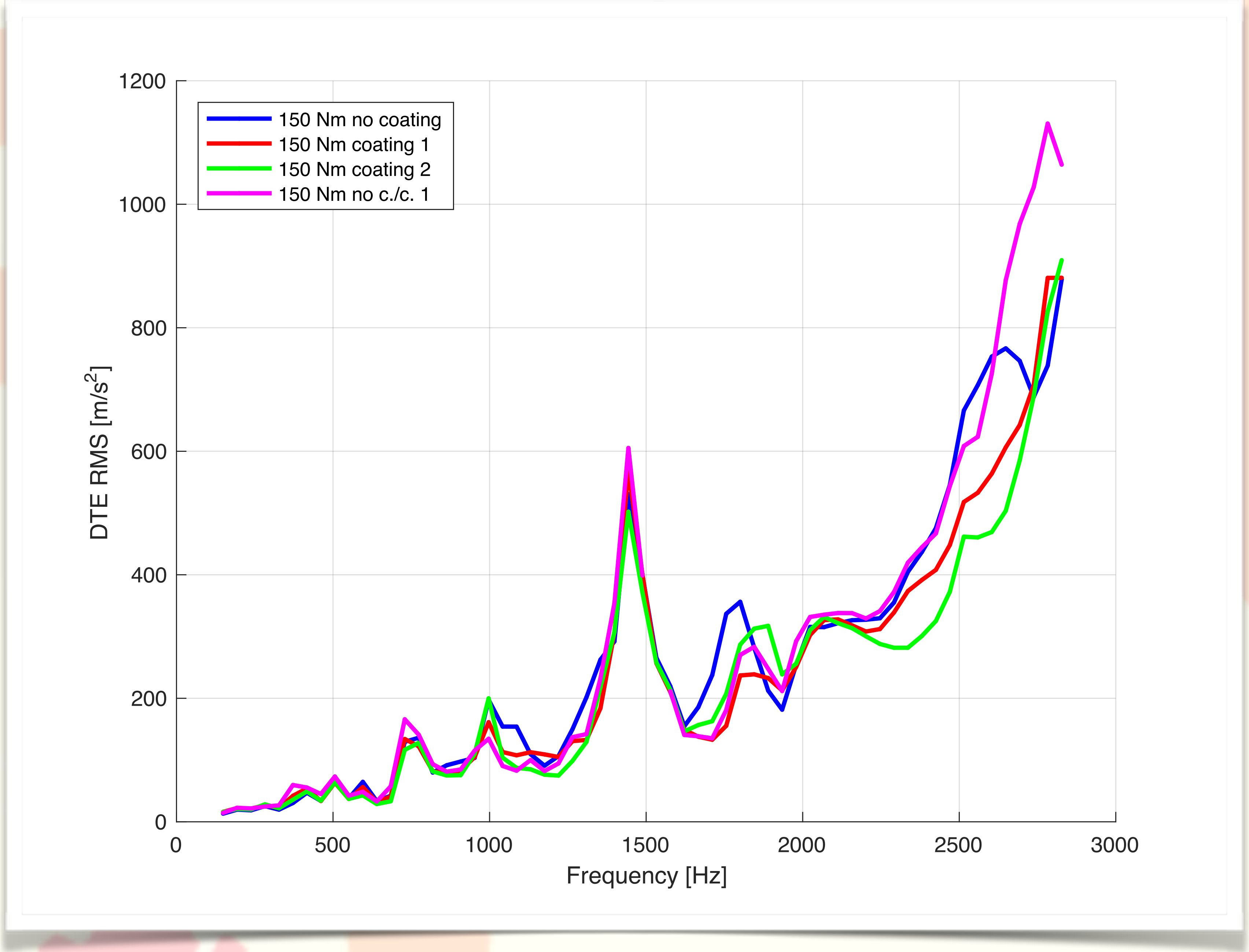
- Coating 2 failure occurs at 150 Nm
- WCC delamination is found close to the tip and root (failure mode: friction, low pressure)

Efficiency comparison



- Coating 1 and Coating 1 + Carburized are reliable up to 200 Nm
- As expected, efficiency increases with torque (churning losses are less relevant)

Efficiency comparison



- Coating 2 reduces vibrations in the gear pair, while coating 1 increases vibration levels
- Coating 1 + Carburized is the most noisy test case

Conclusions

- A comparative test for gear coatings and treatments has been set-up
- Tested WCC show slight efficiency increase with respect to uncoated cases
- Efficiency increase generally produces larger vibration amplitudes
- Optimal efficiency is given by a combination of coating 1 and uncoated pair (coating 2 delaminates at 1.5 GPa)
- In the final part of the project, the role of superfinishing is going to be investigated (as well as surface texturing)

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