DYNAMIC TESTING AND SIMULATIONS OF GEARS AND GEARBOXES

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and

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OptoPhemLab

Lab. Integrated Design and Simulation (LAPIS)

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PEOPLE:

Head of Centre InterMech MO.RE.

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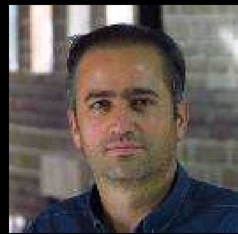

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Lab. Activities

Scientific Research

- European projects
- International cooperations
- Gear, Vibration
- Nanoscience, Materials
- Bio-engineering

Industrial Projects

- FP7 and H2020 programs
- Gear design, simulation and testing
- Vibration analysis and testing
- Development of new theories for thin-walled structures: SPACE APPLICATIONS
- **Service:**
  - Dynamic tests on shaking table
  - Modal analysis (numerical and experimental)
  - Vibration measurements
  - Complex gear train analyses

Education

- International student exchange
- International visiting Professor exchange
- Lab experience for students
- Courses:
  - ✓ Mechanical Vibrations
  - ✓ Multibody Dynamics
  - ✓ Mechanics of Vehicle

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- Europa (**UK**, **France**,...)
- East-Europe/Asia (**Ukraine**, **Russia**, ...)
- Middle East (**Iran**)
- Oceania (**New Zealand)**

- Waitako University (**Hamilton**)
- Semenov Institute (Before:2022, **Moscow**)
- Gear Lab, OHIO State University (**USA**)
- Loughborough University (**UK**)
- McGill University (**CA**)
- INSA Lyon
- KPI University (**Kharkov**)
- Kerman University (**Iran**)

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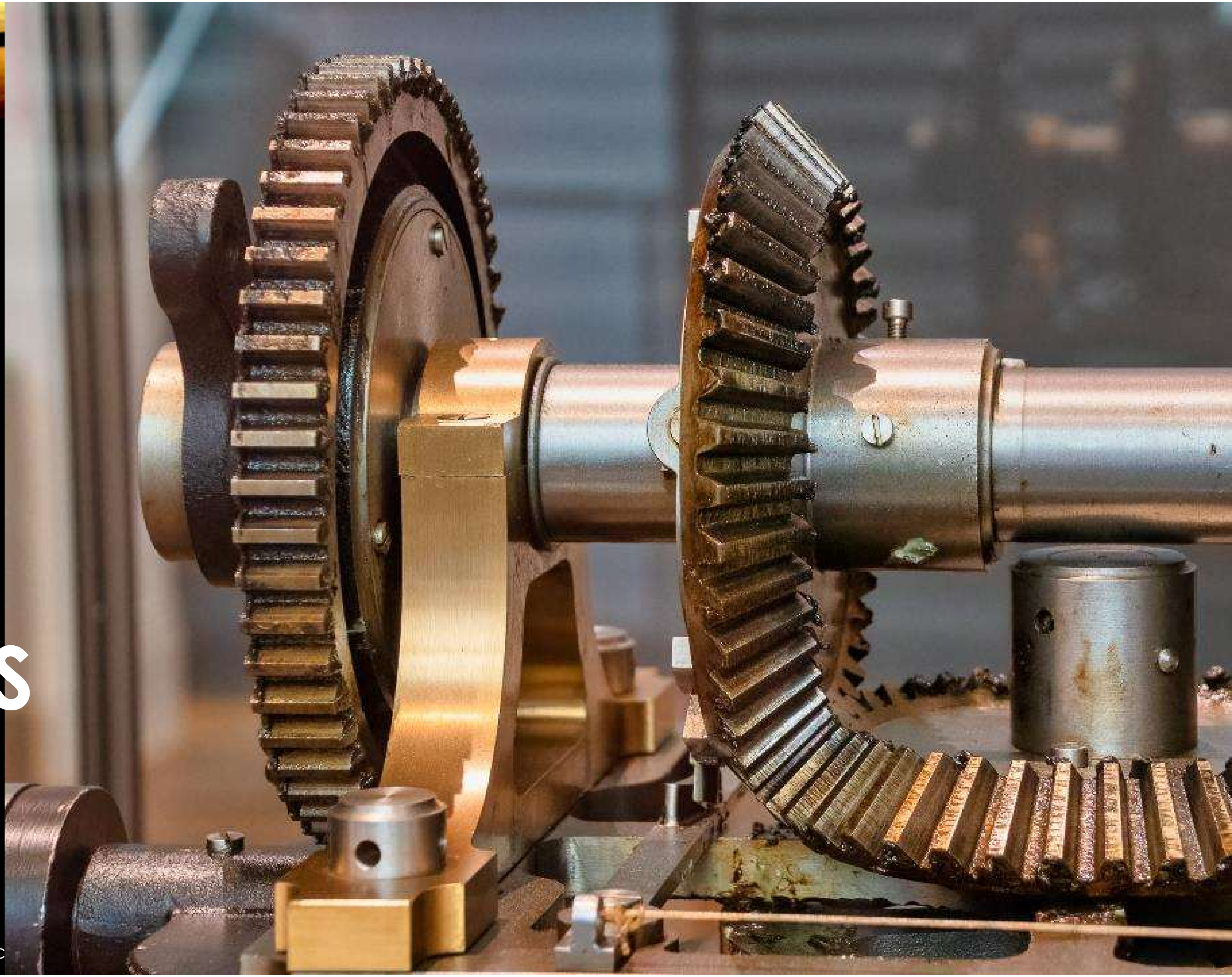
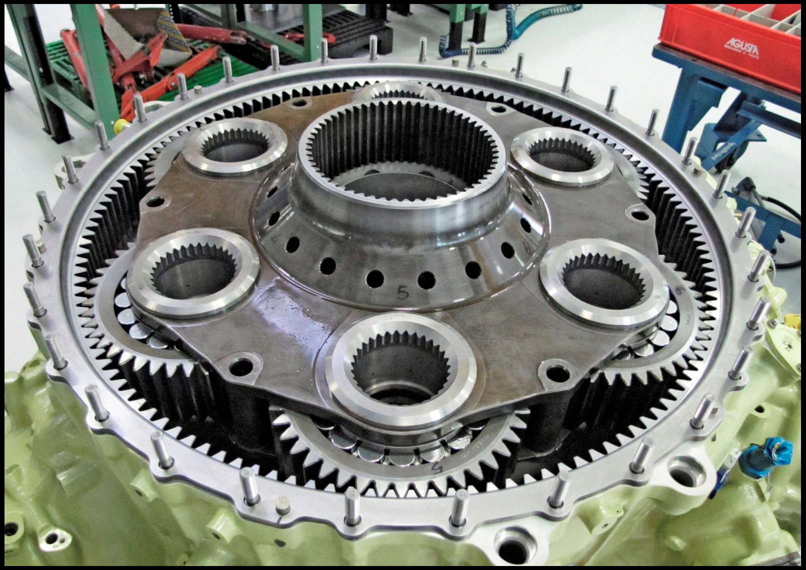
GEARS and GEARBOXES



From the Unimore Course for Engineers:
Design and simulation of gears and transmissions: NVH issues



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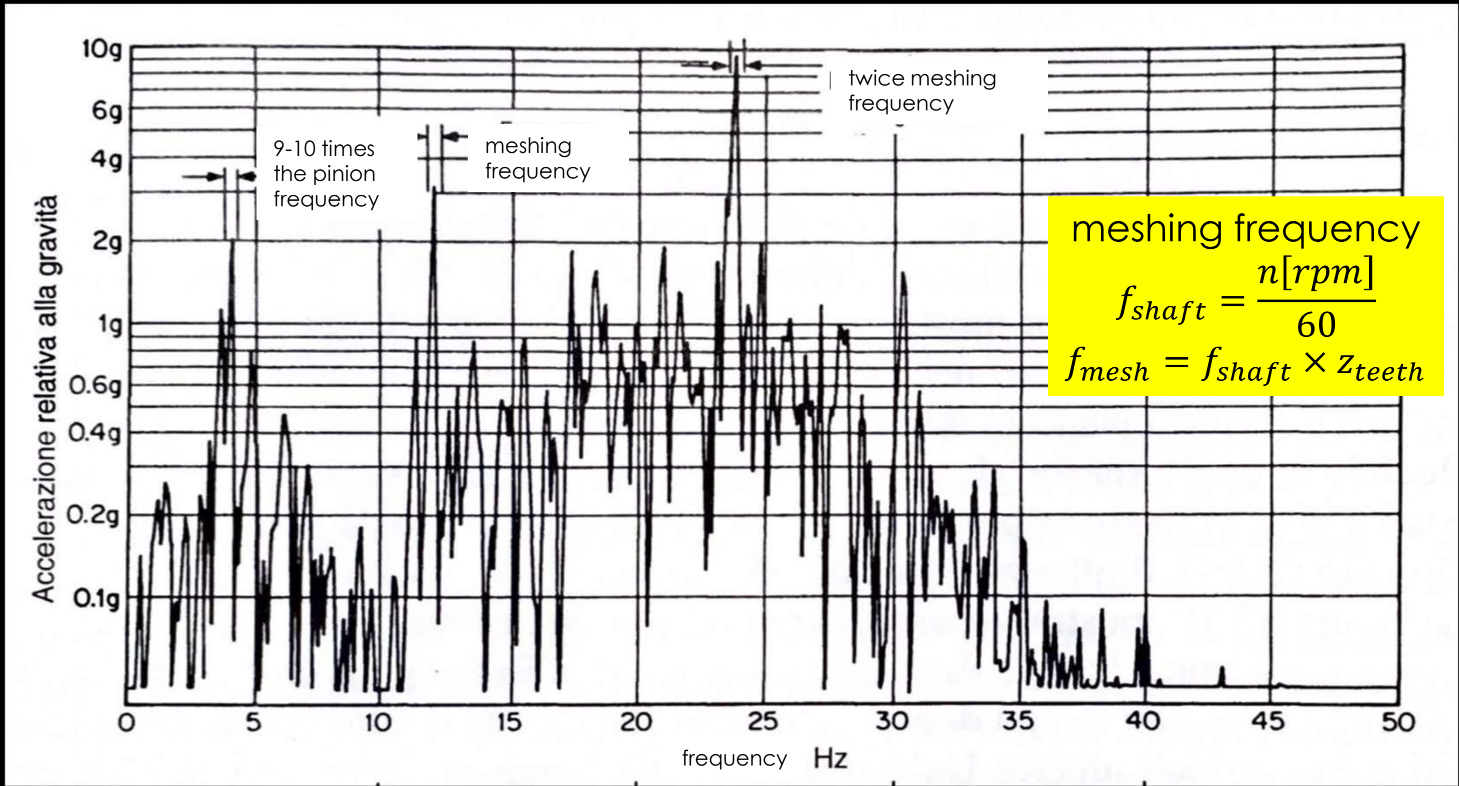
Example of vibration spectrum of a gearbox



meshing frequency

$$f_{shaft} = \frac{n[rpm]}{60}$$

$$f_{mesh} = f_{shaft} \times z_{teeth}$$

This gearbox had a failure due to wear, but the vibrations were not high

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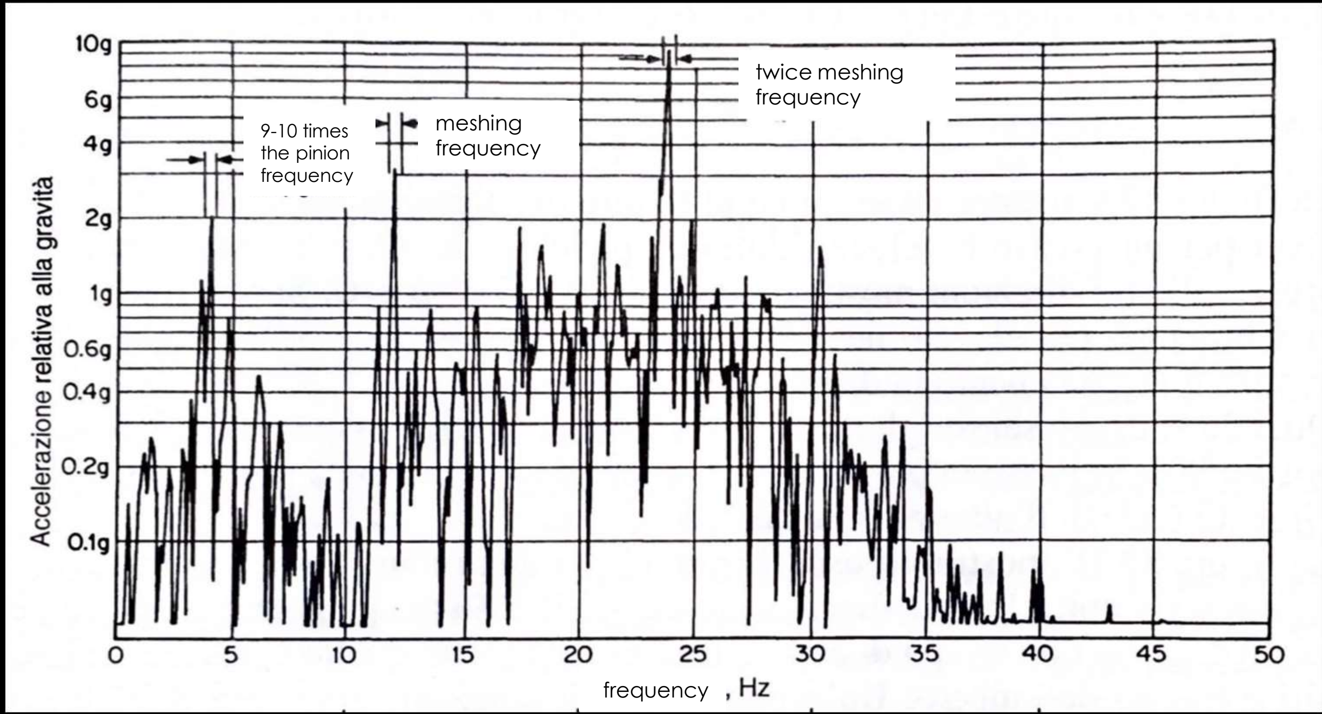
Example of vibration spectrum of a gearbox

- Normally measurements are carried out on different points with triaxial accelerometers
- Peaks at the meshing frequency and its double
- A peak at 9 times the pinion frequency
- There was still wear on 9 teeth (9 bumps)
-



- The accelerations themselves do not appear worrying as a level
- The peak at 9× pinion frequency is disturbing (Dudley)
- There is no reason for the presence of this peak except the damage

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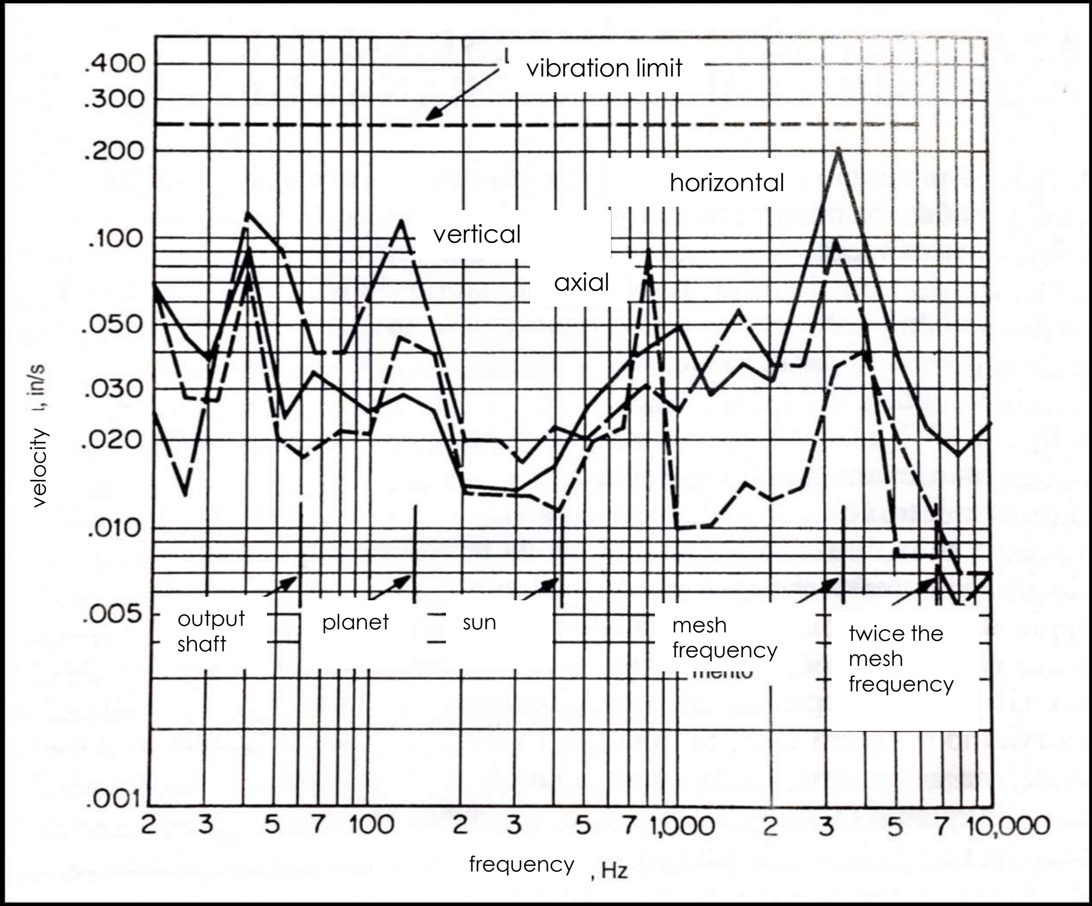
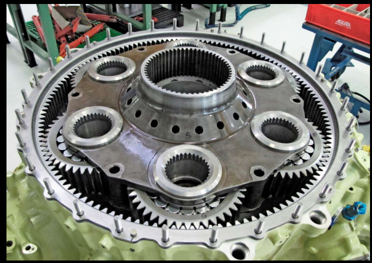
NVH ISSUES

Example of vibration spectrum of a planetary gearbox

- Proximity sensor measurements
- The maximum is at the meshing frequency
- At twice the mesh frequency there is nothing
- Interesting the different character of the various spectra

$$\omega_{mesh} = \omega_{sun} \frac{z_{sun} z_{ring}}{(z_{sun} + z_{ring})}$$





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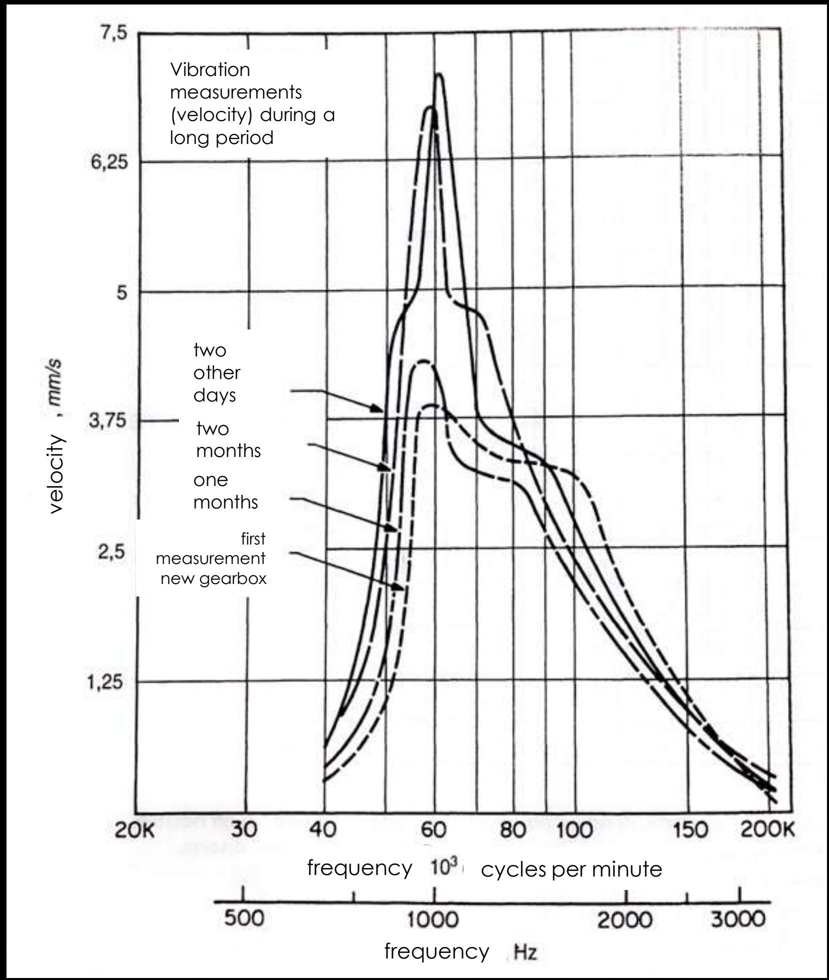
Example of vibration spectrum of a gearbox with failures

- Effect of progressive failures
- As soon as it was put into service, the first survey was carried out and the spectrum was recorded.
- A progressive increase in vibrations was observed.
- Given the sudden increase in vibrations, it is decided to take it out of service



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VIBRATIONAL LIMITS

- Gear applications range from nano to macro scale, with powers ranging from fractions of W to thousands of kW
- Vibration limits are necessarily variable
- The type of gear (e.g. straight or helical teeth) greatly influences the levels
- The gear material influences levels as well as treatments and machining
- The material of the housing influences the outer vibration levels (cast iron dampens more than steel, aluminium dampens less)

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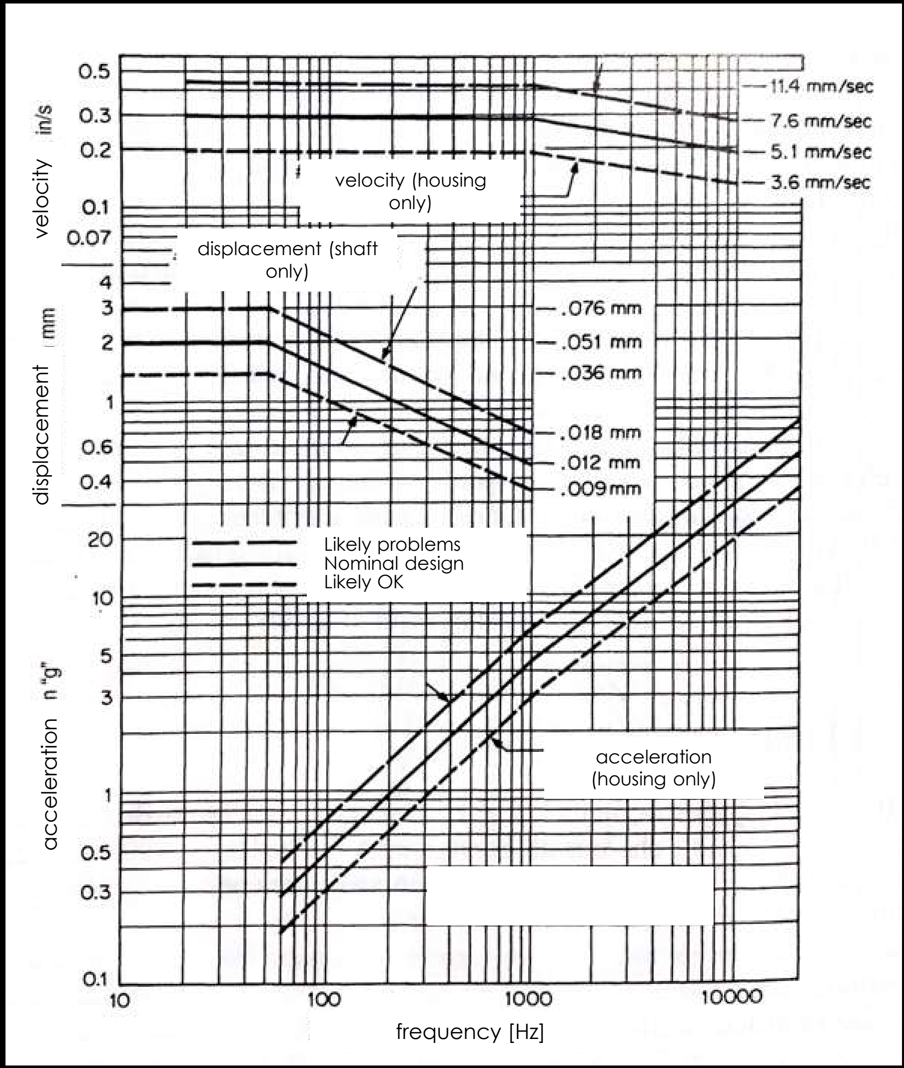
VIBRATIONAL LIMITS

- Normogram suggested by Dudley
- Top curve: suspected problems
- Continuous curve: smooth operation
- Dashed curve: tightly controlled gear



Note: Displacement measurements lose meaning at high speeds because sensors cannot dynamically measure much below microns

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VIBRATIONAL LIMITS

Factors influencing the vibration limits

Factor	Increases the vibration limits	Reduces the vibration limits
Durability (full power)	Less than 2000h	More than 20000h
Material hardness	38HRC or less (hardened steel)	50-65 HRC (case-hardened)
Type of gear	Spur or bevel gear	Helical or Spiral bevel gear
Accuracy of teeth	Average gears (machined)	High accuracy (grinding)
Peripherical primitive speed	Less than 25.4 m/s	More than 25.4 m/s
Max power	Less than 400kW	More than 1500kW
Weight	Light	Heavy

Example (Dudley)
A lightweight planetary gearbox of 10,000 Hz and 1000kW can see 40g and last 30,000 hours (note: 10,000Hz for 33 teeth -> 18,000rpm)

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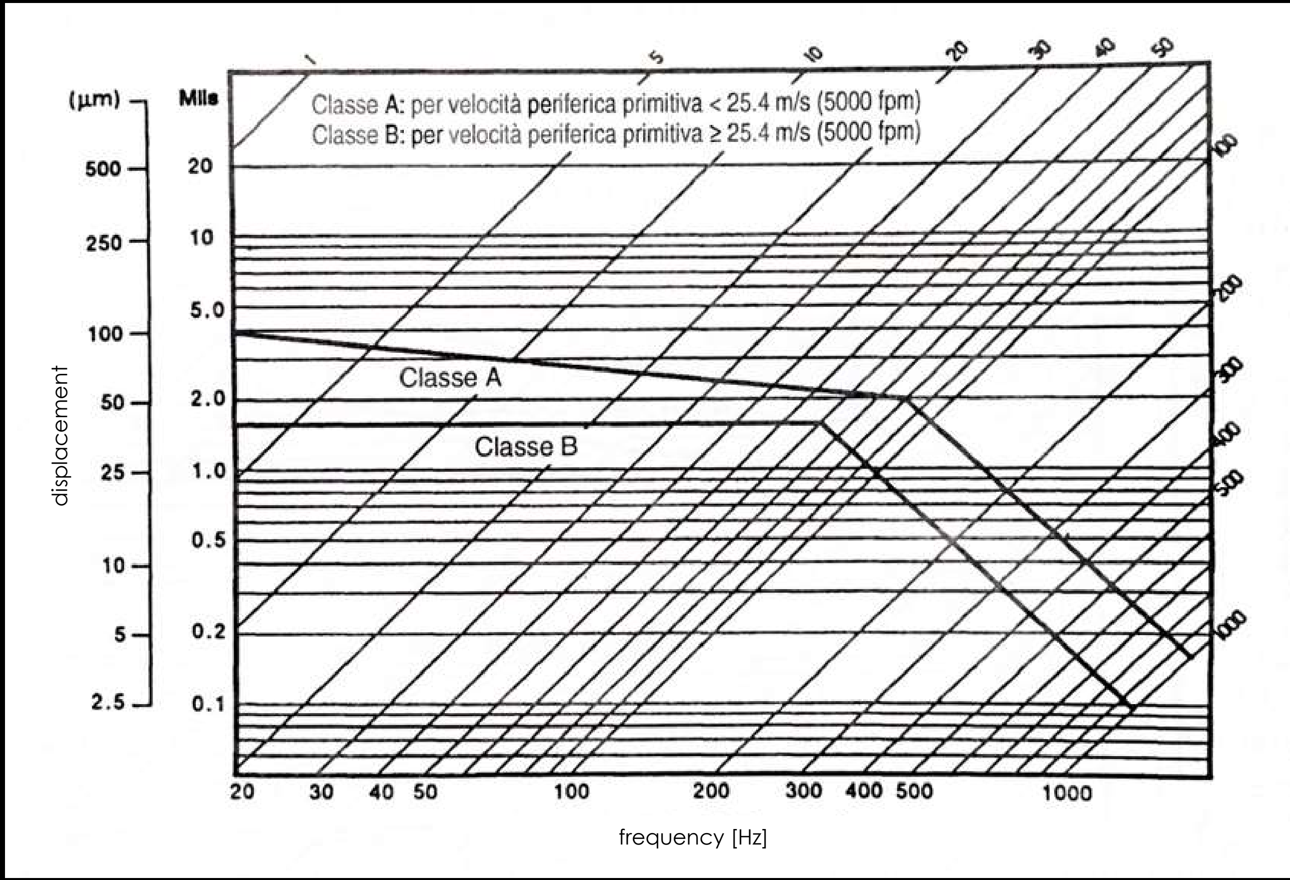
VIBRATIONAL LIMITS

Displacement amplitude normogram (AGMA 6000-A88)



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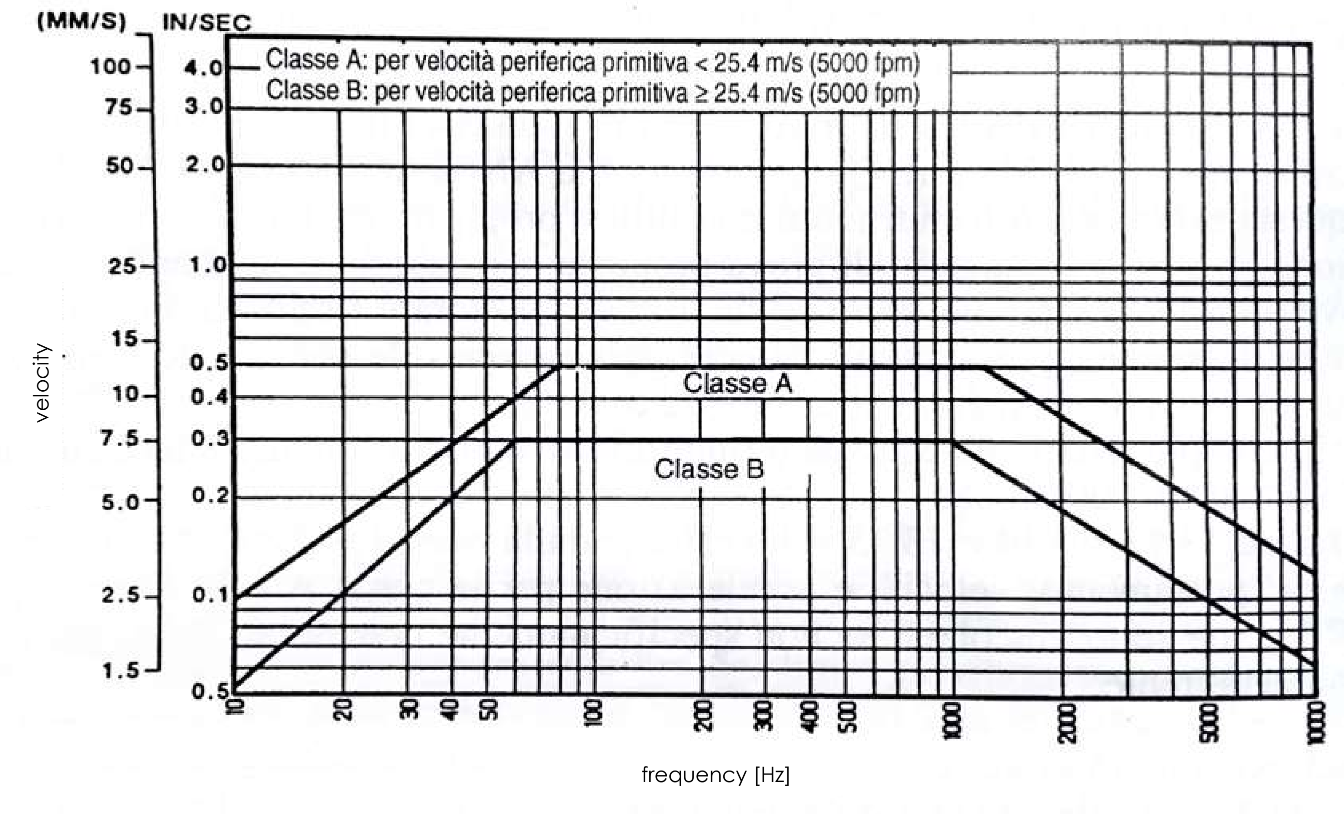
VIBRATIONAL LIMITS

Normogram speed amplitudes (AGMA 6000-A88)



fpm: feeth per minute

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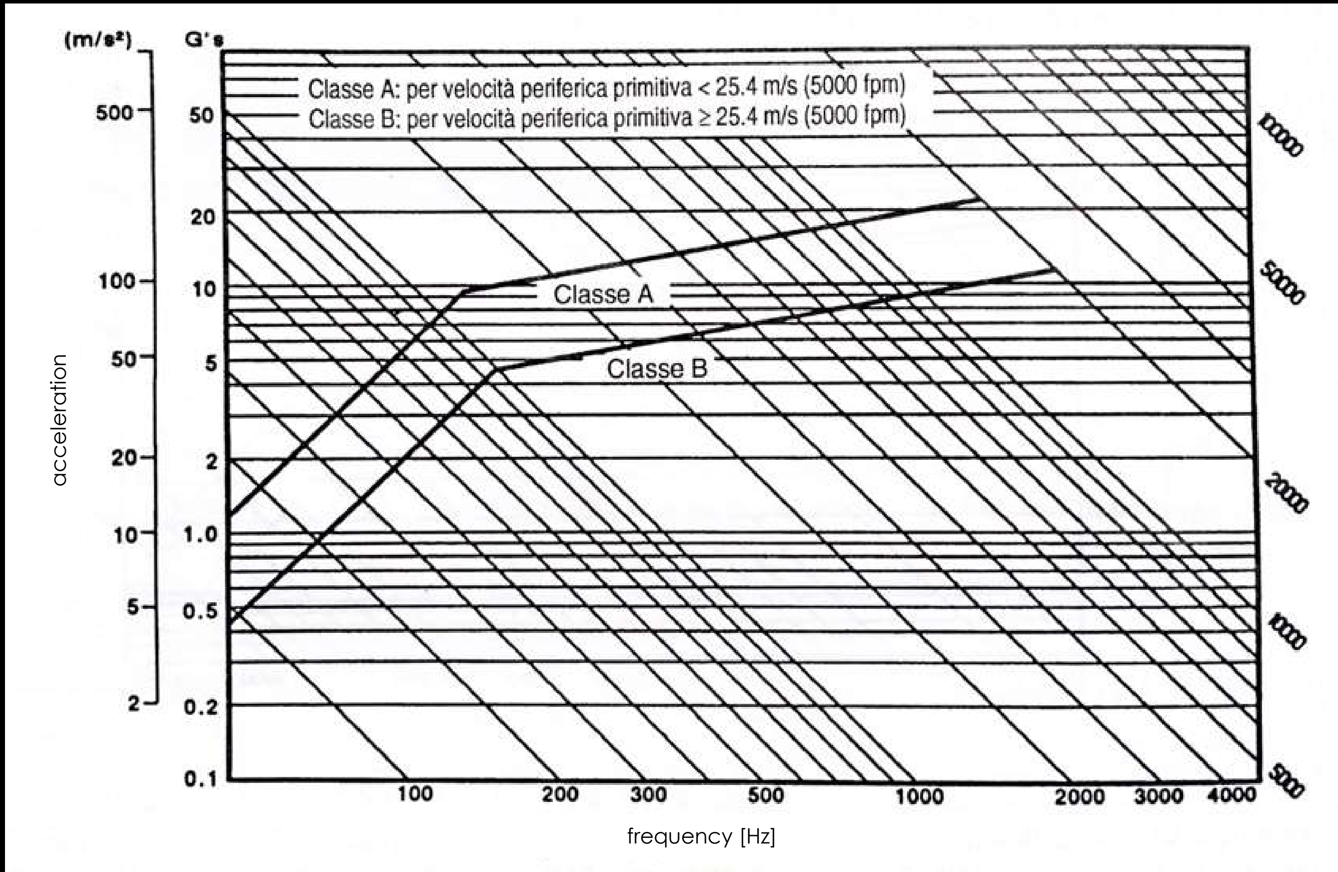
VIBRATIONAL LIMITS

Normogram amplitudes in acceleration (AGMA 6000-A88)



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QUALITY CLASSES: COMPARISON OF STANDARDS

more accurate

less accurate

STANDARD	CLASS												
AGMA 390.03 (USA)	15	14	13	12	11	10	9	8	7	6	5	4	3
DIN 3962 (GERMANIA)	1	2	3	4	5	6	7	8	9	10	11	12	
ISO 1328 (ITALIA)	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	
BS 4 (REGNO UNITO)					A^1	A^2	B	C	D				
3SEIS (FRANCIA)				A	B	C	D	E					
JIS B 1702 (GIAPPONE)				0	1	2	3	4	5	6	7	8	
KS B 1405 (COREA)				0	1	2	3	4	5	6	7	8	

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Quality class: standards

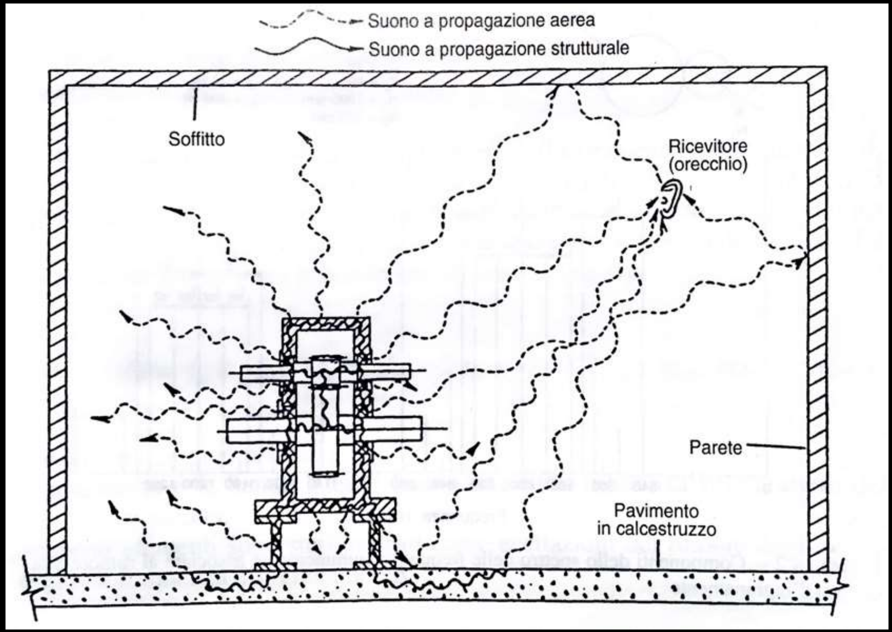
- ISO 1328-1
  - Pitch
    - Single Pitch Deviation fpt
    - Cumulative Pitch Deviation fpk
    - Total Cumulative Pitch Deviation fp
  - Profile error
    - Total Profile Deviation fa
    - Profile Form Deviation ffa
    - Profile Slope Deviation fha
  - Helix
    - Total helix deviation fb
    - Helix Form Deviation ffb
    - Helix Slope Deviation fhb

- ISO 1328-2
  - Total Radial Composite Deviation (Fi)
  - Runout Error of Gear Teeth (Fr)

VIBRATION AND NOISE

Paths

- The vibration source is produced by the meshing
- The vibration is transferred through forces on the shafts
- Then it is transferred to the housing
- Finally, the housing radiates sound into the environment



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VIBRATION AND NOISE

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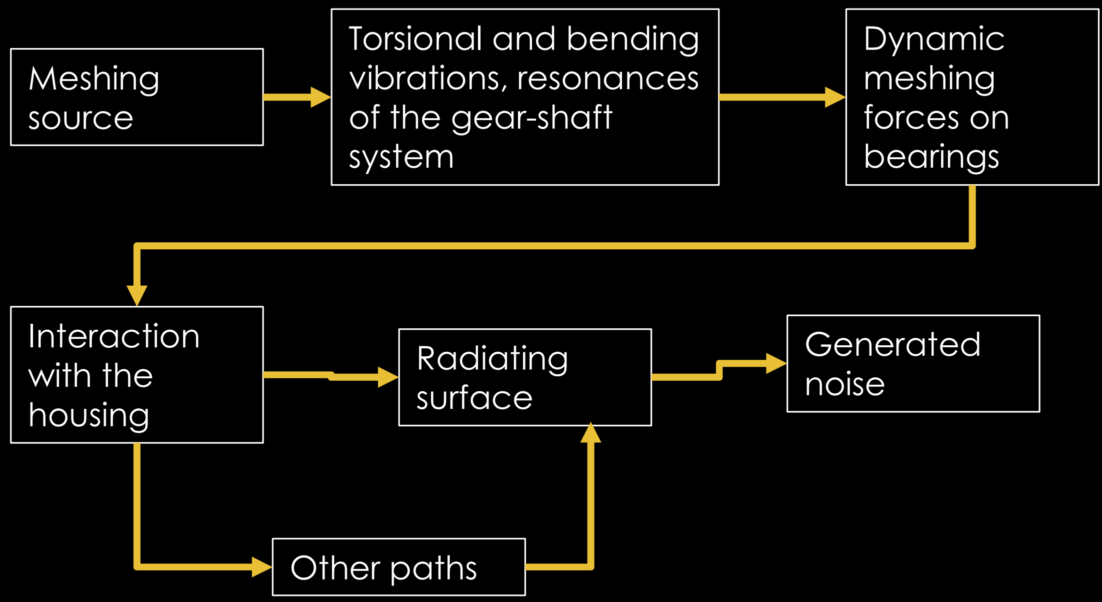

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NVH ISSUES

Suggestions to reduce emitted noise (Houser)

- Reduce gear excitation
- Imperfect action between conjugated surfaces causes vibration: OPTIMIZE MICRO-GEOMETRY
- Reduce vibration transmission paths between gear and casing
- Reduce the acoustic emissivity of the housing
- Change the environment where the gearbox operates: isolation and confinement devices

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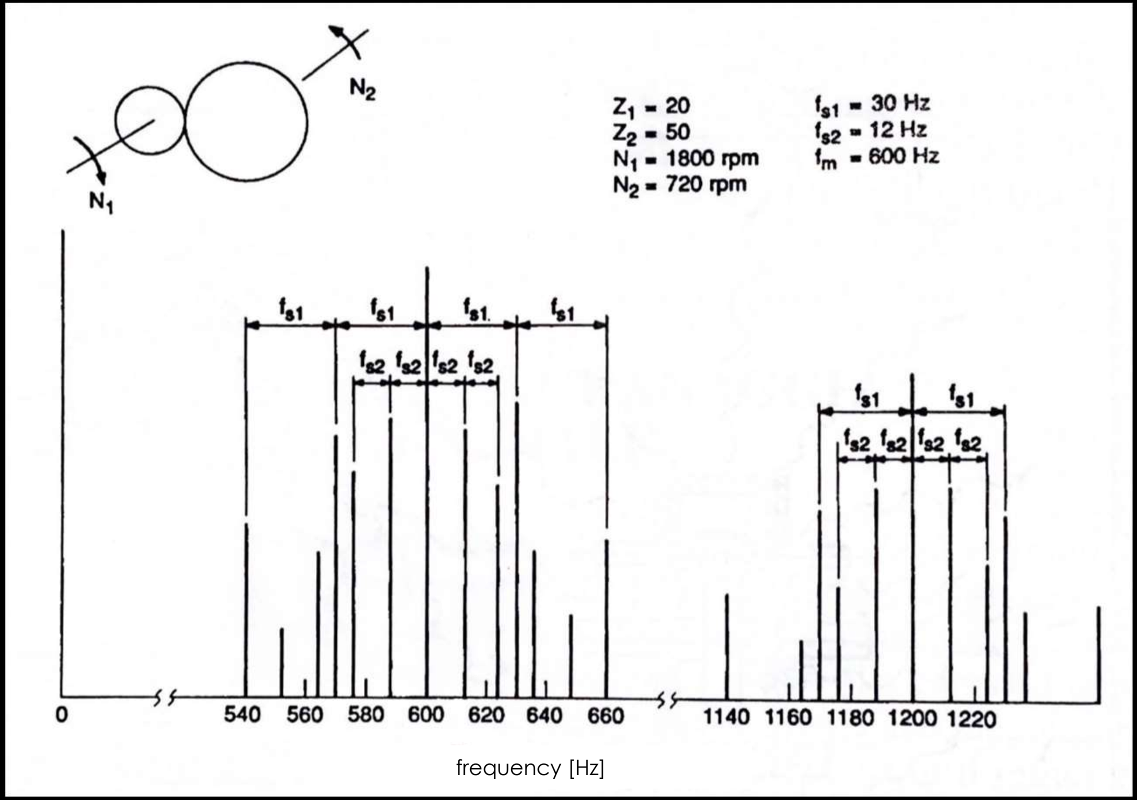
VIBRATION AND NOISE

NVH ISSUES

Most of the noise is related to:

- Mesh frequency
- Multiples of the gear frequency
- Side-bands (quasiperiodicity or amplitude modulation)



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VIBRATION AND NOISE

NVH ISSUES

Most of the noise is related to:

Shaft frequency 1

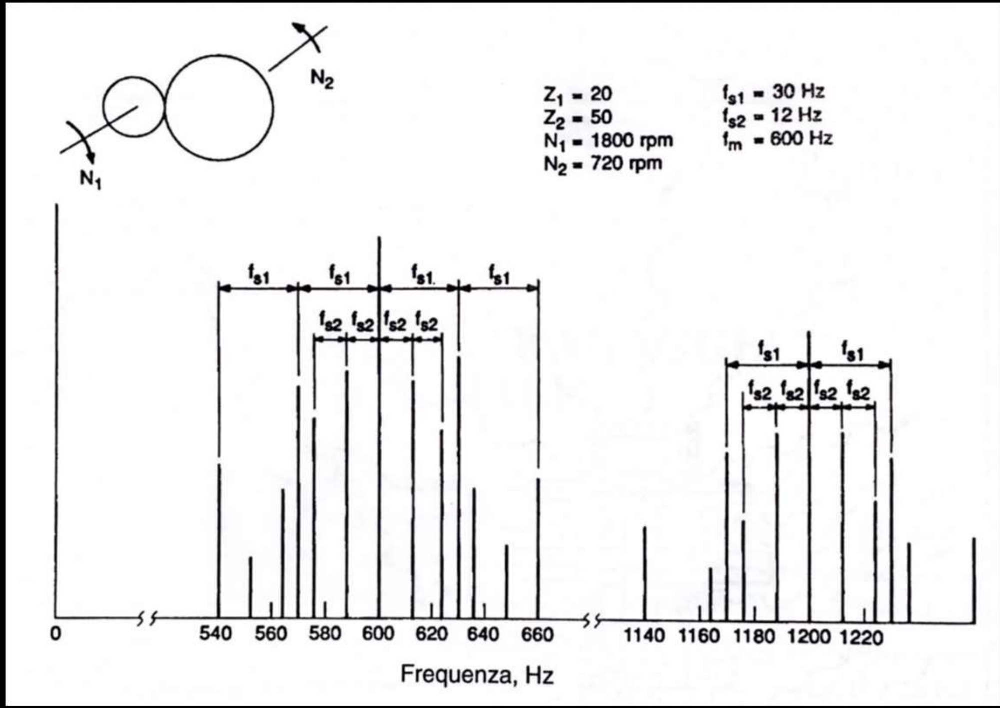
$$f_{s1} = \frac{N_1}{60}$$

Shaft frequency 2

$$f_{s2} = \frac{N_2}{60} = \frac{f_{s1} z_1}{z_2}$$

Mesh frequency

$f_m = f_{s1} z_1 = f_{s2} z_2$



N_i Shaft rotation speed i z_i number of teeth wheel i

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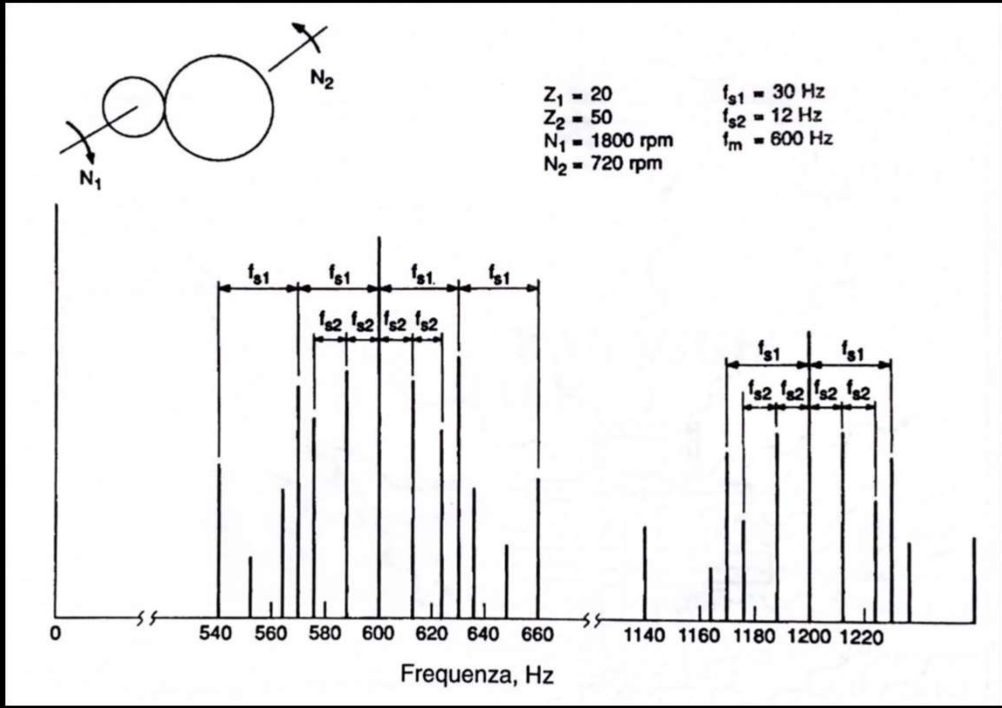
NVH ISSUES

- Peaks in the spectrum (harmonics) are determined at integer multiples of the meshing frequency f_m, $2f_m$, $3f_m$, $4f_m$
- Sidebands generally depend on the rotation frequency of the wheel:



$$f_m \pm n f_{s1}$$

$$f_m \pm n f_{s2}$$

$$2f_m \pm n f_{s1}$$

$$2f_m \pm n f_{s2}$$

And so for every harmonics

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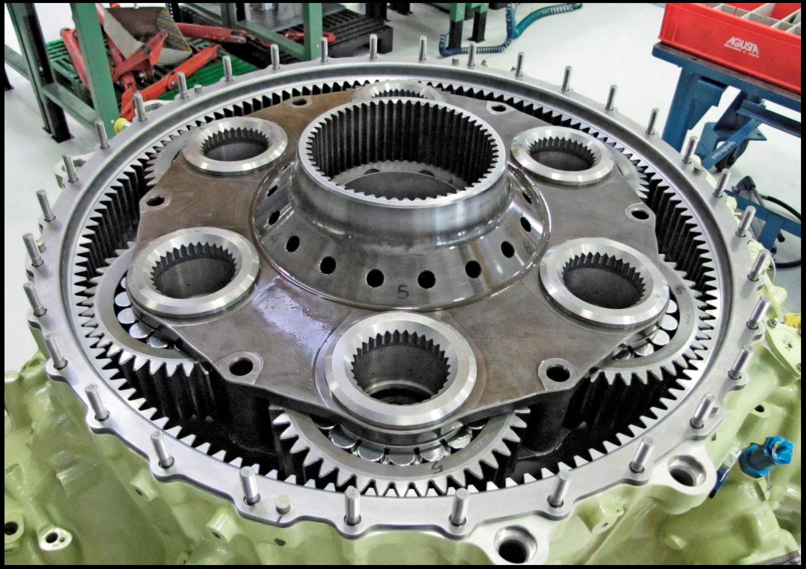
Mesh frequency for planetary



$$\omega_{mesh} = \omega_{sun} \frac{z_{sun} z_{ring}}{\left(z_{sun} + z_{ring}\right)}$$

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NVH ISSUES

Elements that affect vibration at the mesh frequency

- Variation of mesh stiffness
  - Transmission error
- Shocks at the beginning of contact
- Dynamic mesh forces
- Effects of frictional forces
- Lubricant entrapment

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NVH ISSUES

Transmission error

The literature indicates the **TRASMISSION ERROR TE** as the single **most important factor** in the generation of noise in gears

Difference between the actual position of the driven wheel and the position it would occupy if the two wheels were perfectly conjugated (perfect and rigid wheels)

Angular transmission error $TE = \theta_2 - \frac{z_2}{z_1}\theta_1$ rad

Transmission error on the action line $TE = R_{b1}\left(\theta_2 - \frac{z_2}{z_1}\theta_1\right)$ mm o in.

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NVH ISSUES

Transmission error

It depends on several factors:

If the teeth are unloaded **TE** depends on the

- inaccuracies:
- Profile errors
- Pitch errors
- eccentricity

If the teeth are loaded **TE** still depends on the inaccuracies and errors as before, but not only:

-
- It depends a lot on the stiffness of the meshing
- the elasticity-dependent part of the TE can be mitigated with appropriate reliefs

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NVH ISSUES

Transmission error under load

Under operating conditions the two effects contributing to TE are:

- constant component due to the average elasticity of the tooth
  - affects the definition of the reliefs
  - not very significant as direct impact on noise

- Variable component that depends on geometry and mesh stiffness
  - Significant impact on noise produced

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Transmission error under LTE load

- The mesh stiffness is proportional to the average number of teeth in contact (contact ratio)
- The contact ratio therefore plays an important role
- It must be taken into account that under load the contact ratio changes (is reduced) compared to the geometric / kinematic one
- Theoretically, the contat ratio would seem to have to be an integer number (if we assume a piecewise constant stiffness)
- Wheels with exactly integer contact ratio are rarely designed

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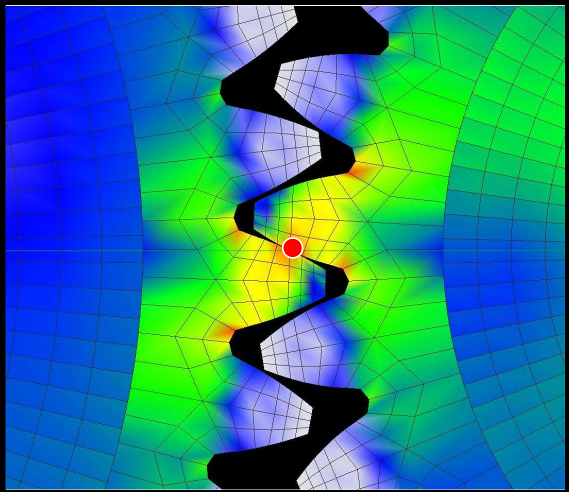
NVH Issues

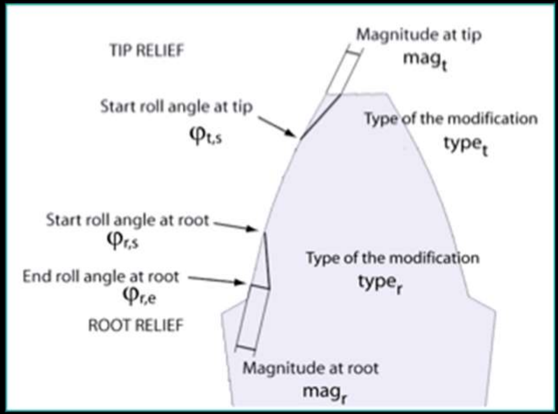
LTE: ELASTICITY EFFECT

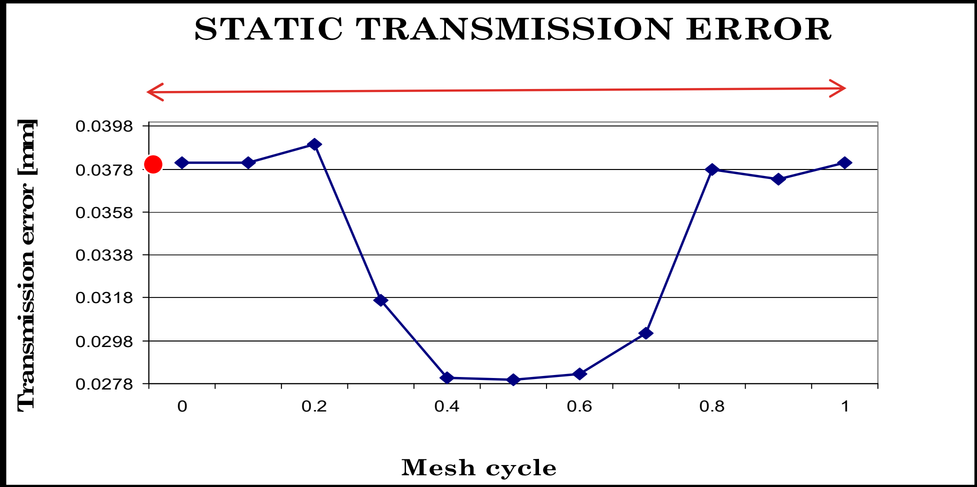
- Errors and profile modifications
- Elastic deformations
- $$1 < \varepsilon_\alpha < 2$$



The static transmission error is periodic in the gear cycle unless there are constructive errors

WE TRY TO REABSORB IT WITH THE RELIEFS





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BEFORE PROCEEDING IT IS USEFUL TO RECALL CONCEPTS OF WHEEL GEOMETRY

MICRO-GEOMETRIC MODIFICATIONS

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MICRO GEOMETRIC MODIFICATIONS

PROFILES MODIFICATIONS

There are many unique technical words related to gearing. Also, there are various unique ways of modifying gears. This section introduces some of most common methods. (KHK Reference Handbook)

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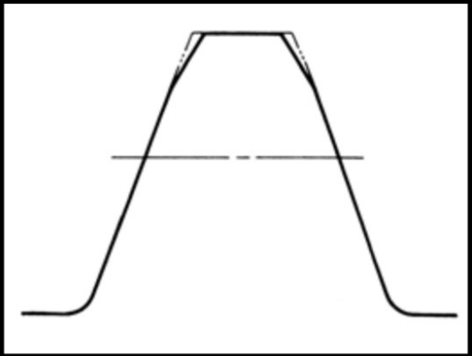
PROFILE MODIFICATIONS

Tooth Profile Modification

- Tooth Profile Modification generally means adjusting the addendum.
- Tooth profile adjustment is done by chamfering the tooth surface in order to make the incorrect involute profile on purpose.
- This adjustment, enables the tooth to vault when it gets the load, so it can avoid interfering with the mating gear.
- This is effective for reducing noise and longer surface life.
- However, too much adjustment may create bad tooth contact as it is functions the same as a large tooth profile error.



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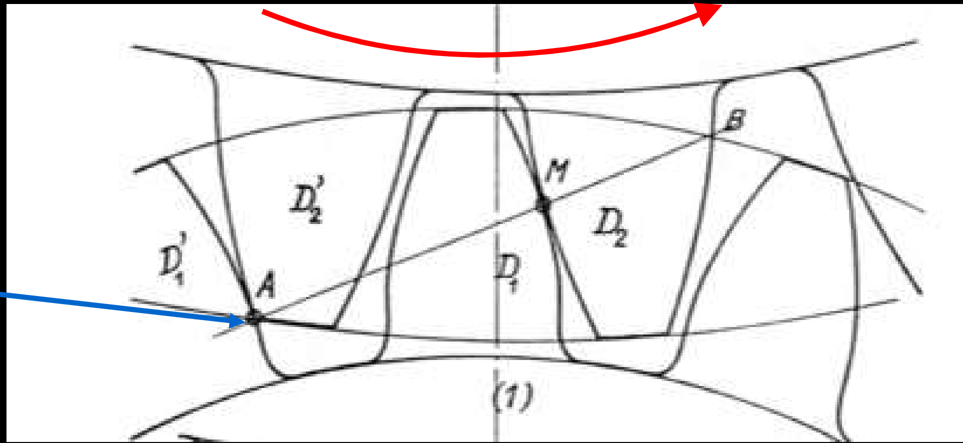
MESHING ISSUES

Error-free profile



Start regular contact

This is true if:

- There are no constructive errors
  - errors in pitch, profile, assembly, alignment
- Elastic deformations are sufficiently small
  - Low loads compared to sustainable maxima

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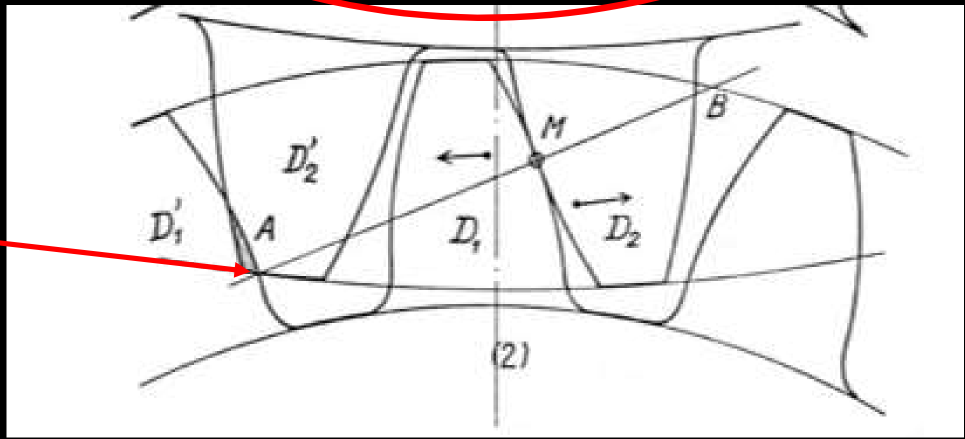
MESHING ISSUES

Teeth with error or deformation

Approach: irregular contact



- At the approach D_1' and D_2' the two profiles are not conjugated for some reason
  - Pitch error, profile error, deformation
- Impact takes place at the contact start
- The tooth end of wheel 2 tends to penetrate the tooth side of wheel 1
- Pressure peaks are found locally
- Problems of poor lubrication, wear, early pitting may arise locally
- Considerable noise is produced

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MESHING ISSUES: MITIGATION TECHNIQUES

TIP AND ROOT RILIEF

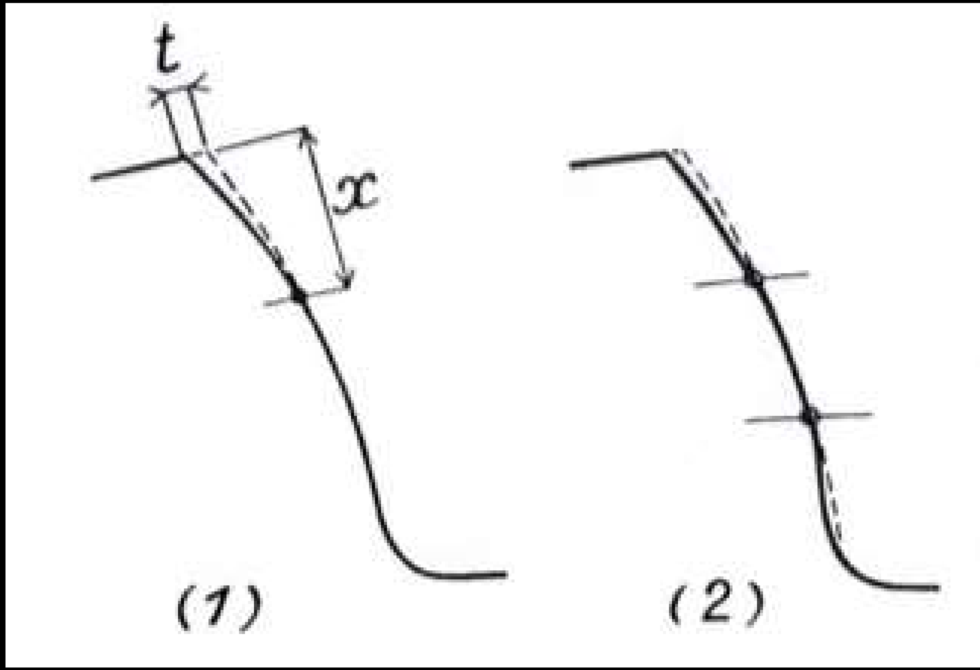
- Henriot suggests a slight tip relief on the driven wheel

or alternatively

- a slight root relief on the driver wheel to avoid incoming impacts

In addition

- He also suggests a slight tip relief on the driver wheel to reduce the pressures during the exit



Tip relief

Tip and root relief

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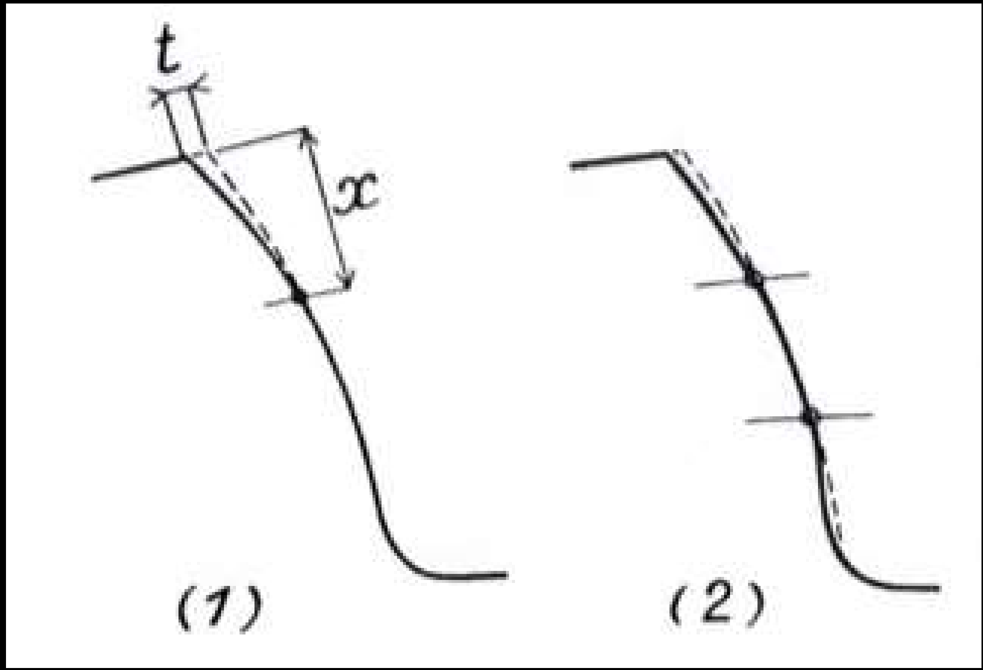
MESHING ISSUES: MITIGATION TECHNIQUES

TIP AND ROOT RILIEF

At high load and high speed it is good to have reliefs on both wheels, with priority on the driven

Henriot gives a qualitative indication of the relief magnitude

$$1\ \mu\text{m for } 20\,\frac{\text{N}}{\text{mm}} \text{ unitary load}$$



Tip relief

Tip and root relief

Suggestion for the relief *t*

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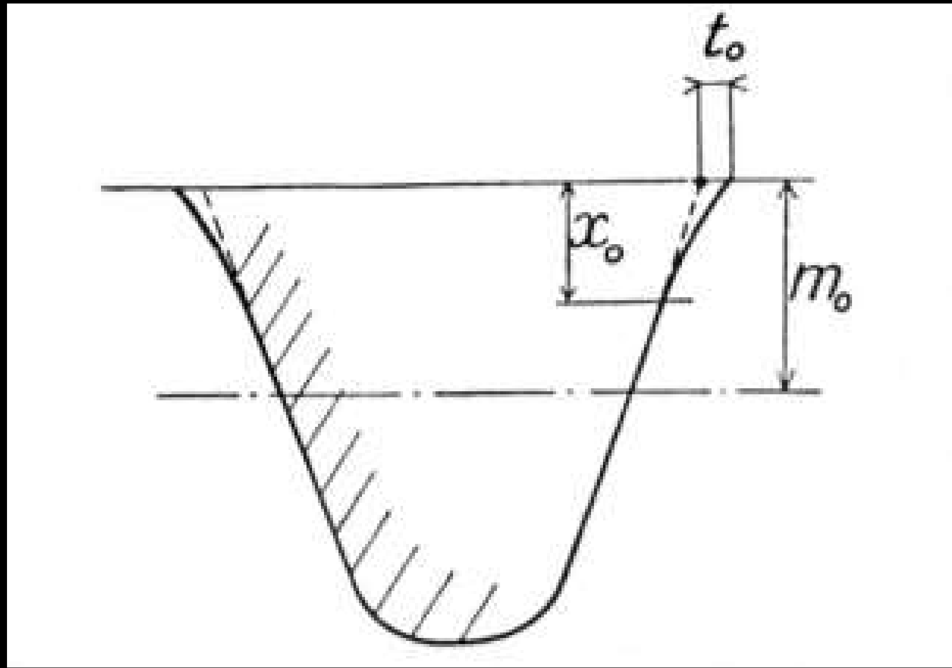
PROFILE MODIFICATIONS

TIP AND ROOT RILIEF

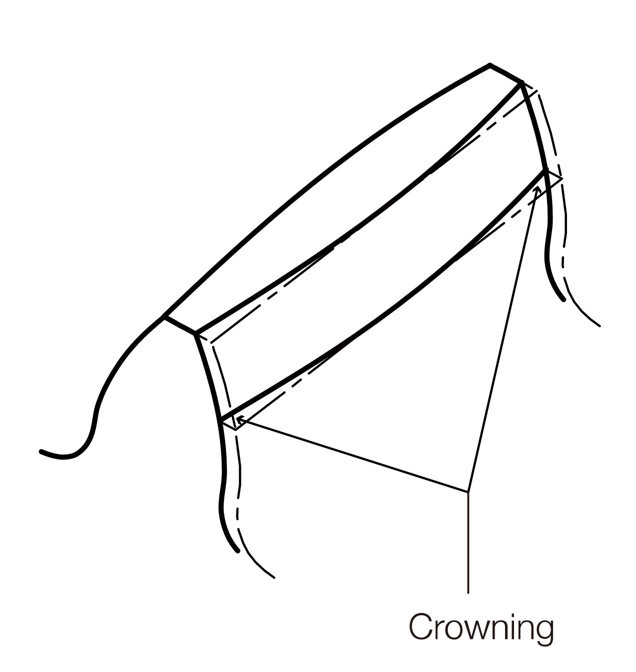
Profile modifications are usually obtained by griding
However, they can also be obtained with modified rack



TOOTH MODIFICATIONS

Crowning and End Relief

- Crowning is the removal of a slight amount of the tooth from the center on out to the reach edge, making the tooth surface slightly convex.
- This method allows the gear to maintain contact in the central region of the tooth and permits avoidance of edge contact.
- Crowning should not be larger than necessary as it will reduce the tooth contact area, thus weakening the gears strength.
- End relief is the chamfering of both ends of tooth surface.



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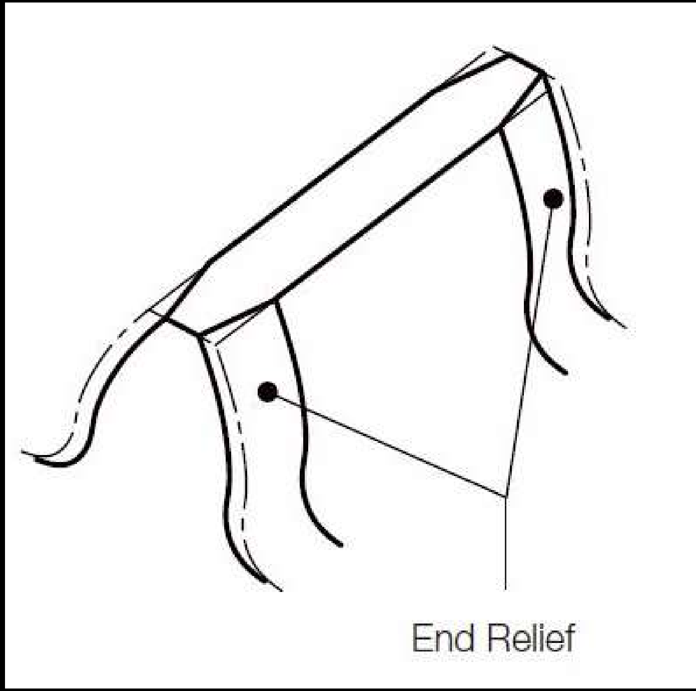

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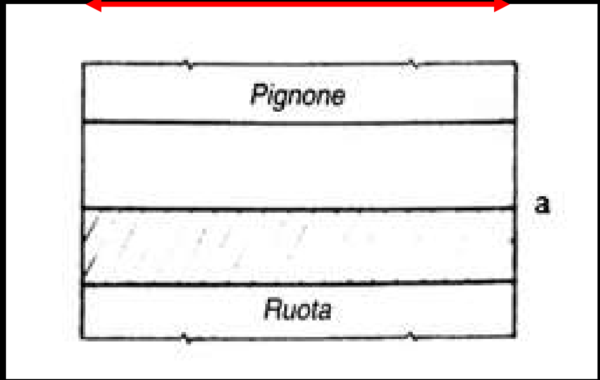
CONTACT CONDITIONS ALONG THE WIDTH

facewidth

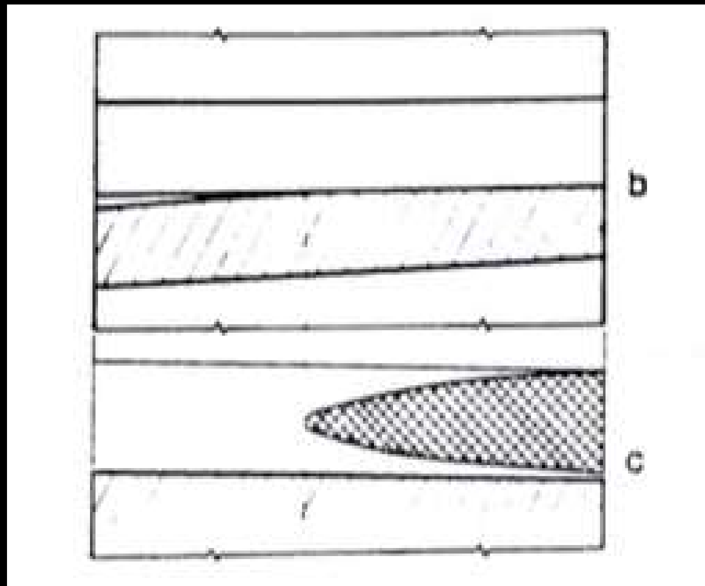


Regular meshing
Perfectly parallel wheels

Wheels with misalignment or shaft failure
Remedies

- Symmetrical crowning for weak distortions
- helix correction for large distortions

Crowning increases Hertzian pressure



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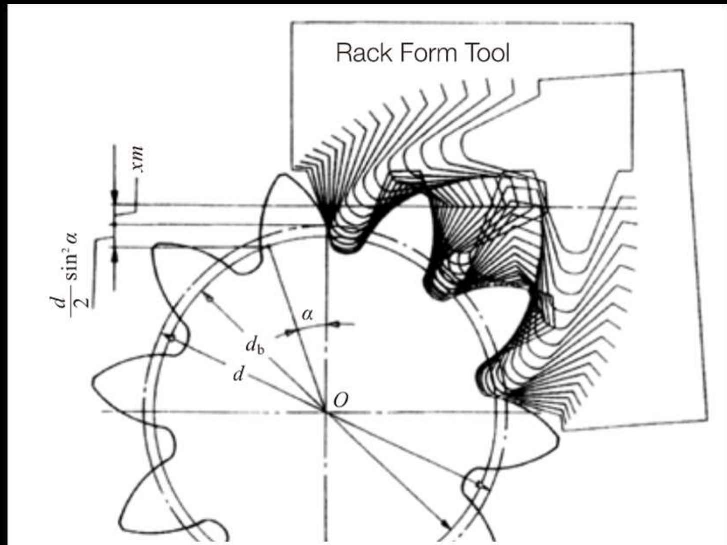
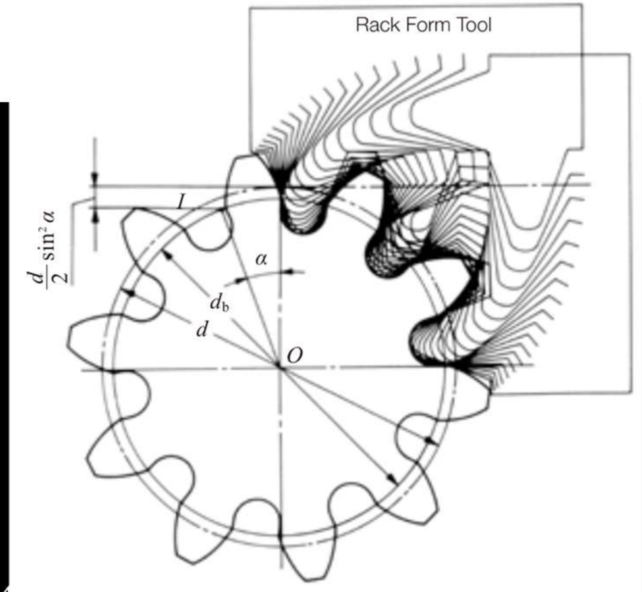
TOOTH MODIFICATIONS

Topping and Semitopping

In topping, often referred to as top hobbing, **the top or tip diameter** of the gear **is cut simultaneously with the generation of the teeth**.
The figures indicate topping and generating of the gear by rack type cutters.
An advantage is that there will be no burrs on the tooth top.
Also, the tip diameter is highly concentric with the pitch circle.





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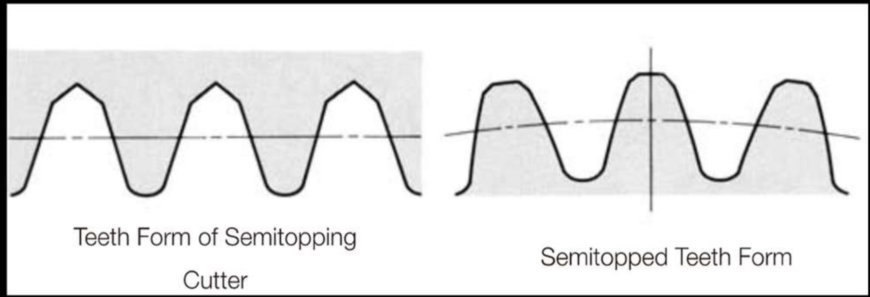
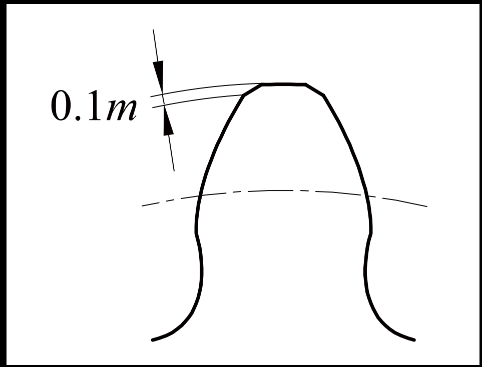
PROFILE MODIFICATIONS

Topping and Semitopping

Semitopping is the chamfering of the tooth's top corner, which is accomplished simultaneously with tooth generation.
The Figs. show a semitopping cutter and the resultant generated semitopped gear.
Such a tooth end prevents corner damage and has no burrs (bave).



Semitopping Cutter and the Gear Profile



Magnitude of Semitopping

NVH ISSUES

LOADED TOOTH CONTACT ANALYSIS
LTCA

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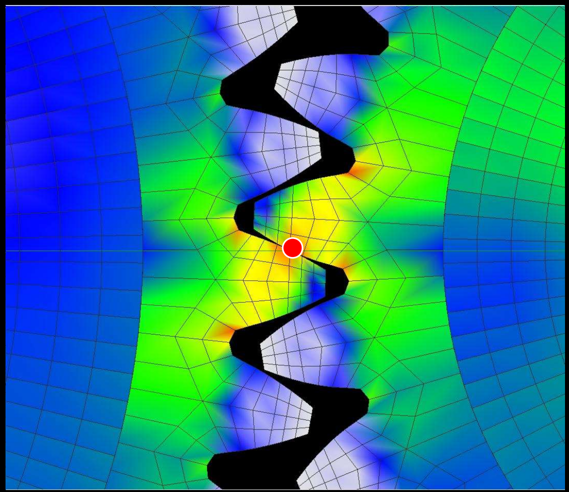
NVH Issues

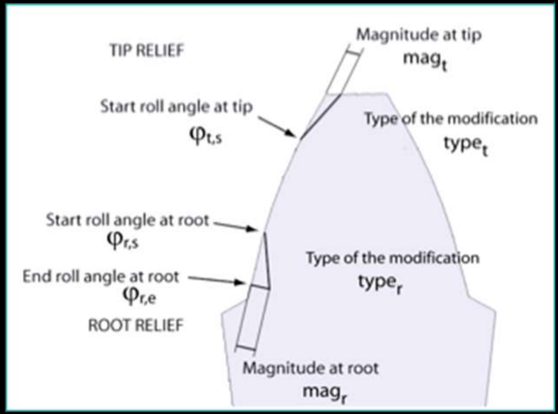
LTE: ELASTICITY EFFECT

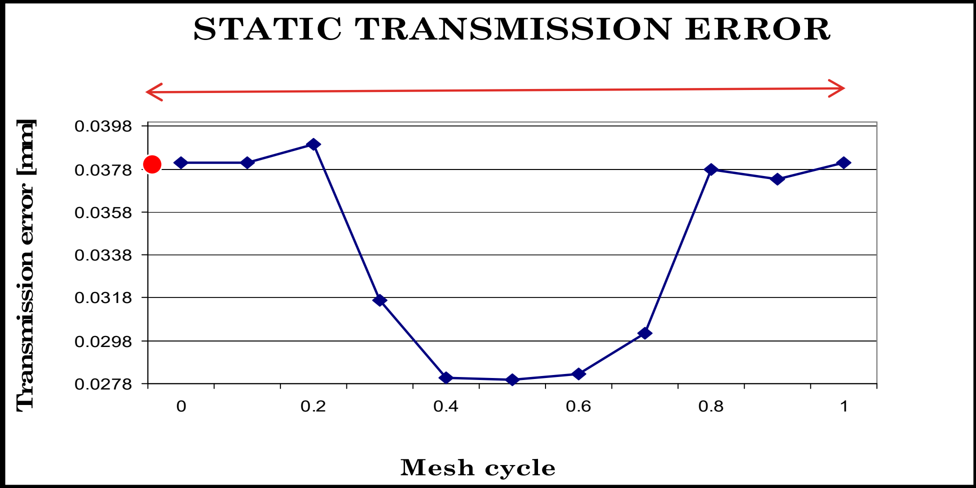
- Errors and profile modifications
- Elastic deformations
- $1 < \varepsilon_\alpha < 2$



The static transmission error is periodic in the gear cycle unless there are constructive errors

WE TRY TO REABSORB IT WITH THE RELIEFS





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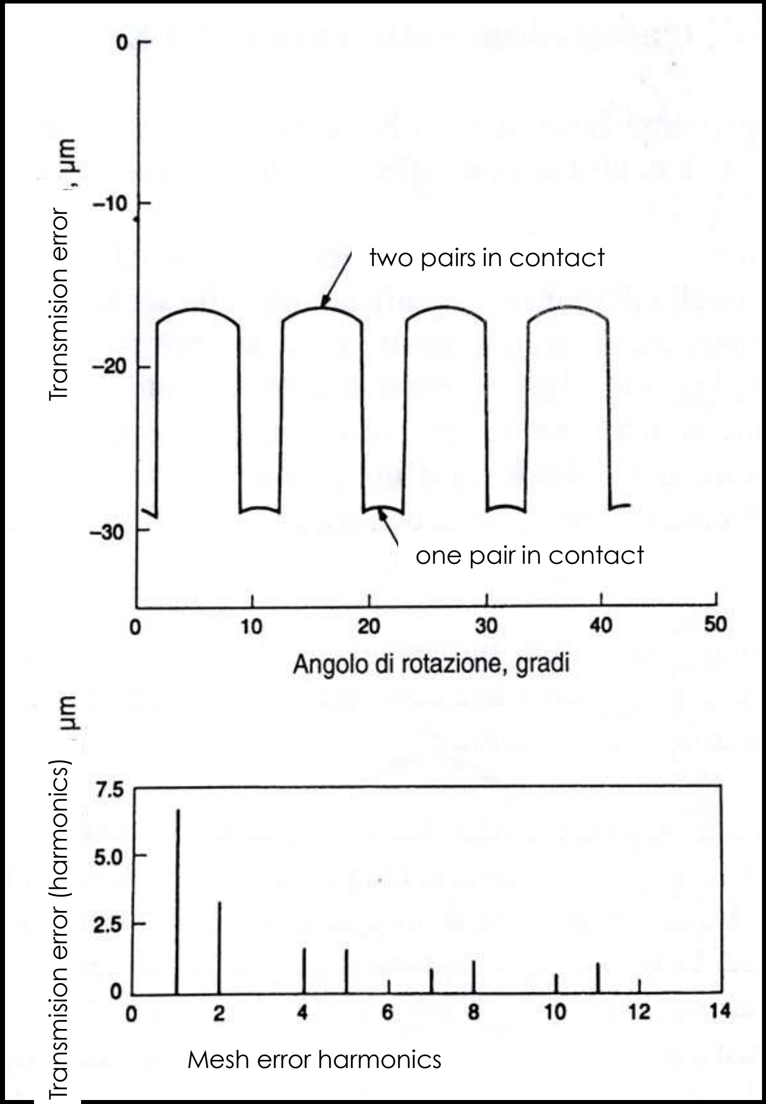
Problematiche NVH

LTE Transmission Error Under Load

- We often assume piecewise constant stiffness (or almost)
- The transition between a pair of teeth in contact and two pairs in contact is modelled with a jump
- The harmonics of the transmission error are evaluated
  - The first harmonic usually dominates
  - With the reliefs it is possible to lower the first two-four harmonics



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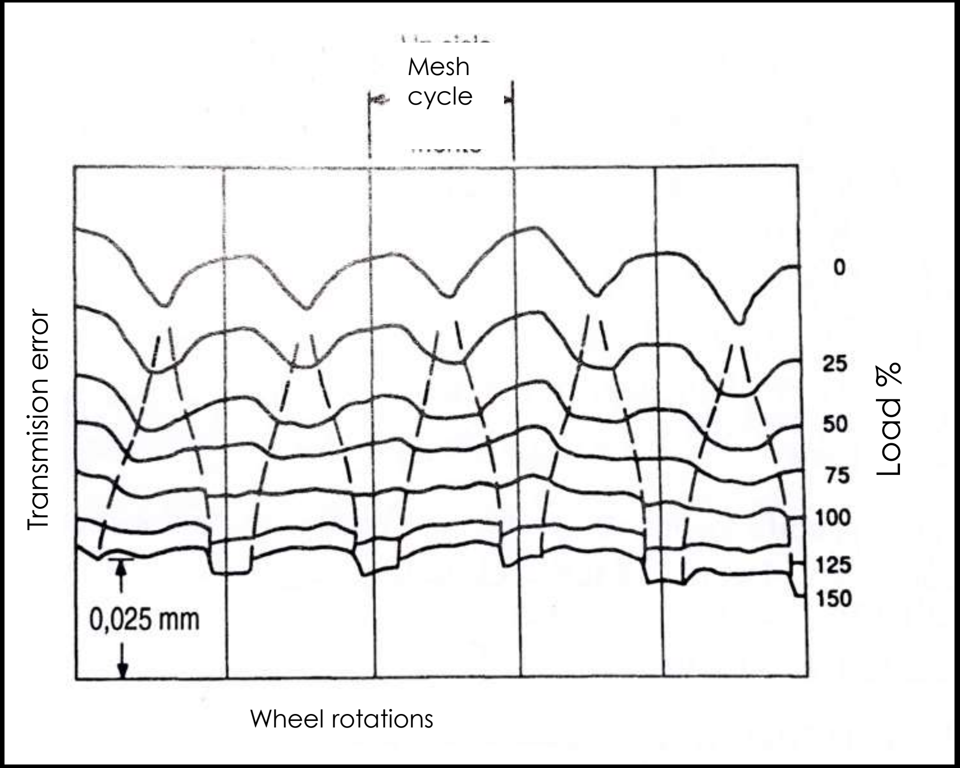
LTE Transmission Error Under Load

- The reliefs modify the meshing
- Can be intrusive enough to change kinematics
- If they change the kinematics worsen the response to very low torque
- If they are well made at nominal load they reduce the TE on some aspect:
    - Peak peak
    - Single harmonics or average



LTE Transmission Error Under Load

Without reliefs

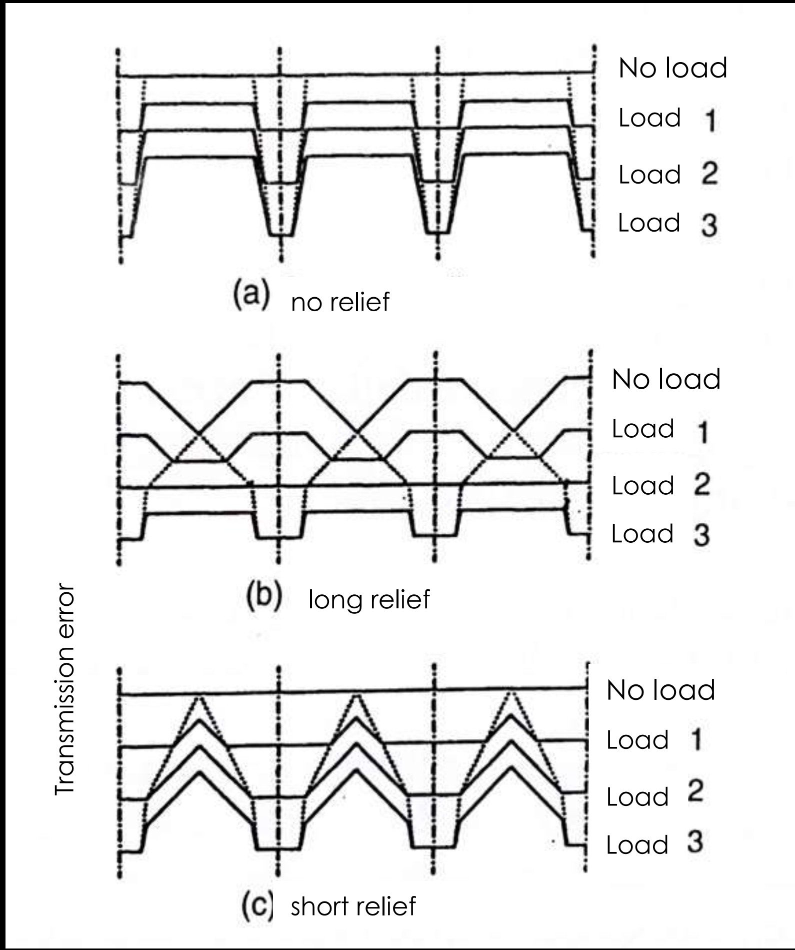
- Without load: perfect kinematic
- With load: LTE increasing

Long relief

- No load (or low loads): kinematics not perfect, non-homokinetic transmission, possible noise problems
- With load: very low LTE compared to pure involute wheels

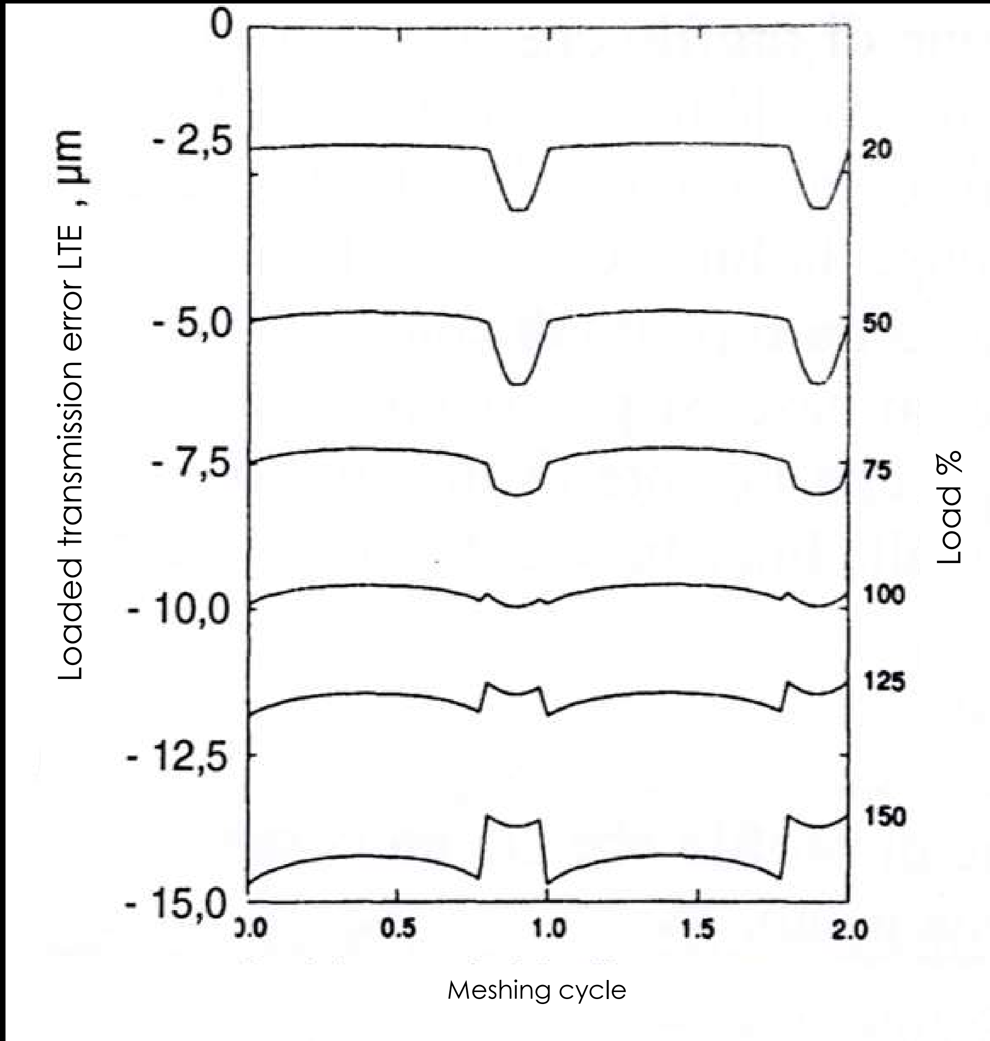
Short relief

- Without load: perfect kinematic
- With load: LTE increasing
- Much less effective than long modifications at nominal load



LTE Load Sensitivity

- Spur gears
- High Conduct Ratio (2.2)
- In the figure simulated results (Houser non-FEM approach)
- The improvements are clear in all conditions compared to a gear with a lower gear ratio (e.g. 1.6-1.8)
- However, the constructive tolerances must be very tight



HARRIS MAPS

Effect of reliefs on transmission error

Unloaded wheels

Pure involute reference line

Deviation (TE) due to material removal (relief)

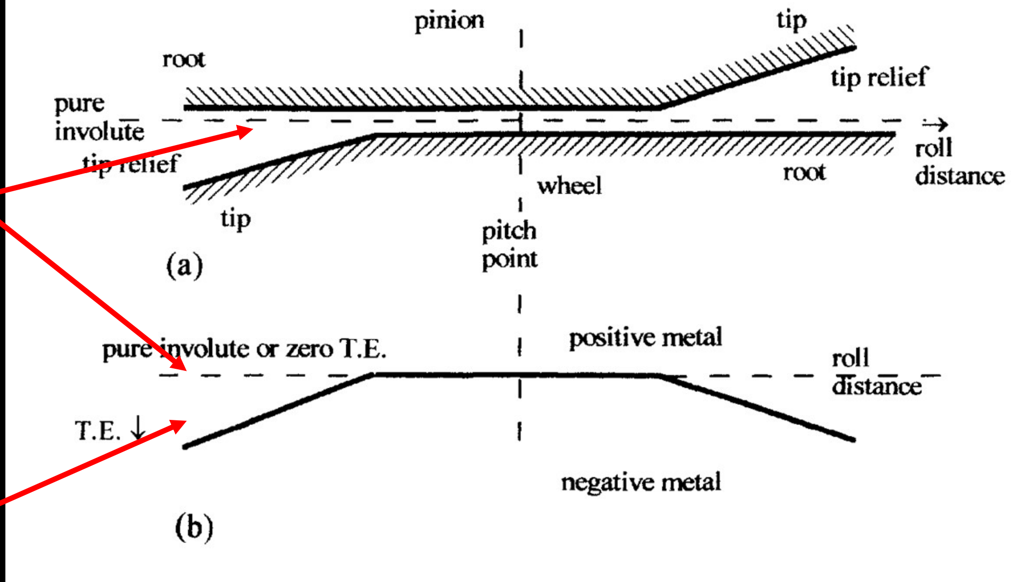


Fig 2.6 Effects of mating two spur gear profiles, each with tip relief.

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Effect of reliefs on transmission error

Unloaded wheels

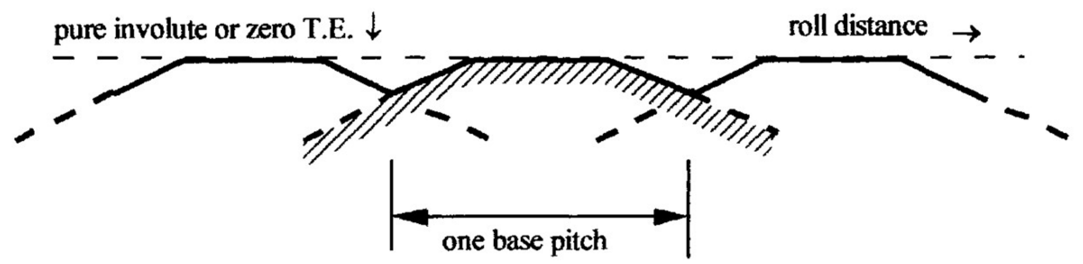

Fig 2.7(a) Effect on T.E. of handover to successive teeth when there are no elastic deflections.

Effect of meshing of successive teeth

It is also called "Intentional transmission error"

Note: In the experimental measurement, the dotted line is lost

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Effect of reliefs on transmission error

Unloaded wheels

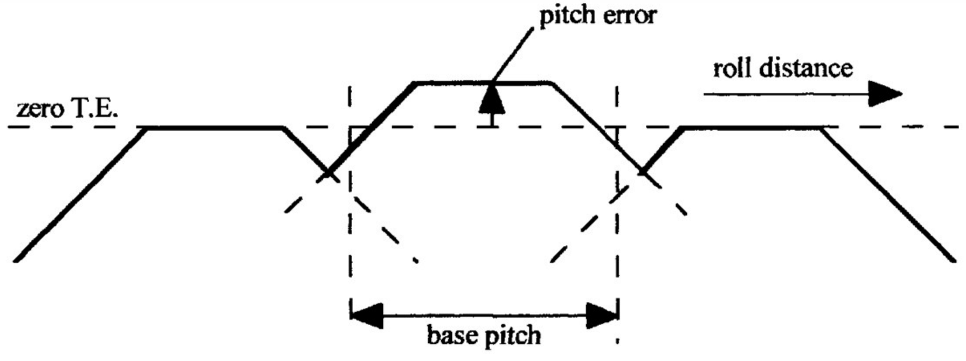

Fig 2.7(b) Effect of pitch error on position of handover and T.E.

Effect of meshing of successive teeth
Combination with pitch error

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HARRIS MAPS

Effect of reliefs on transmission error

Loaded wheels

One tries to predict the TE under load

It assumes constant stiffness!
But the distribution of the load (two or more teeth) is taken into account.

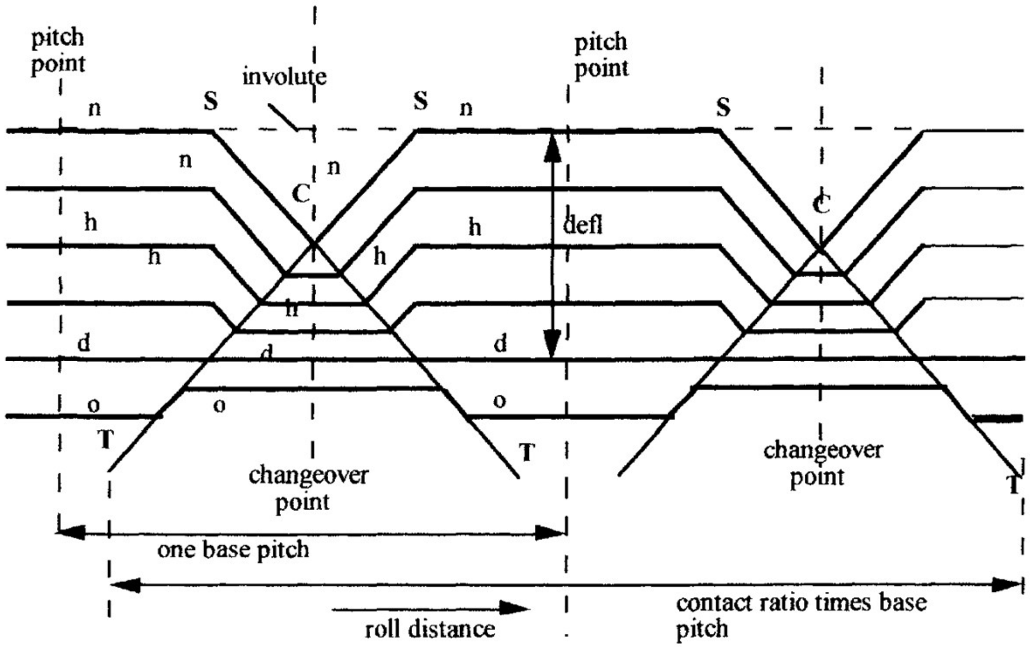

Fig 2.8 Harris map of interaction of elastic deflections and long tip relief.

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HARRIS MAPS

Effect of reliefs on transmission error

Loaded wheels

No load

Half load

Design load
TE cancelled

Overload
TE positive
when load is
shared

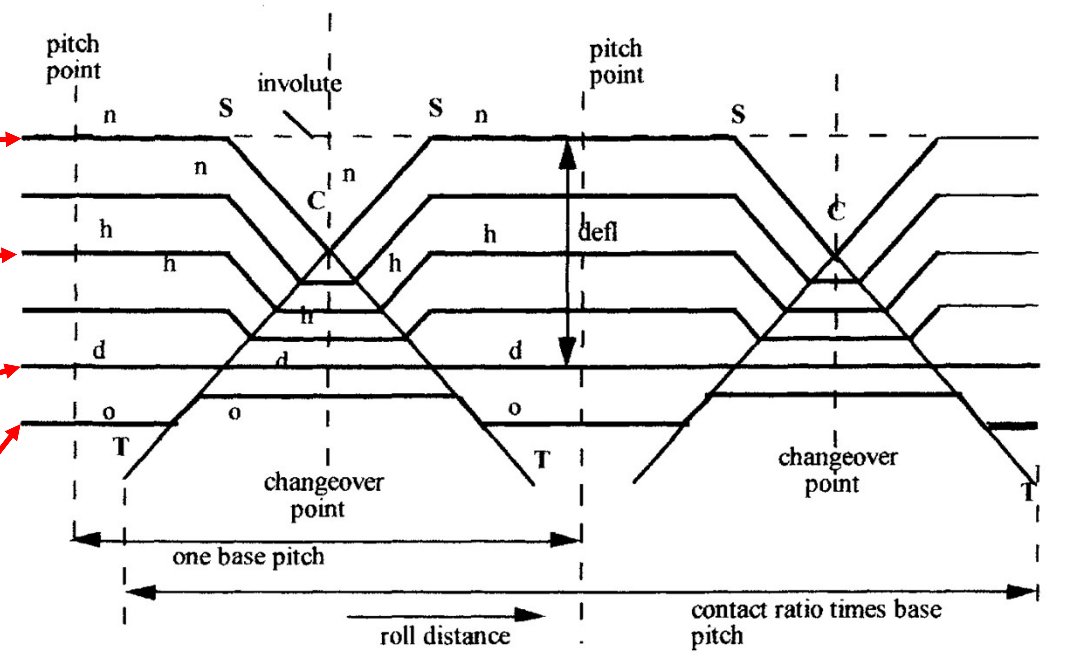

Fig 2.8 Harris map of interaction of elastic deflections and long tip relief.

long relief

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MODERN APPROACH TO RELIEF OPTIMIZATION

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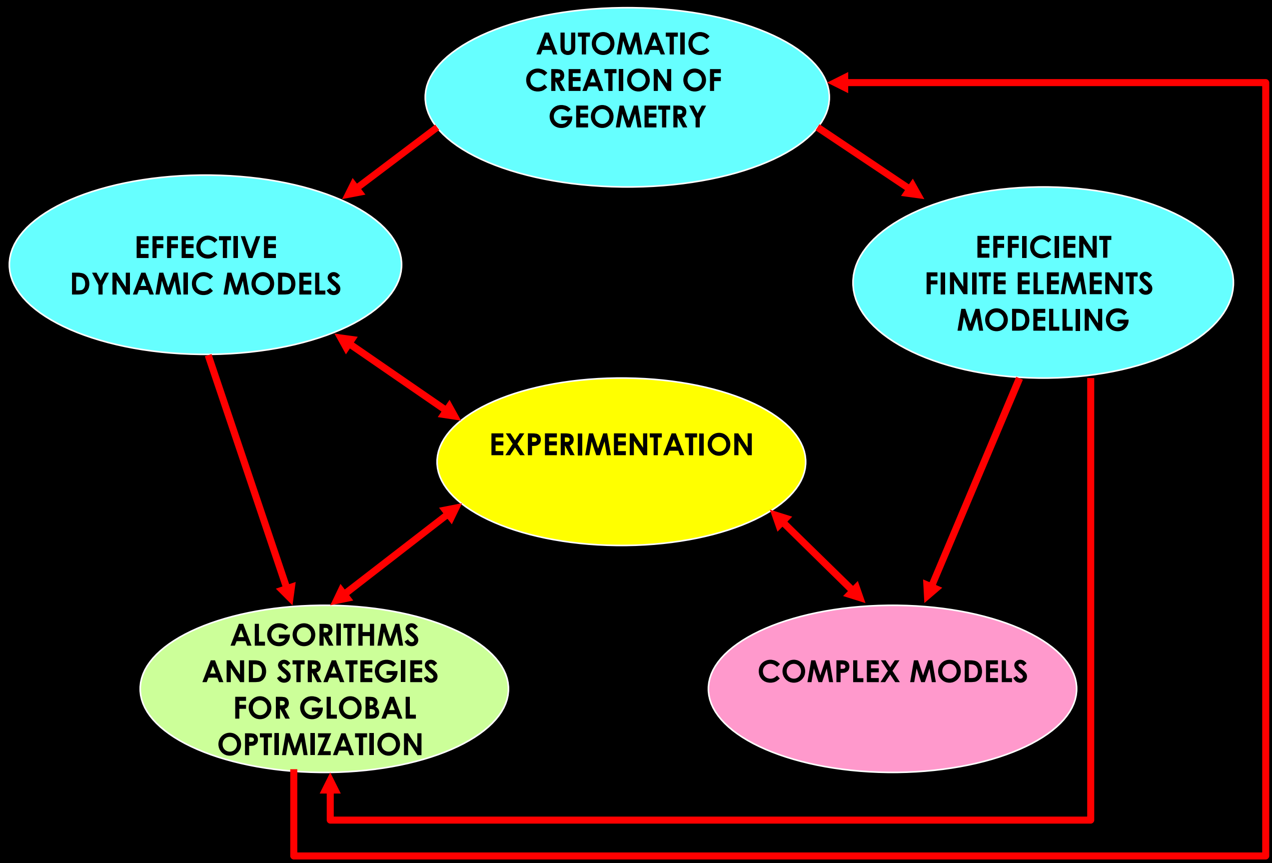
NVH ISSUES



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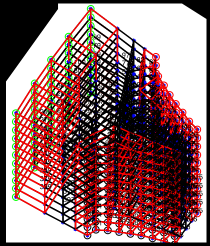
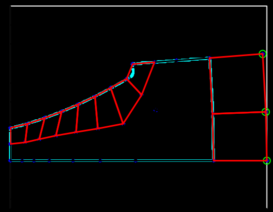
Automatic Geometry Creation

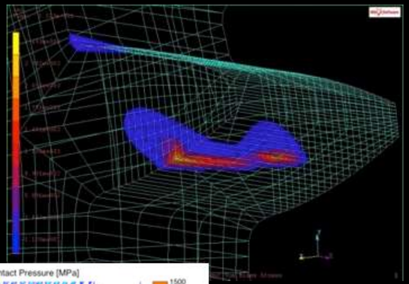
- 3D geometries (envelope by points) NURBS methodology
- Arbitrary corrections (reliefs, crowning) misalignments

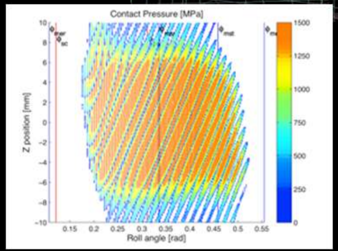




Automatic creation of FEM models

- 2-3D ready-2-run models
- LTCA Load Contact Analysis
- Profile correction and misalignment effects
- Fatigue analysis





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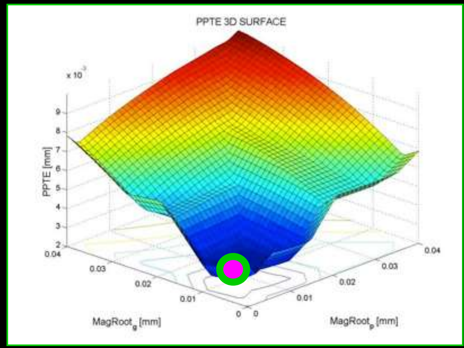
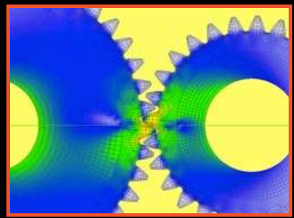


Gear optimization

- Static and dynamic
- Genetic algorithms
- Random search+linear prog

Gear trains



Dynamic models

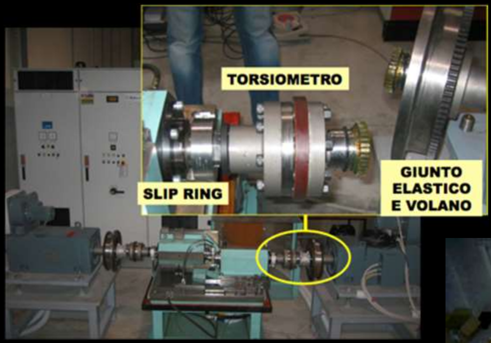
- Natural frequencies and modes of vibration
- Nonlinear response

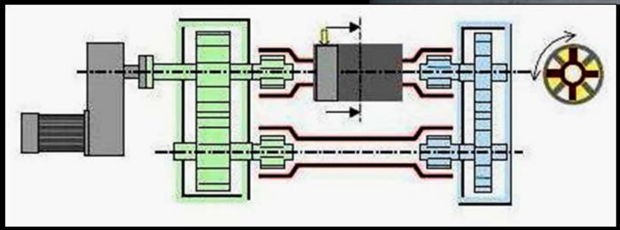
Planetary

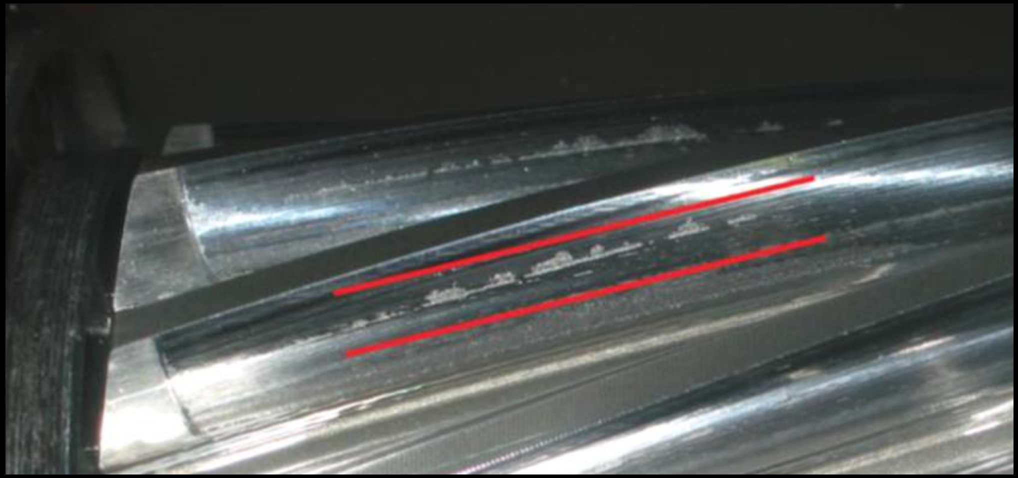
PROBLEMATICHE NVH

Testing

- Test bench design
- Dynamic tests
- Endurance test (pitting fatigue)







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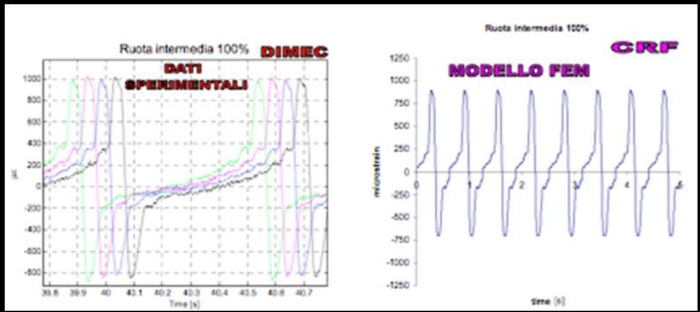
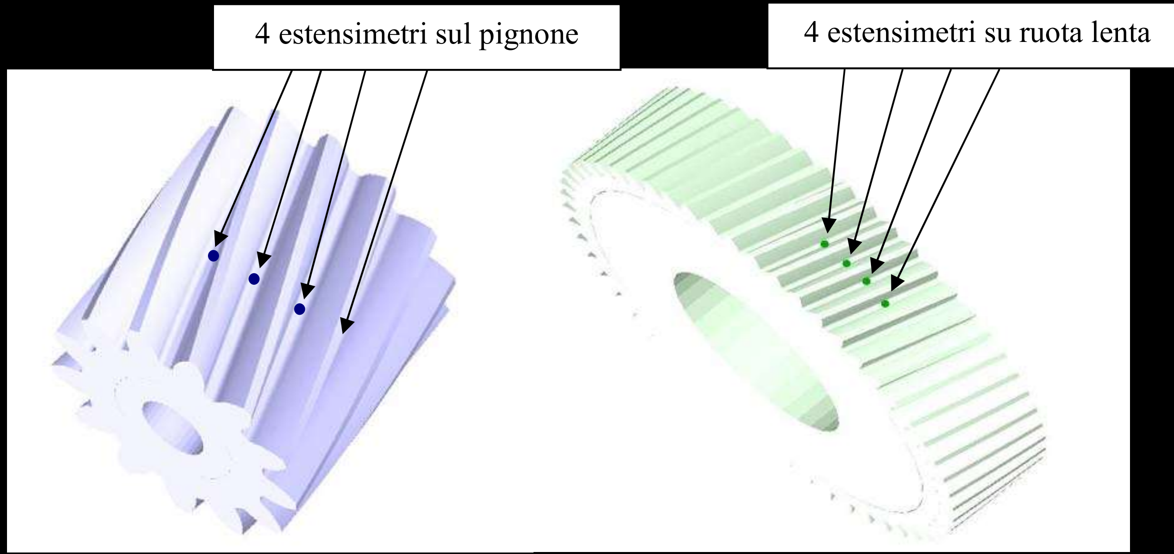
Testing

PROBLEMATICHE NVH

- **Tooth-based deformation measurement under operating conditions**
- **Comparison with FEM models**





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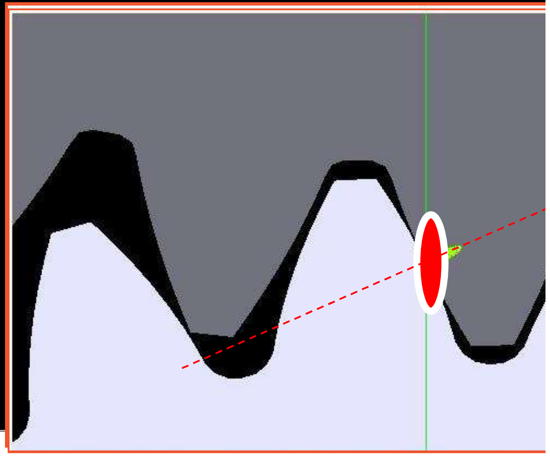
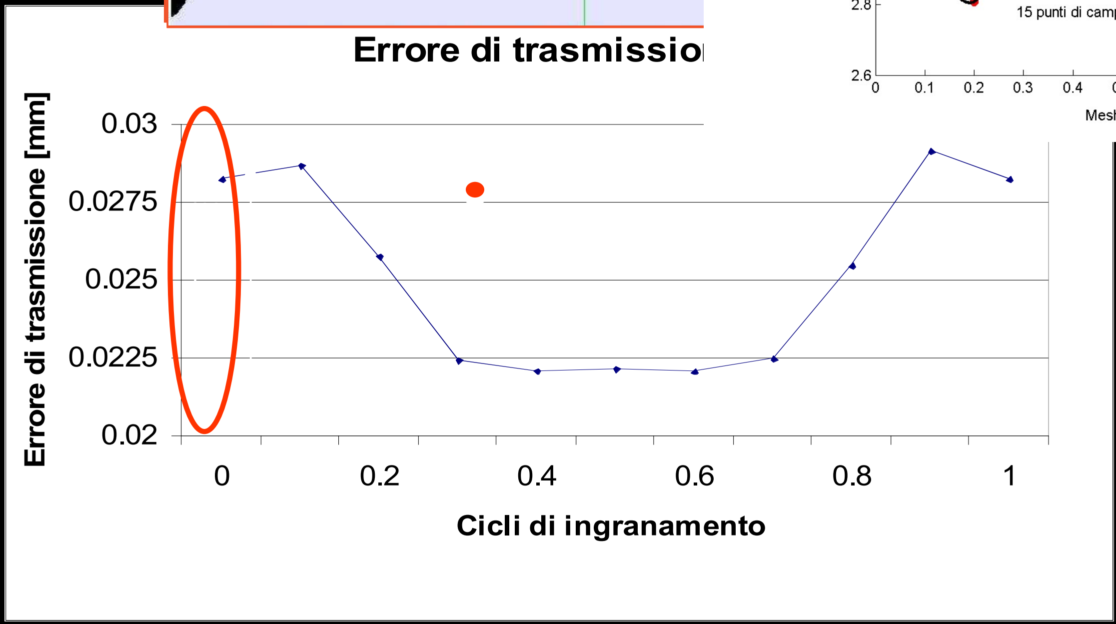
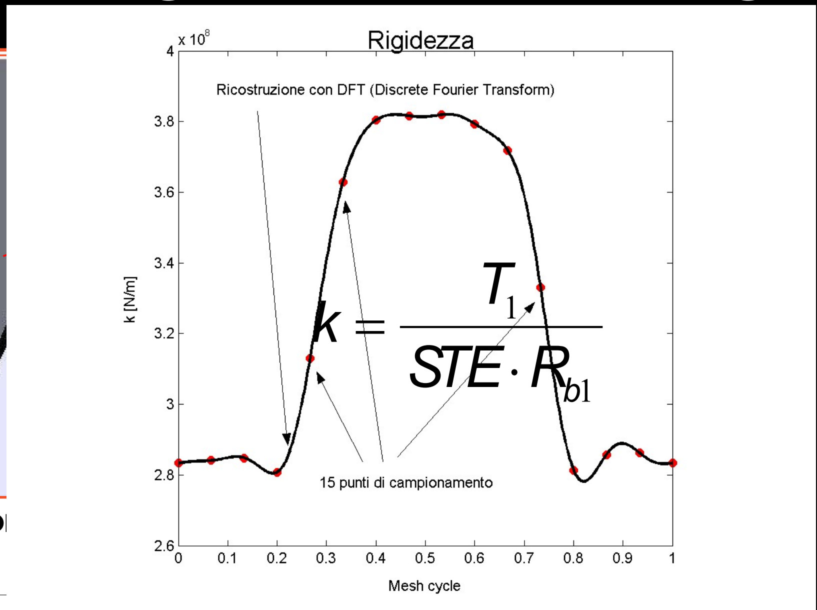
TRANSMISSION ERROR AND VIBRATION





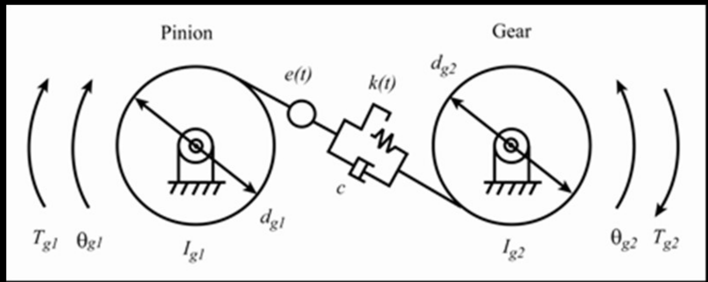


DTE AND DYNAMIC MODEL

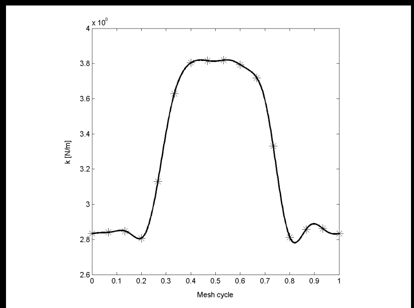
STE
At low speed

→ **Dynamic model** →

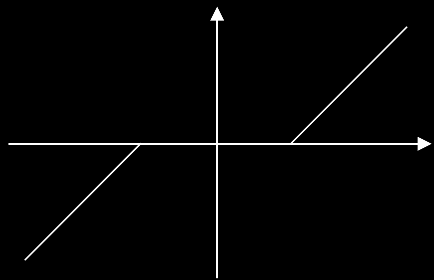
DTE
Depends on the speed (inertia)



TIME VARIANT

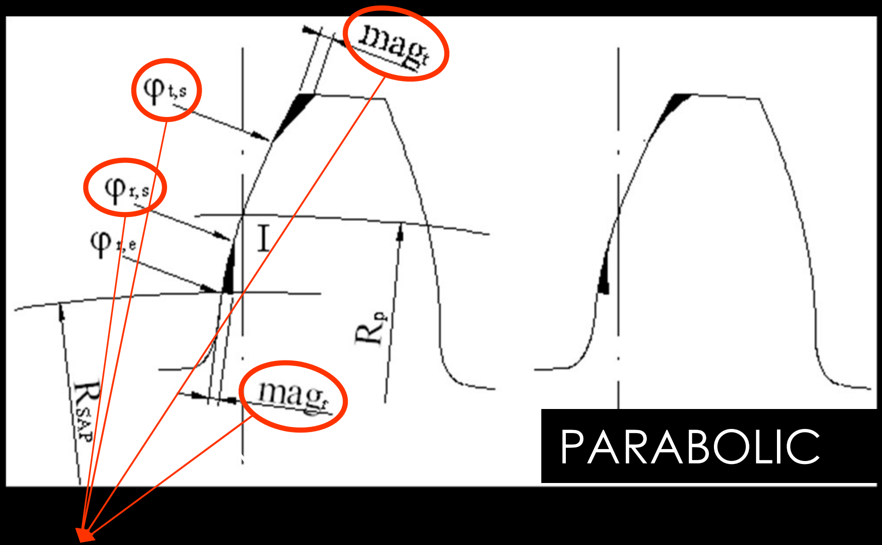


NON LINEAR



DTE AND DYNAMIC MODEL

Profile modifications



$$m_e\ddot{x}(t) + c(\dot{x}(t) - \dot{e}(t)) + k(t)\,f\,(x(t) - e(t)) = T_g(t)$$

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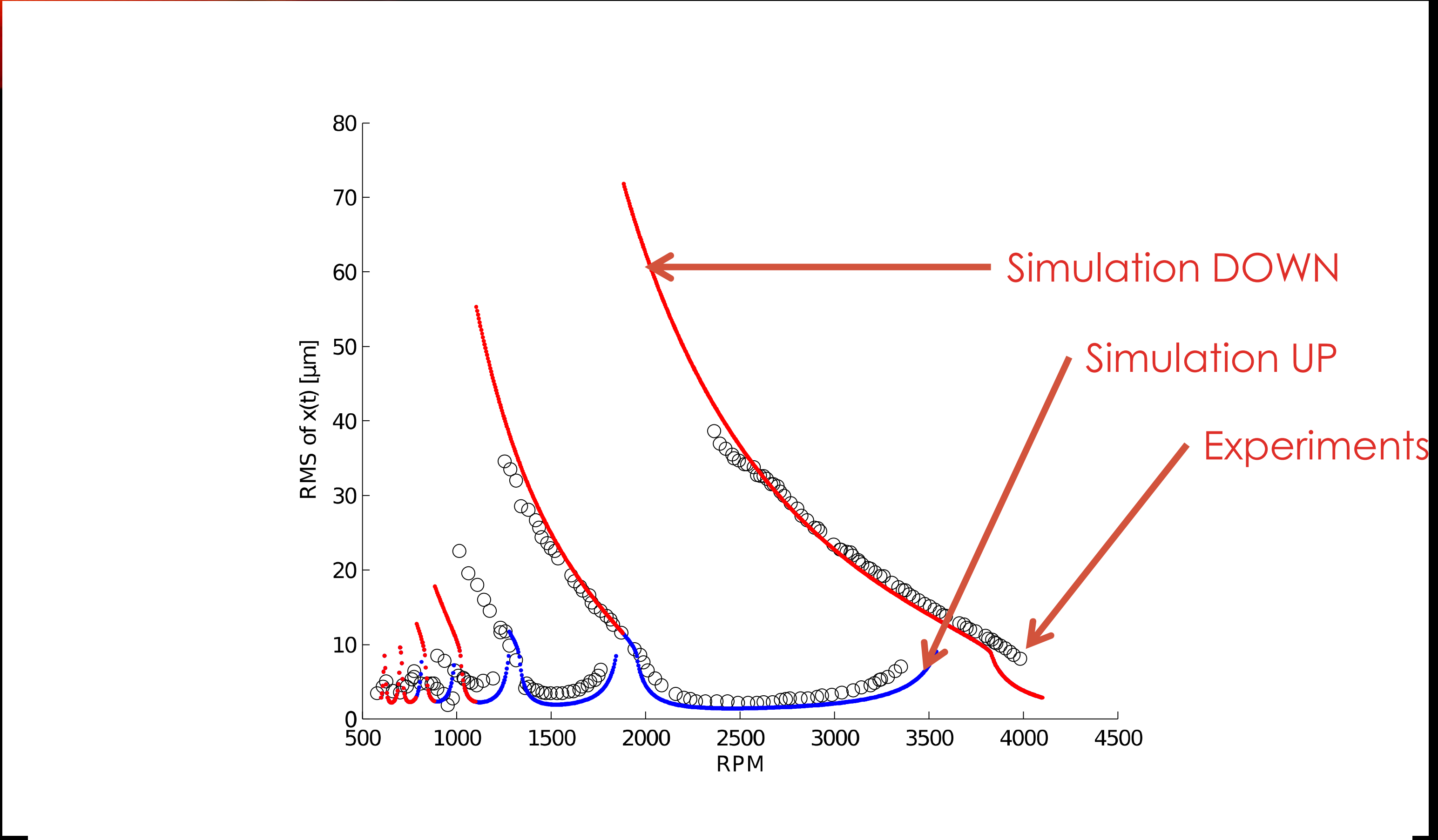
MODEL VALIDATION



$$m_e \ddot{x}(t) + c(\dot{x}(t) - \dot{e}(t)) + k(t)\, f\,(x(t) - e(t)) = T_g(t)$$

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OPTIMIZATION

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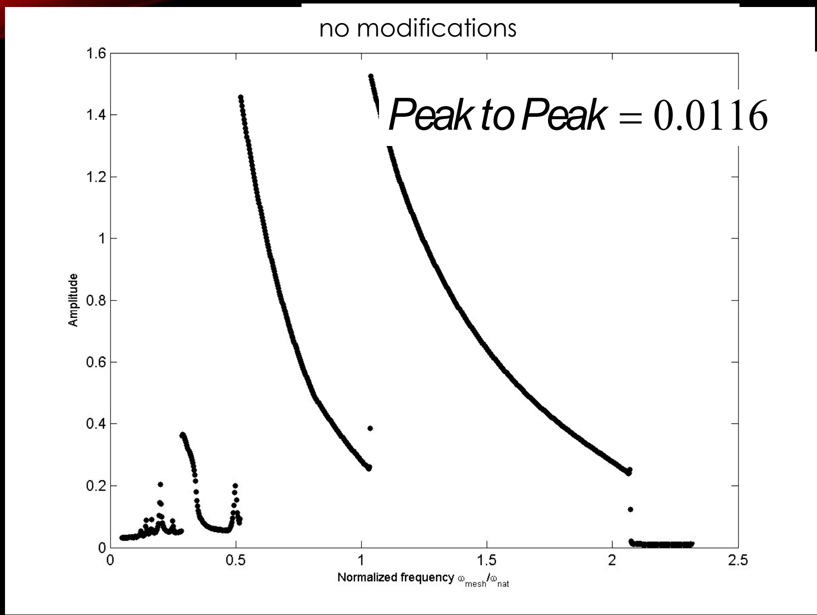
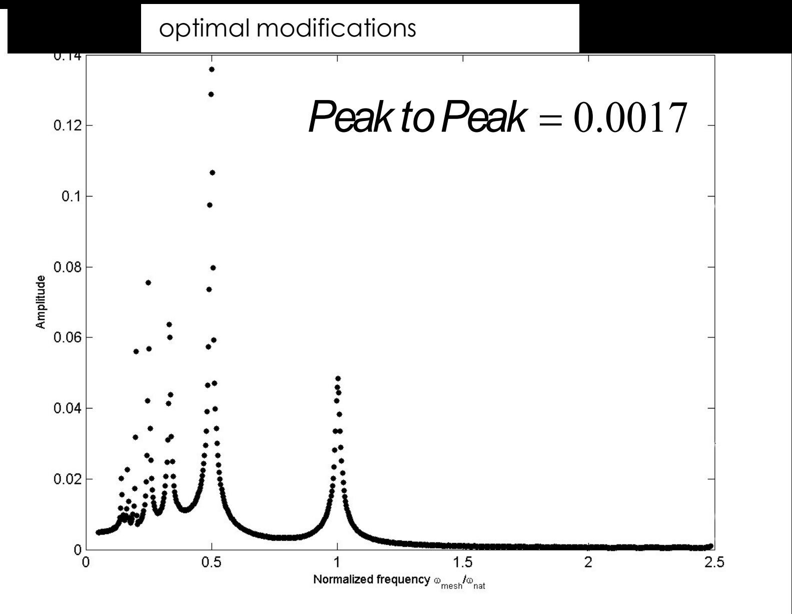
DTE AND DYNAMIC MODEL





EFFECT OF STATIC TRANSMISSION ERROR REDUCTION

- Maximum width reduced by ten times
- Fully linear behavior (without contact loosing)
- No parametric resonance
- Superharmonics more important than resonance at the natural frequency

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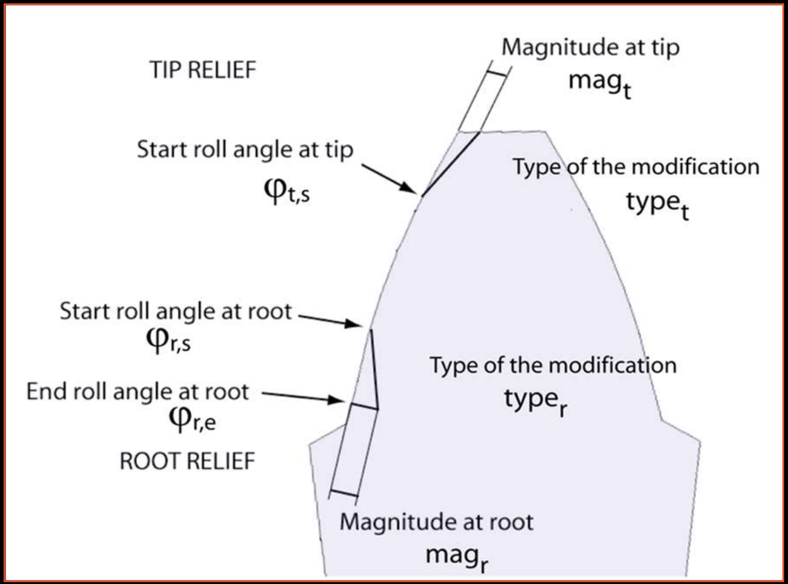
DEVELOPMENT OF OPTIMIZATION METHODS

To reduce the vibration of a gear, we act on the static transmission error to reduce its peak to peak:

1. **Acting on the surface finish**
2. **Acting on macrogeometric parameters: conduct ratio, clearance, number of teeth**
3. **Practicing micrometric reliefs to the profile in the head and foot and on the side of the tooth**



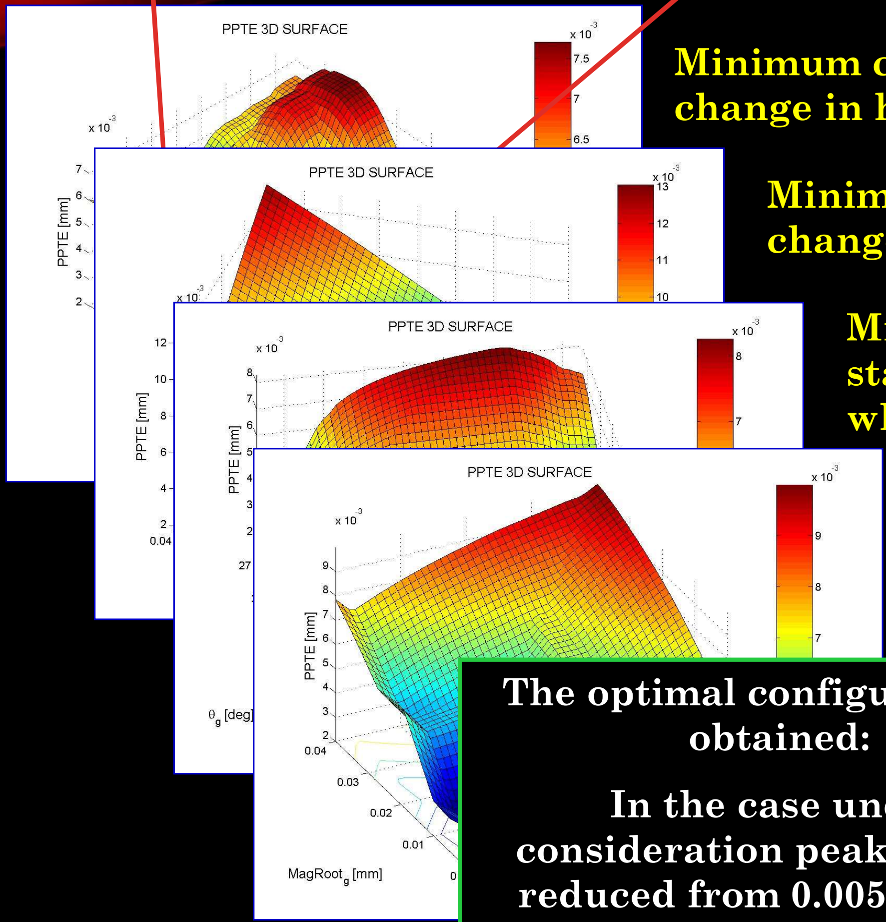
- **Modification parameters:**
- **Crowning values of the flanks**
- **Roll angle of the beginning of the reliefs in the tip and root of the profiles**
- **Extent of the reliefs**

OPTIMIZATION APPROACHES

- BRUTE FORCE: the whole parameter space is spanned
  - A global minimum is found
  - Heavy computational cost
  - Small parameter spaces can be spanned
- EURISTIC: two parameters are considered at each step
  - The computational cost is reduced
  - The optimum is not the minimum
  - The optimum depends on the initial set of parameters
- GENETIC ALGORITHMS: simulates the evolution process
  - Finds a global minimum
  - The computational cost is reduced
- STOCHASTIC+SIMPLEX:
  - The Stochastic approach generates random set of parameters. The global minimum is found after a large number of steps.
  - The simplex method is robust, but local.
  - The combination Stochastic+Simplex gives good results

ITERATIVE STATIC OPTIMIZATION

Minimum neglected: high gradients **Optimal engineering**



Minimum compared to the roll angle of start change in head on pinion and wheel

Minimal compared to the amplitude of changes in the head on pinion and wheel

Minimum compared to the roll angle of the start of modification to the foot on pinion and wheel

Minimal compared to the amplitude of the foot modifications on pinion and wheel

The optimal configuration is obtained:

In the case under consideration peak to peak reduced from 0.0051 mm to 0.00198 mm (-61%)

...ng configuration

...l configuration

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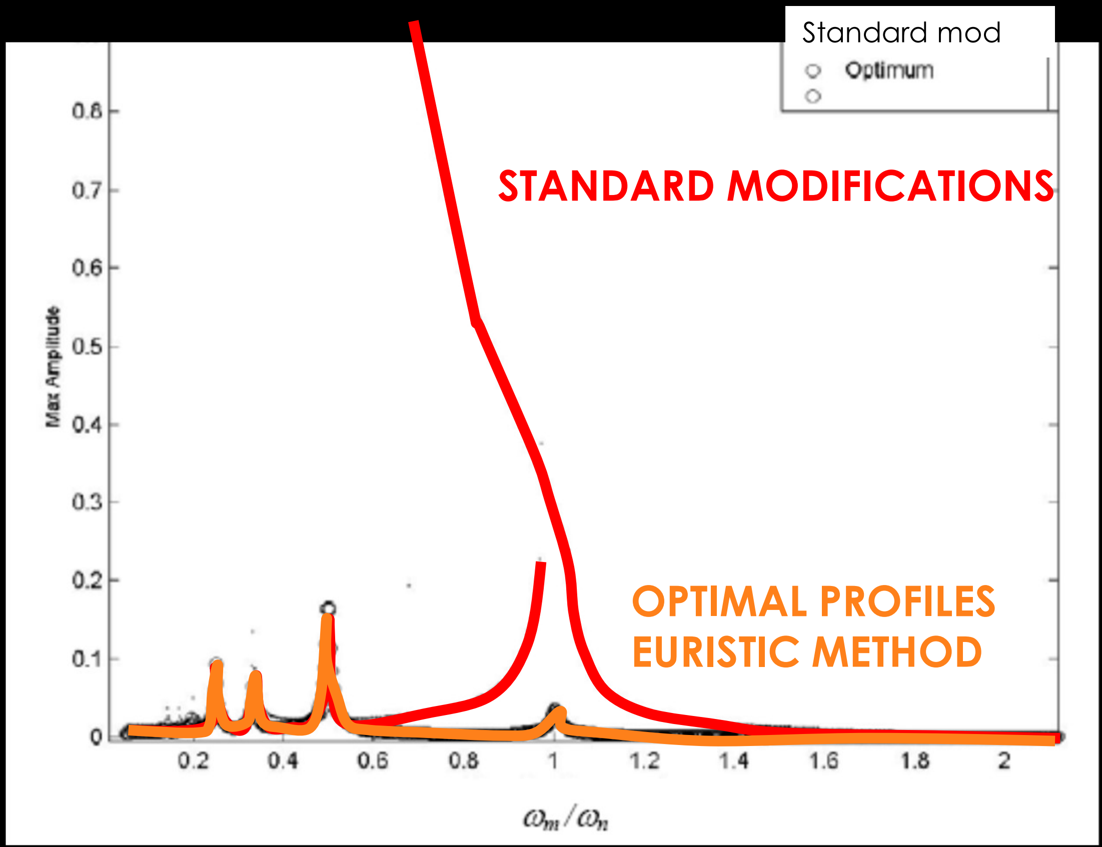
EURISTIC OPTIMIZATION: DYNAMICS



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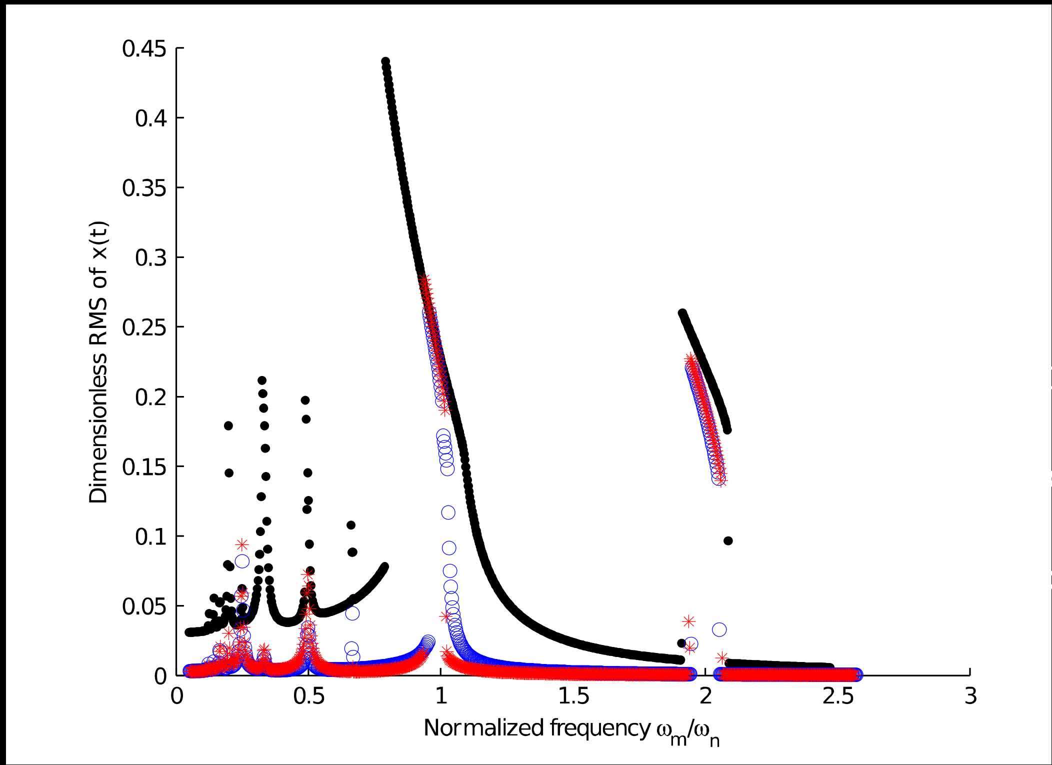
MODELLING TECHNIQUES: OPTIMIZATION



OVERALL VIBRATION REDUCTION AFTER GENETIC ALGORITHM OPT REASONABLY HIGH COMPUTATIONAL COST

on UNIMORE-CNH)

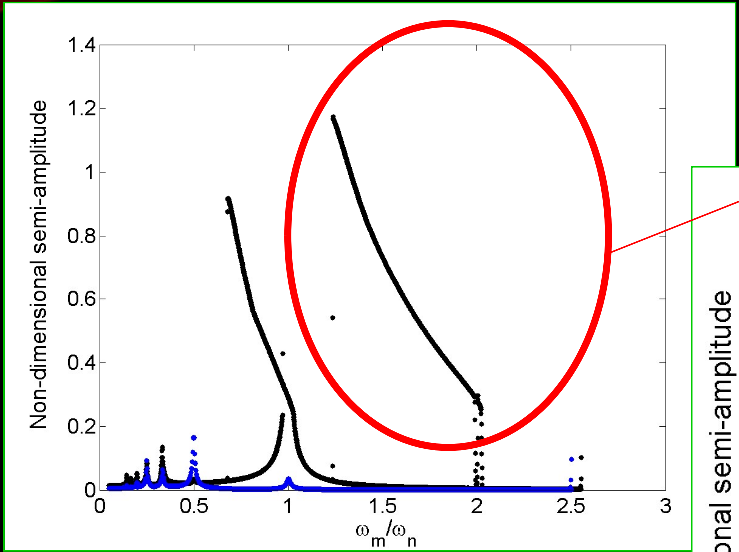
on UNIMORE-CNH)

ean project)

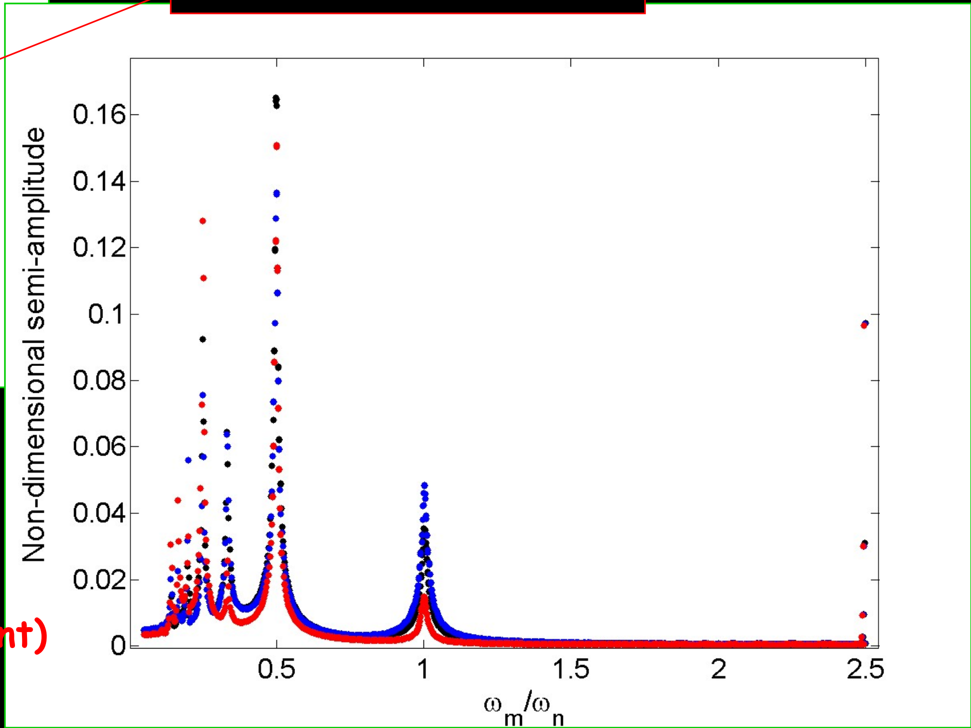
arboxes opt)

G. Bonori, M. Barbieri and F. Pellicano, "Optimum Profile Modifications of Spur Gears by Means of Genetic Algorithms", J. of Sound and Vibration, **313** (2008) 603–616.

EURISTIC VERSUS GENETIC OPTIMIZATION



PARAMETRIC INSTABILITY



BLACK: EURISTIC
BLUE: GENETIC (PPSTE)
RED: GENETIC (harmonic content)

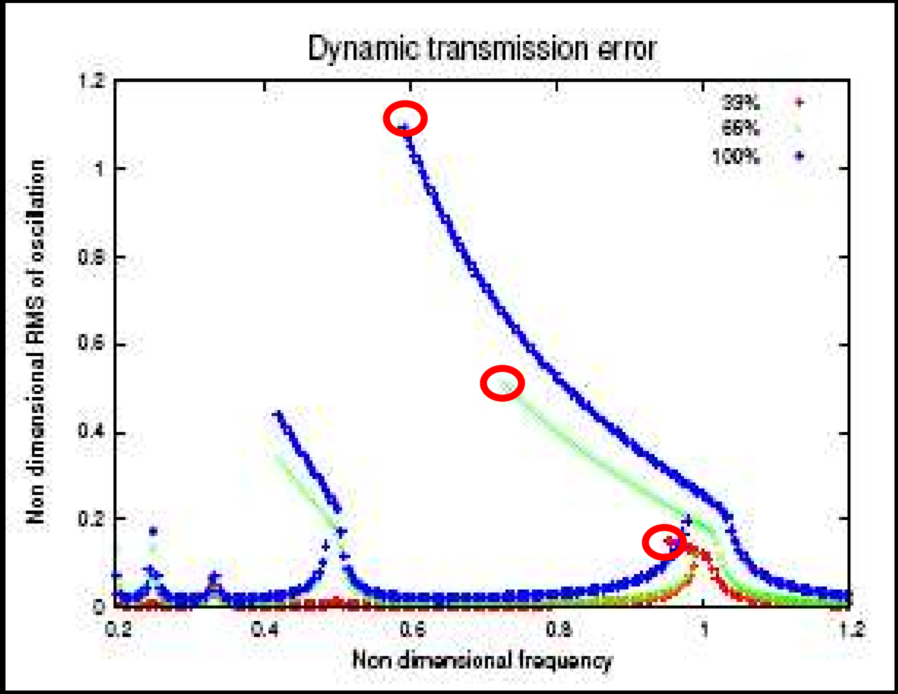
GENETIC ALGORITHM: GLOBAL OPTIMIZATION METHOD, 8 PARAMETERS

DYNAMIC OPTIMIZATION

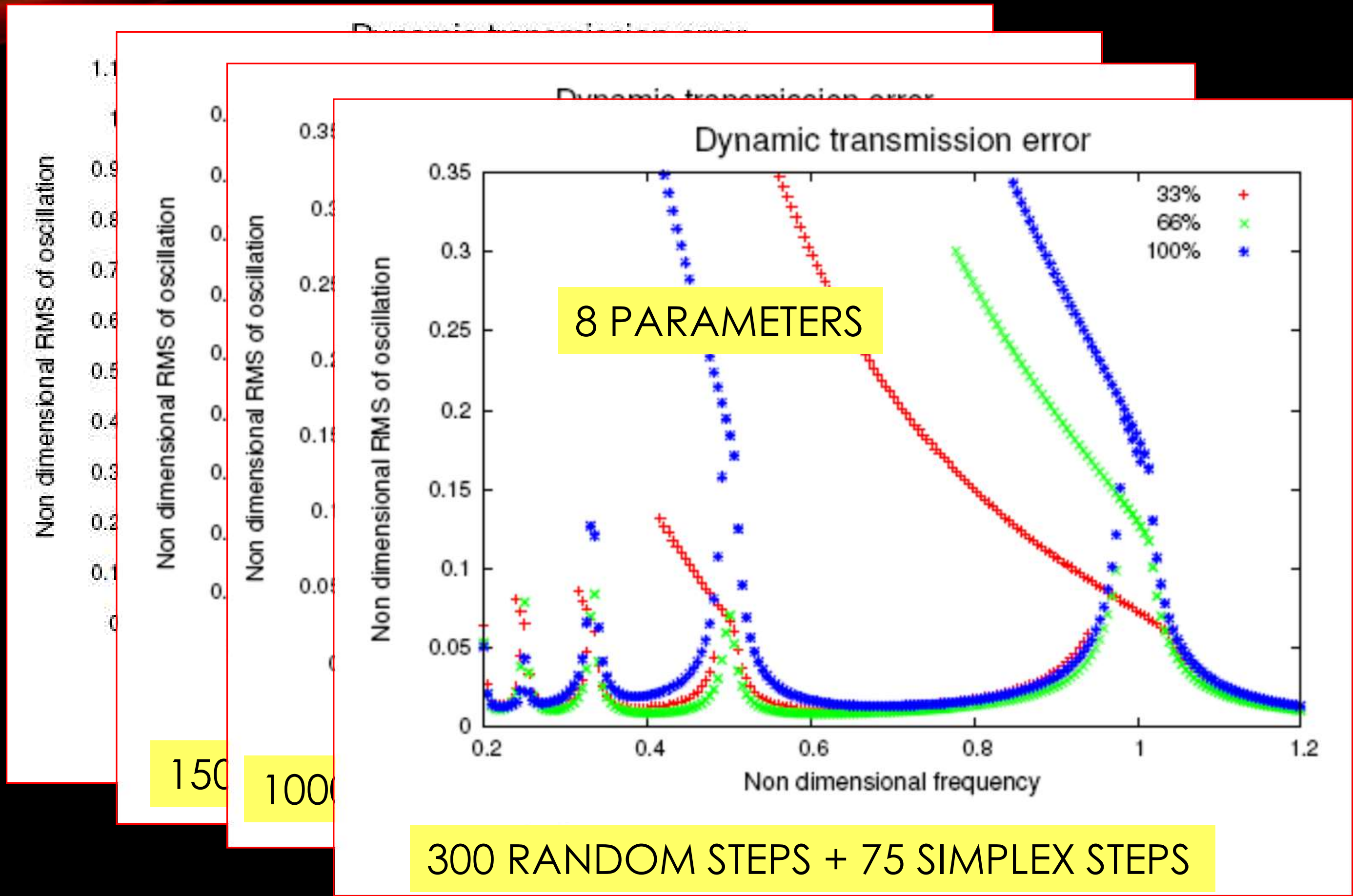
THE DYNAMICS CHANGES AS THE TRANSMITTED TORQUE VARIES

THE OBJECTIVE FUNCTION IS RELATED TO THE DYNAMIC SCENARIO

- **MAX AMPLITUDE**
- **AVERAGE RMS**
- **MAXIMUM RMS**
- **….**



DYNAMIC OPTIMIZATION



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COMPUTATIONAL COST

EVALUATION OF THE STATIC TRANSMISSION ERROR (or $k(t)$)
15 or more nonlinear FEM analyses

STATIC OPTIMIZATION

EURISTIC: 4×25^2 = 2500 PPTE evaluations; 2500 $\times$15=37500 FEM analyses

GENETIC ALGORITHMS 50 cases (population)$\times$100 iterations= 5000 PPTE eval.
75000 FEM ANALYSES

DYNAMIC OPTIMIZATION

RANDOM (1000) + SIMPLEX (150) 1150 STE evaluations: 17250 FEM ANALYSES
200 time histories for each dynamic scenario
230000 time histories

PLANETARY DYNAMIC MODELS

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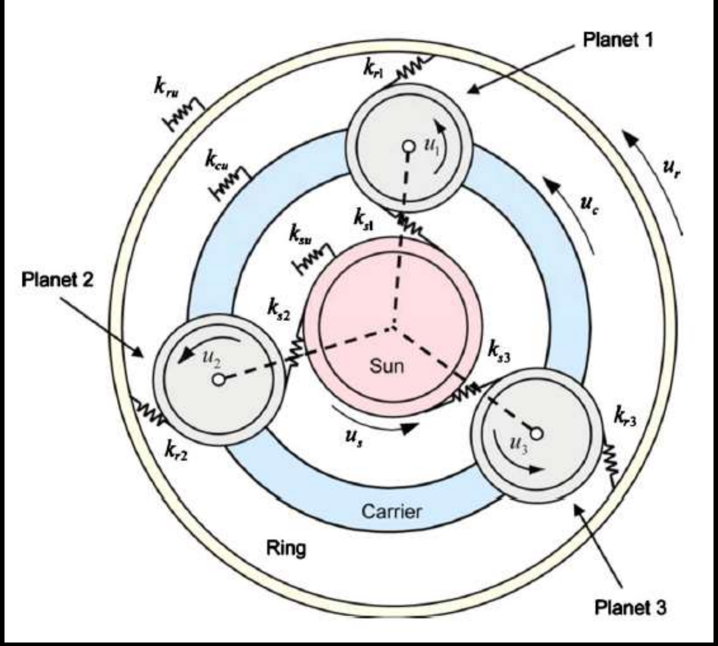
DYNAMIC MODELS: PLANETARY

- Concentrated parameter models
- Equivalent stiffness from nonlinear FEM analyses
- Linear models and invariant time: proper frequencies, modes
- Linear time-variant models: global static transmission error
- Nonlinear models: non-stationary dynamics and broadband responses (noise)



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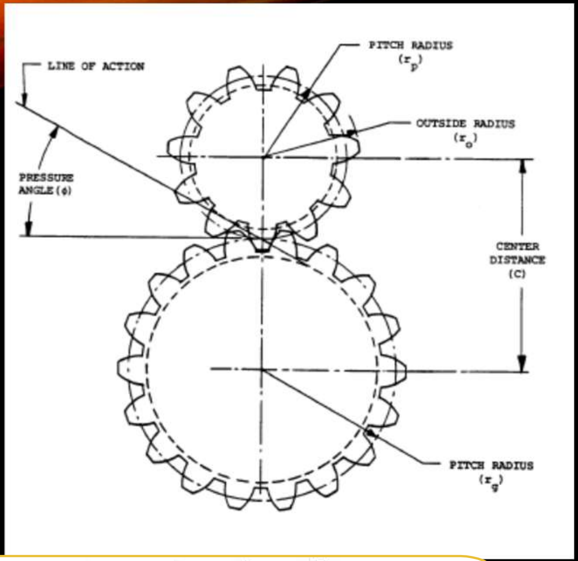
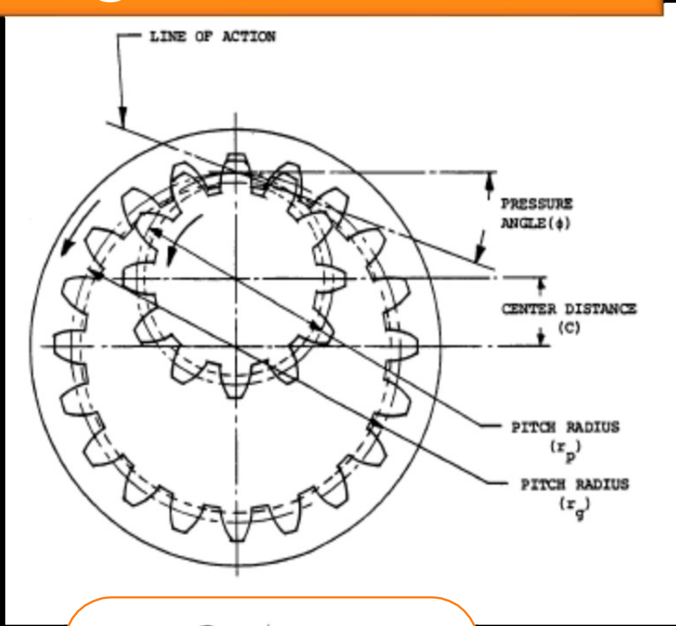
SUN-PLANET AND RING-PLANET MESH STIFFNESS

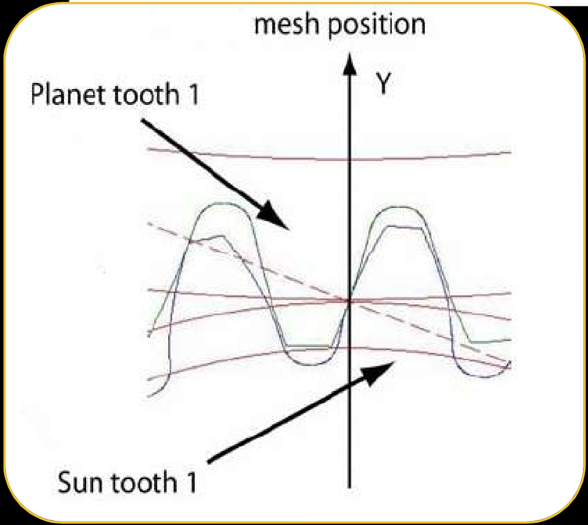
1. Gear geometry generation.
2. Automatic meshing of gears (FEM model and mounting).
3. Generation of an output, compatible with a FEM software (MSC/marc).
4. Launching calculations for a desired load case and extracting results in terms of meshing stiffness.

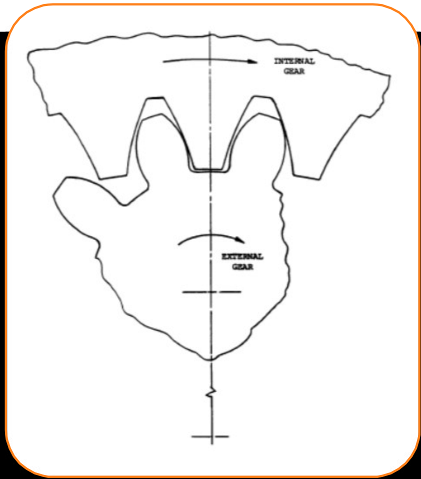
The following results have been obtained using the GearDesign software

Ring-Planet Mesh









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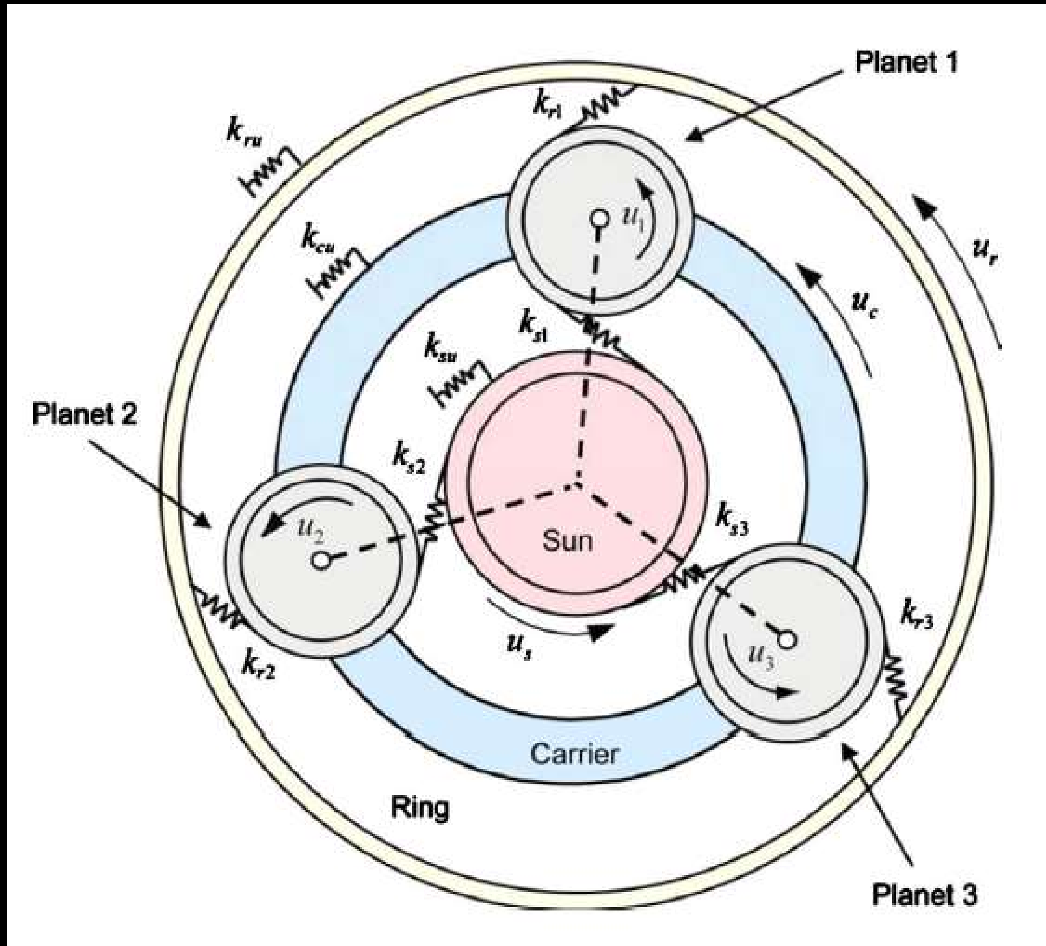
AMIC ODEL

CONCENTRATED PARAMETER MODEL ROTATIONAL DEGREES ONLY



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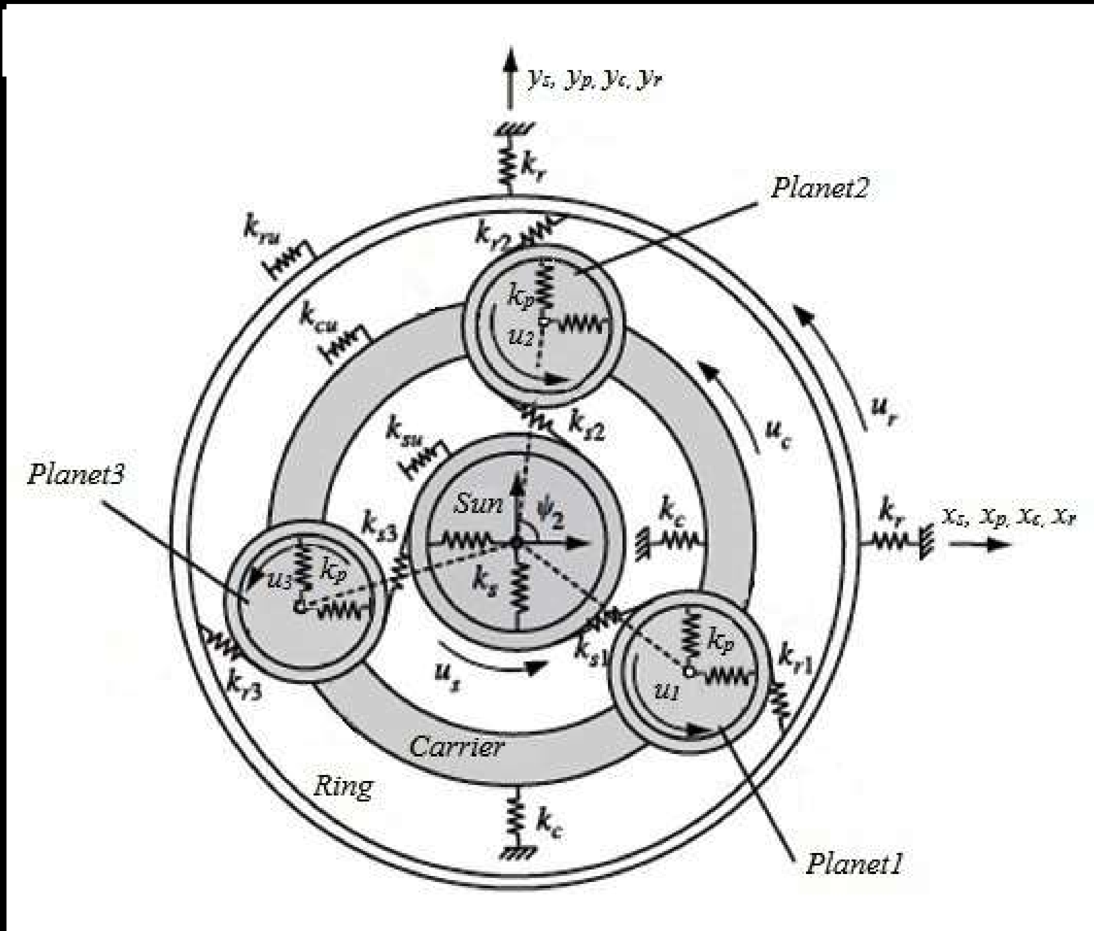
MIC
DEL

- LUMPED PARAMETER MODEL
- ROTATIONAL AND TRANSLATIONAL DEGREES
- TAKES INTO ACCOUNT THE COMPLIANCE OF THE BEARINGS



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SUN GEAR EQUATIONS

Inertia

Bearing Viscosity

Bearing Stiffness

$$-M_s.\ddot{x}_s - C_s.\dot{x}_s + \sum_{n=1}^{N}\Big[\big(c_{sn}(\dot{x}_n-\dot{x}_s).\sin^2(\psi_n-\alpha_s)\big) + \big(-c_{sn}(\dot{y}_n-\dot{y}_s).\sin(\psi_n -$$

$$\alpha_s).\cos(\psi_n-\alpha_s)\big) + \Big(c_{sn}\big(\dot\theta_s.r_{bs}+\dot\theta_n.r_{bn}\big).\sin(\psi_n-\alpha_s)\Big)\Big] - k_s.x_s +$$

Sun-Planet Mesh viscosity

$$\big(k_{sn}(f_{s\theta}^{***}).\sin(\psi_n-\alpha_s)\big)\Big] = 0$$

Piecewise Linear Functions for simulating sun-planet backlash

$$\dot{y}_s).\cos^2(\psi_n-\alpha_s)\big) + \Big(-c_{sn}\big(\dot\theta_s.r_{bs}+\dot\theta_n.r_{bn}\big).\cos(\psi_n-\alpha_s)\Big)\Big] - k_s.y_s +$$

$$\sum_{n=1}^{N}\Big[\big(-k_{sn}(f_{sx}^{*}).\sin(\psi_n-\alpha_s).\cos(\psi_n-\alpha_s)\big) + \Big(k_{sn}\big(f_{sy}^{**}\big).\cos^2(\psi_n-\alpha_s)\Big) +$$

$$\big(-k_{sn}(f_{s\theta}^{***}).\cos(\psi_n-\alpha_s)\big)\Big] = 0$$

x Translational DOF

y Translational DOF

$$T_s - I_s.\ddot\theta_s - C_{su}.\dot\theta_s + \sum_{n=1}^{N}\big[\big(-c_{sn}(\dot x_n-\dot x_s).\sin(\psi_n-\alpha_s).r_{bs}\big) + \big(c_{sn}(\dot y_n - \dot y_s).\cos(\psi_n -$$

$$\alpha_s).r_{bs}\big) + \big(-c_{sn}(\dot\theta_s.r_{bs}+\dot\theta_n.r_{bn}).r_{bs}\big)\big] - k_{su}.\theta_s + \sum_{n=1}^{N}\big[\big(-k_{sn}(f_{sx}^{*}).\sin(\psi_n-\alpha_s).r_{bs}\big) +$$

$$\big(k_{sn}(f_{sy}^{**}).\cos(\psi_n-\alpha_s).r_{bs}\big) + \big(-k_{sn}(f_{s\theta}^{***}).r_{bs}\big)\big] = 0$$

Rotational DOF

PIECEWISE LINEAR FUNCTIONS FOR SIMULATING SUN-PLANET BACKLASH

$$f_{sx} = \begin{cases} x_n - x_s - \dfrac{b_s}{\sin(\psi_n - \alpha_s)} & \Delta_s \geq b_s \\ 0 & |\Delta_s| < b_s \\ x_n - x_s + \dfrac{b_s}{\sin(\psi_n - \alpha_s)} & \Delta_s \leq -b_s \end{cases}$$

Contact
Tooth Separation
Backside Contact

$$f_{sy} = \begin{cases} y_n - y_s + \dfrac{b_s}{\cos(\psi_n - \alpha_s)} & \Delta_s \geq b_s \\ 0 & |\Delta_s| < b_s \\ y_n - y_s - \dfrac{b_s}{\cos(\psi_n - \alpha_s)} & \Delta_s \leq -b_s \end{cases}$$

Contact
Tooth Separation
Backside Contact

$$f_{s\theta} = \begin{cases} \theta_s . r_{bs} + \theta_n . r_{bn} - b_s & \Delta_s \geq b_s \\ 0 & |\Delta_s| < b_s \\ \theta_s . r_{bs} + \theta_n . r_{bn} + b_s & \Delta_s \leq -b_s \end{cases}$$

Contact
Tooth Separation
Backside Contact

$$\Delta_s = [(x_n - x_s).\sin(\psi_n - \alpha_s) - (y_n - y_s).\cos(\psi_n - \alpha_s) + (\theta_s . r_{bs} + \theta_n . r_{bn})]$$

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SYSTEM EQUATIONS

- THE EQUATIONS MUST BE REPLICATED FOR
- All satellites
- The ring
- The train carrier
- For a three-satellite system we obtain a system of 18 NON-SMOOTH (nonlinear) time-variant differential equations (the ring is locked)

VALIDATION

Comparisons with Cheon-Jae Bahk, R.G. Parker (2011) Journal of Computational and Nonlinear Dynamics **"Analytical Solution for the Nonlinear Dynamics of Planetary Gears"**

Parameters of the case study planetary gear set

Parameter	Sun	Ring	Planet	Carrier
Number of teeth	27	99	35	-
Module (mm)	2.8677	2.7782	2.8677	-
Pressure angle (deg)	24.60	20.19	24.60	-
Working Center Distance (mm)		88.89		-
Root diameter (mm)	70.485	284.150	91.440	-
Outer diameter (mm)	84.074	304.800	105.004	-
Inner diameter (mm)	57.15	271.73	73.66	-
Base diameter (mm)	70.40	258.130	91.26	177.80
Translational bearing stiffness (N/m)	-	-	-	-
Rotational bearing stiffness (N.m/Rad)	0	2.19e9	0	2.19e9
I/r^2 (Kg)	3.11	4.89		24.80
Mass (Kg)		2.64		

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COMPARISON OF NATURAL FREQUENCIES FOR PURE ROTATIONAL MODEL

Natural Freq (Hz)	Rotational model	Bahk-Parker model
ω_1	1847	1846
$\omega_3 = \omega_2$	760	2744
ω_4	4387	4379

Cheon-Jae **Bahk**, R.G. **Parker** , 2011. “**Analytical Solution** for the Nonlinear Dynamics of Planetary Gears”

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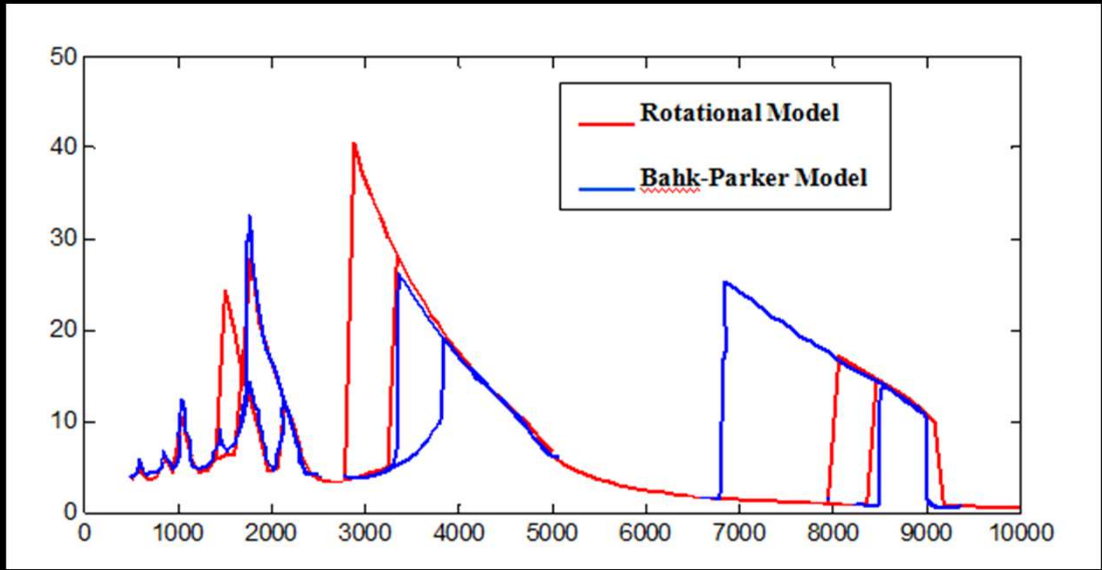
COMPARISON OF THE RMS OF SUN ROTATION (AFTER SUBTRACTION OF THE MEAN VALUE)



The resonance peaks lean to the left, implying **softening nonlinearity** induced by tooth separation.

There are additional resonance peaks around 8000 Hz, 1800 Hz the first distinct mode, and below 1600 Hz mesh frequency. These peaks are the combined effects of parametric instability from higher harmonics of mesh stiffness variation and **nonlinear subharmonic and superharmonic** resonances of the first and second distinct modes.

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COMPARISON OF NATURAL FREQUENCIES FOR ROTATIONAL-TRANSLATIONAL MODEL

Natural Freq (Hz)	Rotational model	Bahk-Parker model
ω1= ω2	1758	1760
ω3	2091	2095
ω4= ω5	3352	3390
ω6	5245	5249

Cheon-Jae **Bahk**, R.G. **Parker** , 2011. "**Analytical Solution** for the Nonlinear Dynamics of Planetary Gears"

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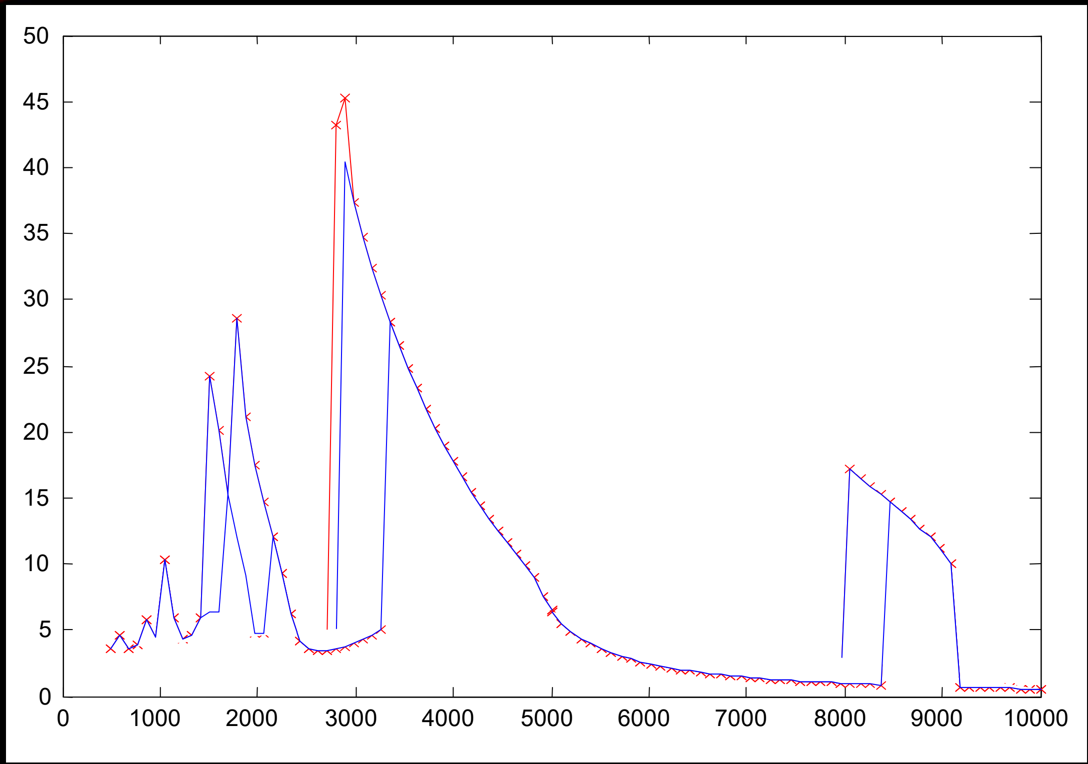
COMPARISON OF THE PURE ROTATIONAL MODEL (BLUE LINE) WITH ROTATIONAL-TRANSLATIONAL MODEL (RED LINE)



Pure rotational model has 4 DOF (rotation of sun and three planets) and the second model has 12 DOF (one rotation and two translation for sun and each planet)

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PLANETARY GEARS

COMPLEXITY
AND
RELIEFS OPTIMIZATION

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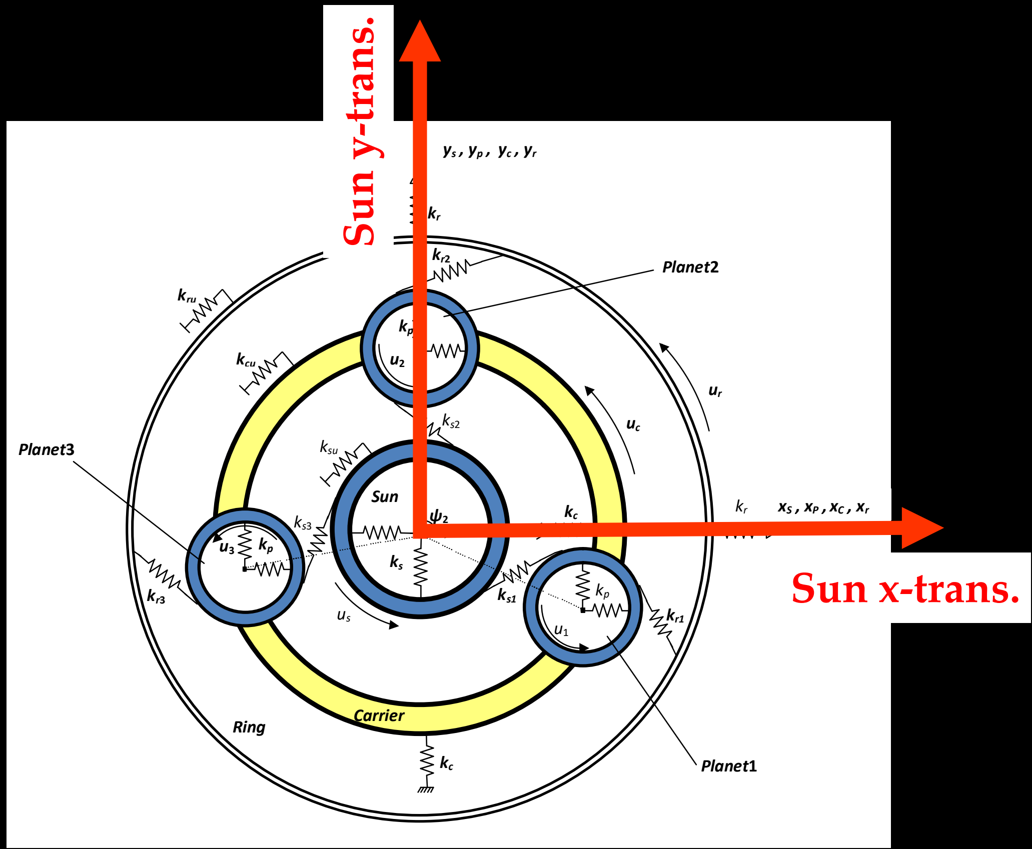
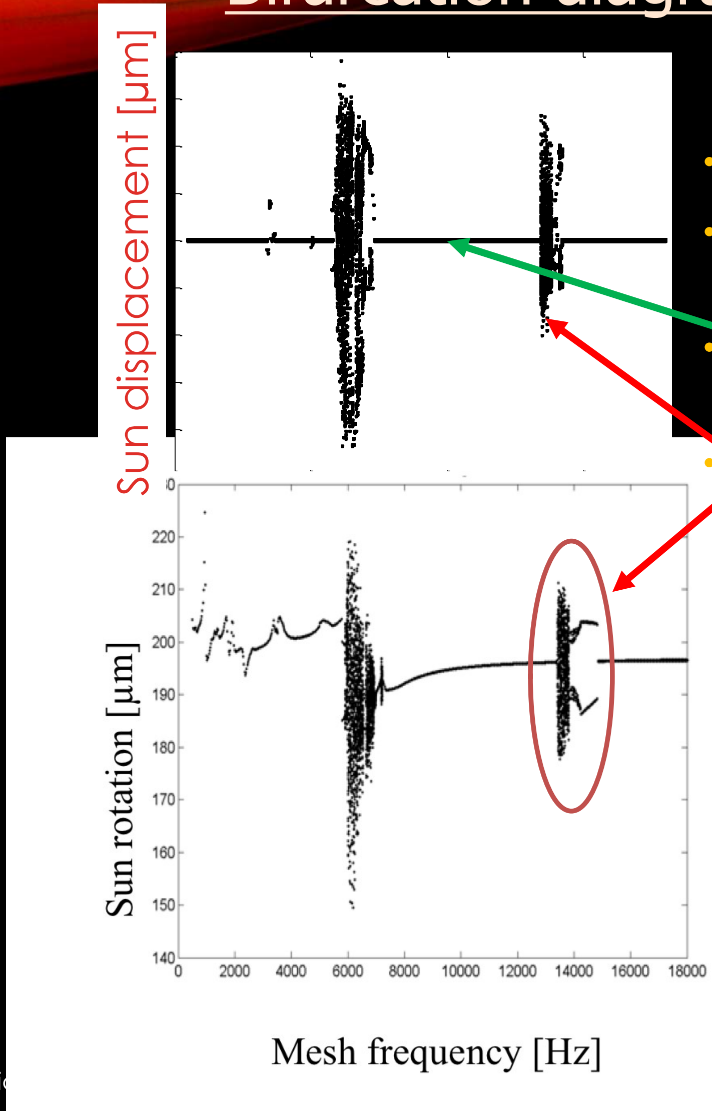
Dynamic imbalance on the sun (and compensation)

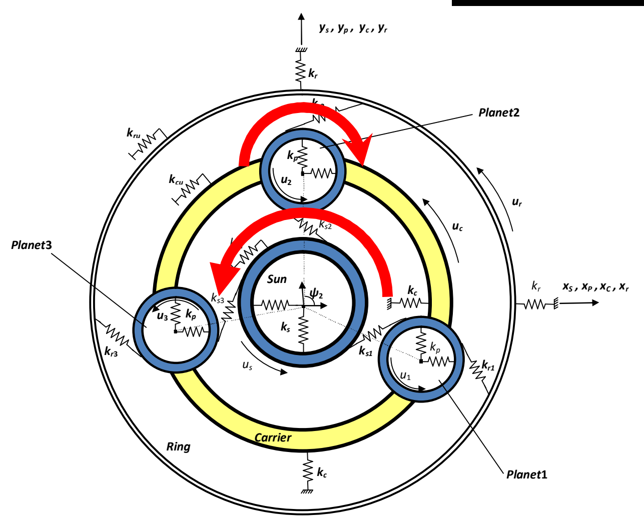
- With 3 planets placed at 120 degrees and phased, forces acting on the sun should be balanced
- When chaos occurs, the sun can loose balancing



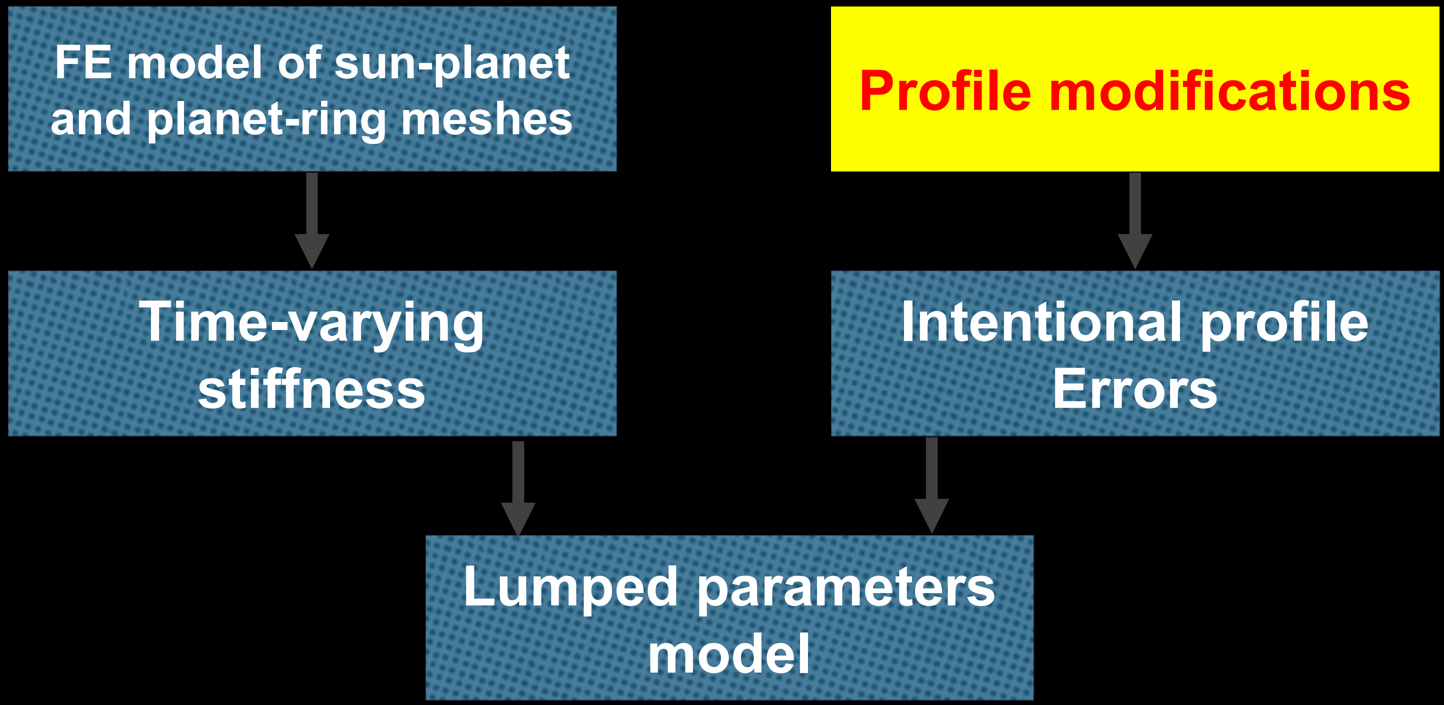
Bifurcation diagrams



- Rotation reduced along the line of action $\theta \times R_{base}$
- Non smooth period doubling bifurcation and chaos
- Sun displacement should be zero due to perfect balancing and phasing
- Chaos induces imbalance



DYNAMIC MODEL OF A PLANETARY GEAR WITH MODIFICATIONS (COMPENSATION)



- The first step is modeling both meshes by FEM, using our software HPGA
  - Later on, rigid rotations due to profile modifications are taken into account

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where,

APPLYING TOOTH PROFILE MODIFICATIONS

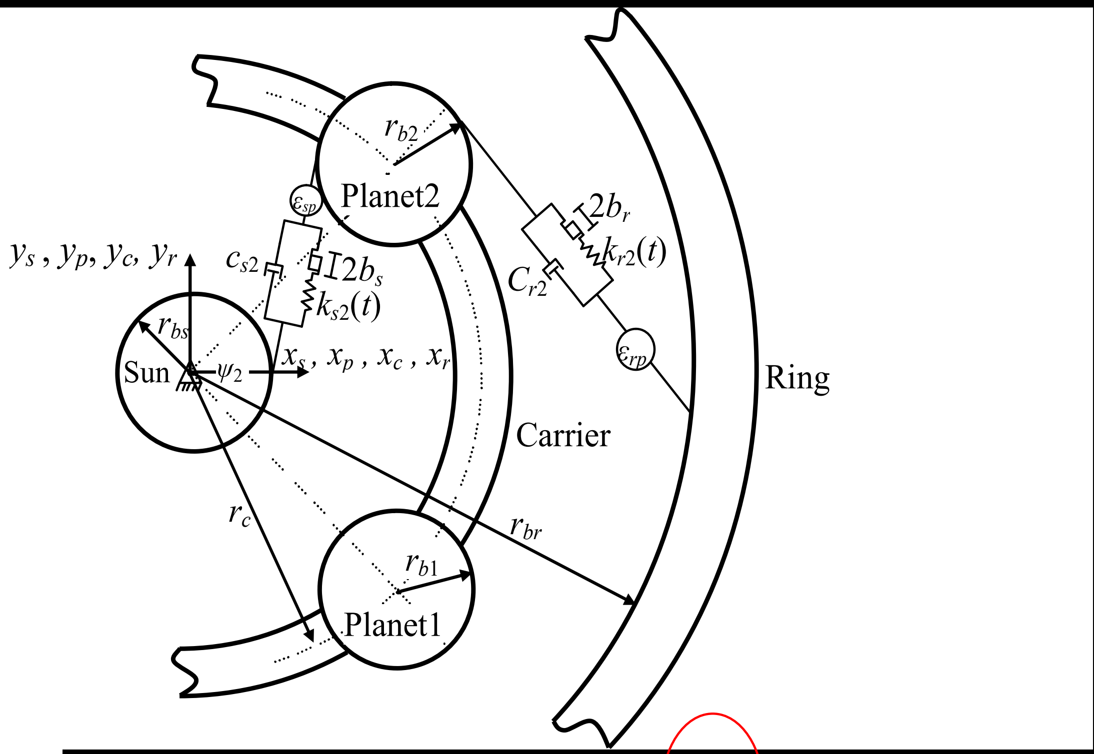
Displacement Function's Correction

piecewise-linear displacement functions for sun-planets meshing:

$$f_{sx} = \begin{cases} x_n - x_s - \dfrac{\varepsilon_{sp}}{\sin(\psi_n - \alpha_s)} \\ 0 \\ x_n - x_s + \dfrac{-\varepsilon_{sp} + 2b_s}{\sin(\psi_n - \alpha_s)} \end{cases}$$

$$f_{sy} = \begin{cases} y_n - y_s - \dfrac{\varepsilon_{sp}}{\cos(\psi_n - \alpha_s)} \\ 0 \\ y_n - y_s - \dfrac{(2b_s + \varepsilon_{sp})}{\cos(\psi_n - \alpha_s)} \end{cases}$$

$$f_{st} = \begin{cases} \theta_s . r_{bs} + \theta_n . r_{bn} + \varepsilon_{sp} \\ 0 \\ \theta_s . r_{bs} + \theta_n . r_{bn} - \varepsilon_{sp} + 2b_s \end{cases}$$



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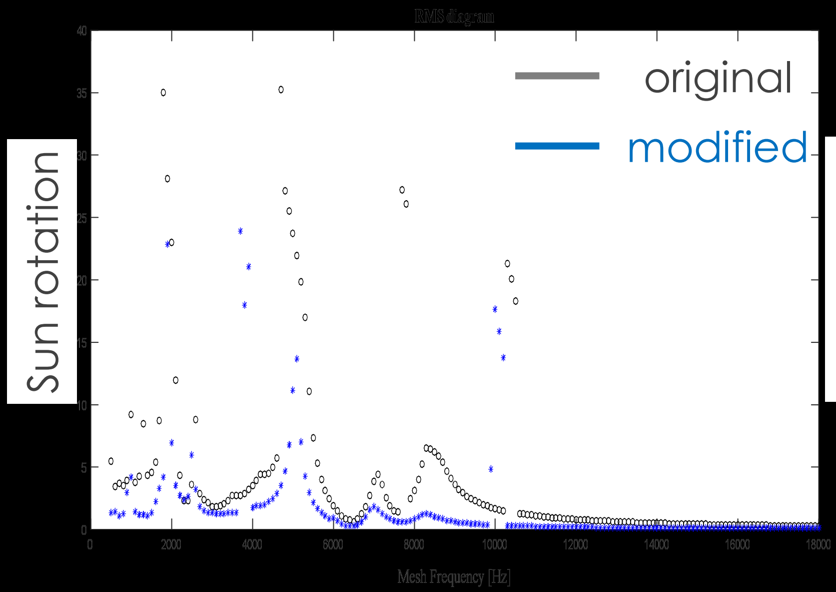
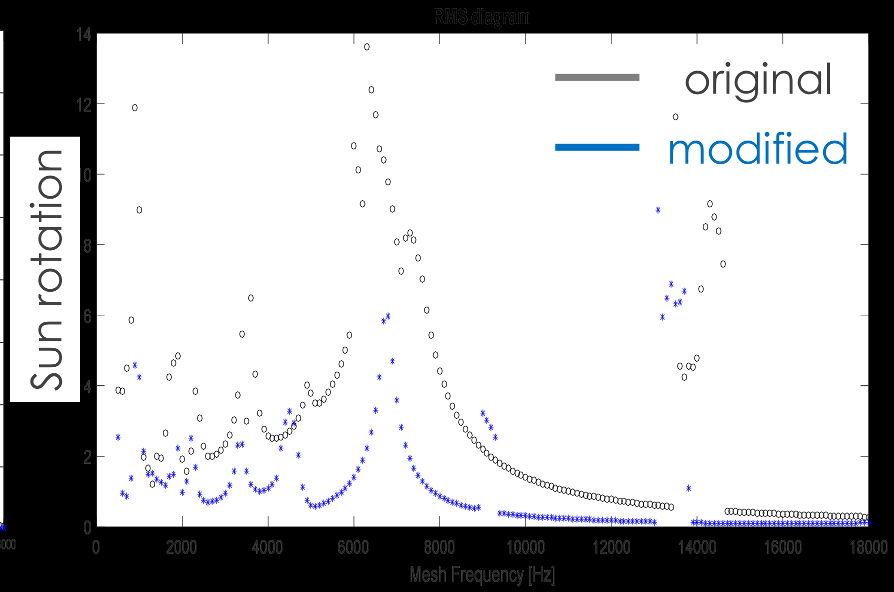
EFFECT OF PROFILE MODIFICATIONS

VIBRATION AMPLITUDE-FREQUENCY



(Planet translational bearing stiffness
is equal to 2.19e9 N/m)



(Planet translational bearing stiffness
is equal to 2.19e8 N/m)

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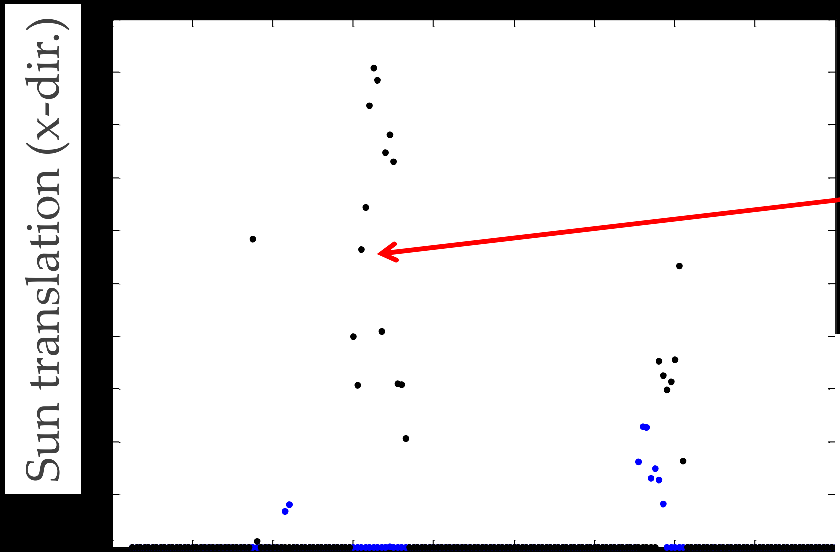
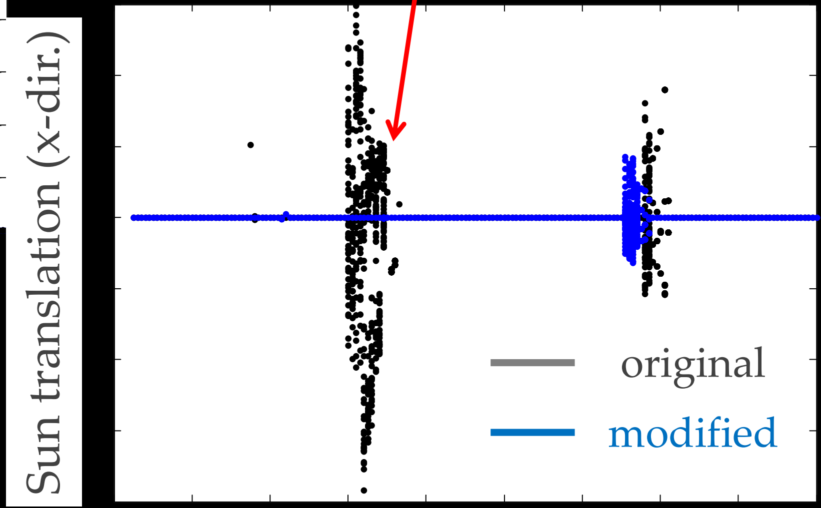
DYNAMIC IMBALANCE ON THE SUN (AND COMPENSATION)

VIBRATION AMPLITUDE



CHAOS INDUCES IMBALANCE

BIFURCATION DIAGRAM



PROFILE MODIFICATIONS CAN REDUCE THE CHAOS INDUCED IMBALANCE

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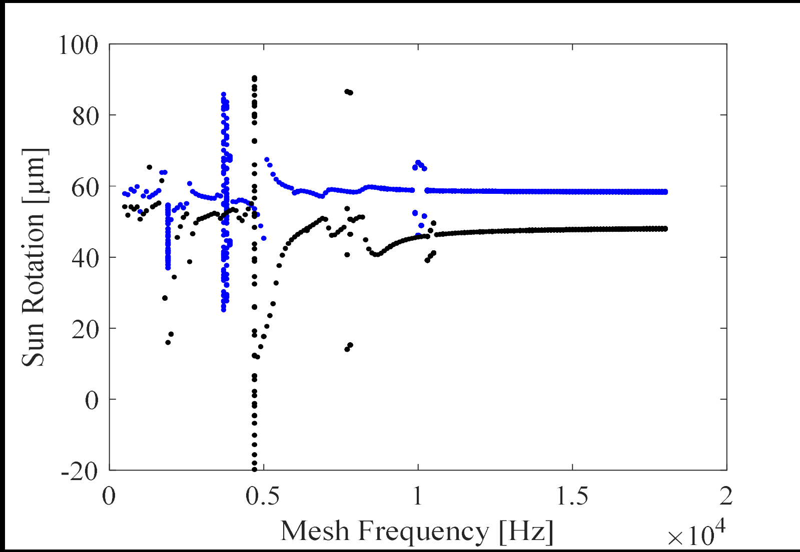
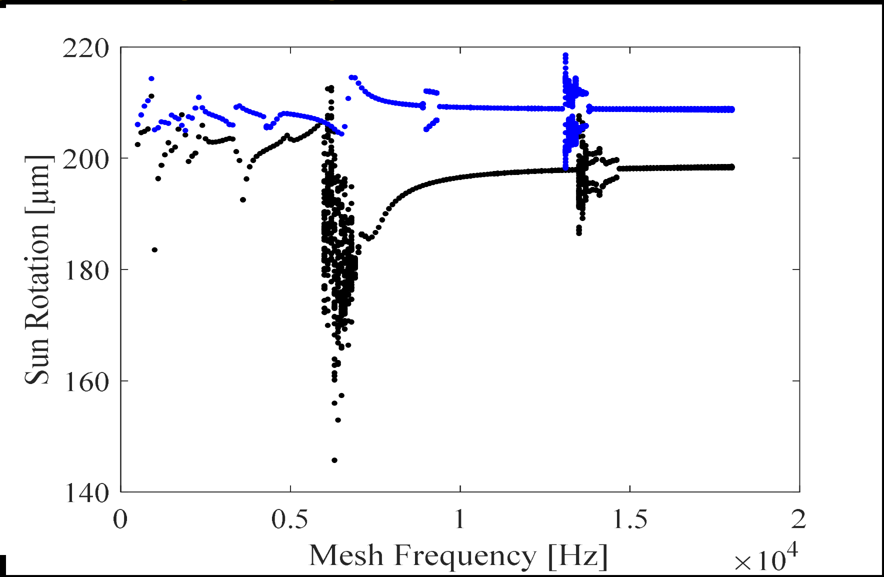
Mesh Frequency [Hz]

EFFECT OF PROFILE MODIFICATION

BIFURCATION DIAGRAMS



(Planet translational bearing stiffness is equal to 2.19e9 N/m)



(Planet translational bearing stiffness is equal to 2.19e8 N/m)

- The bifurcation diagrams show that instability regions are smaller for modified gears
- Instability disappears for the resonance at 6900 Hz

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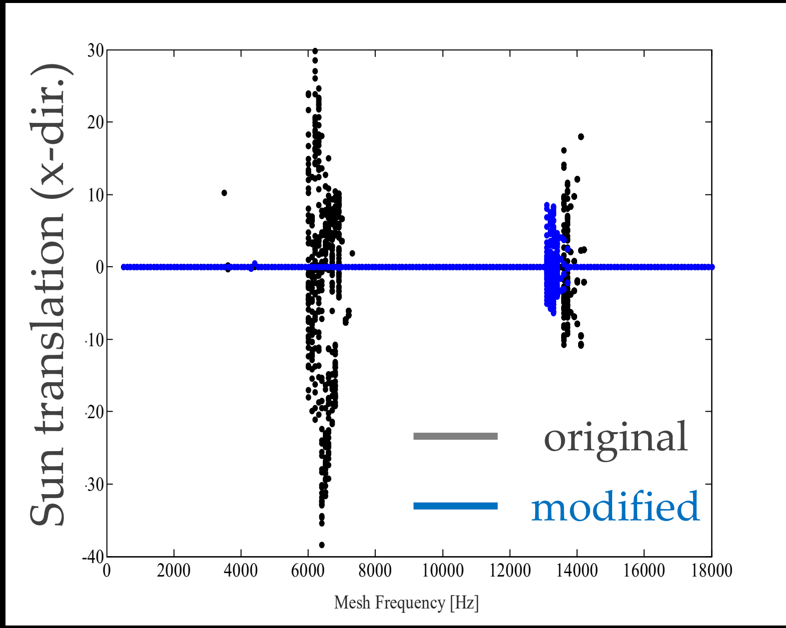
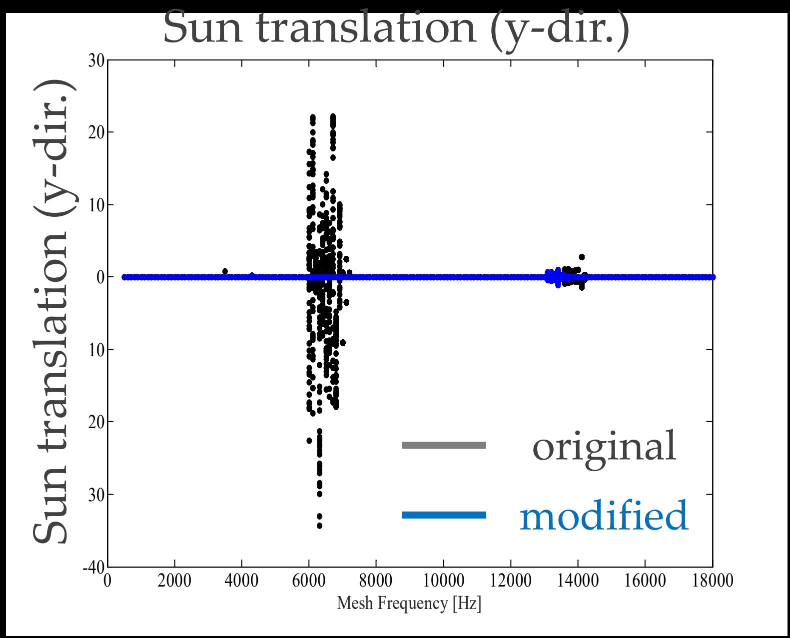
Dynamic imbalance on the sun (and compensation)

BIFURCATION DIAGRAMS





- With 3 planets placed at 120 and phased, forces acting on the sun should be balanced
- When chaos occurs, the sun is no longer balanced

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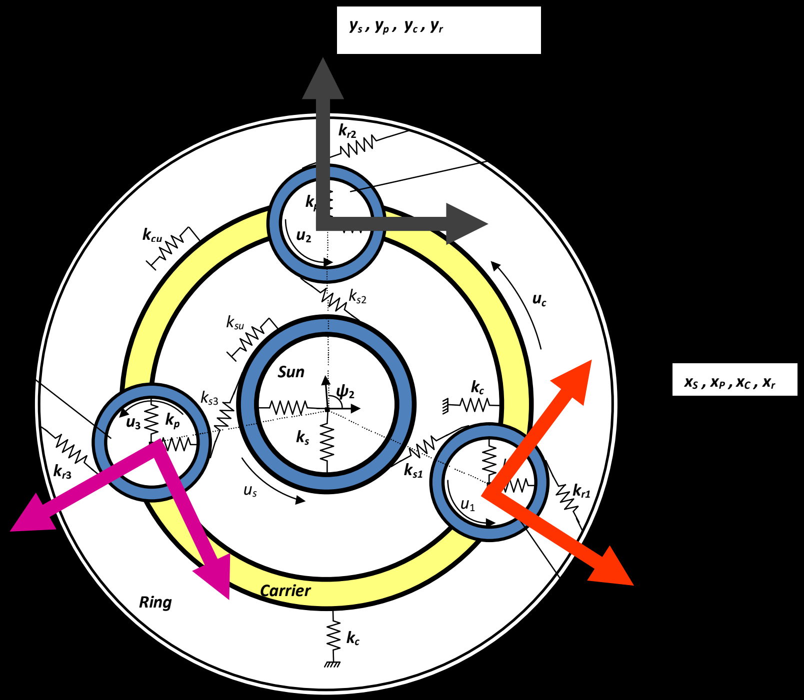
PLANETS RADIAL DISPLACEMENT

Radial and tangential displacements



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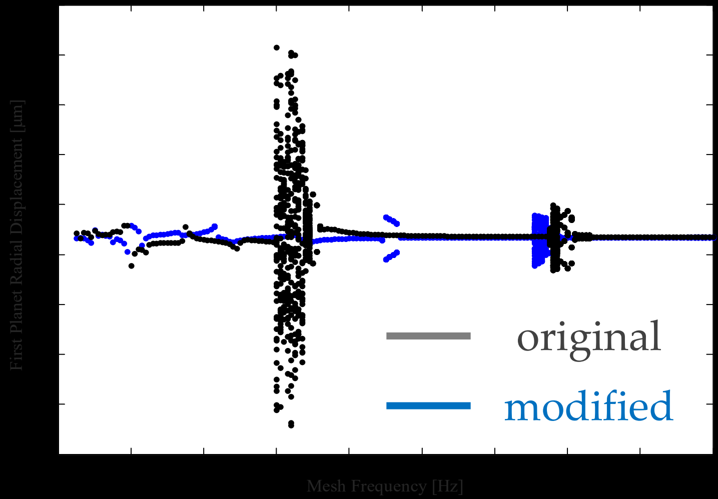
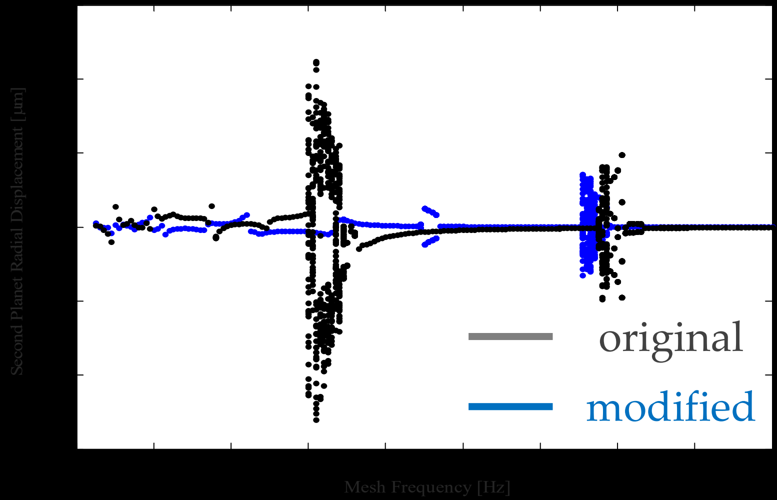
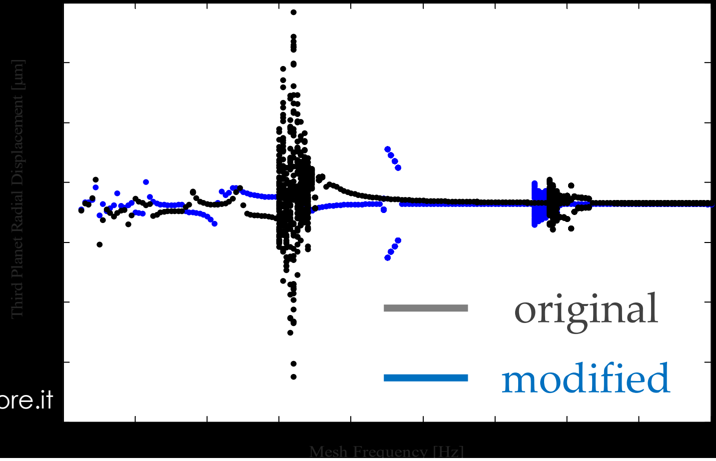
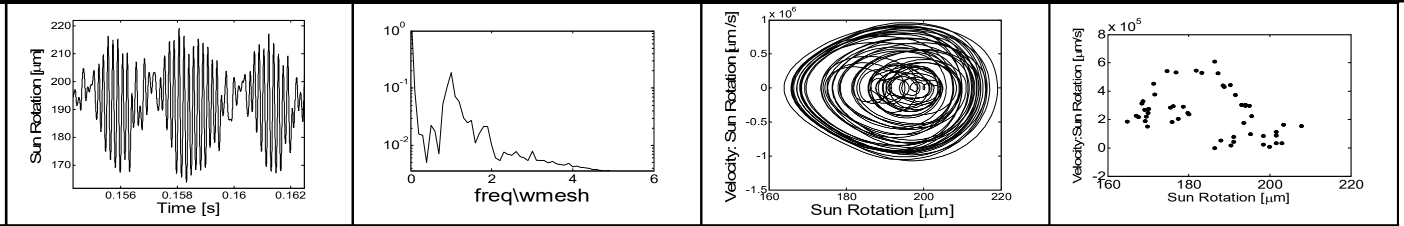
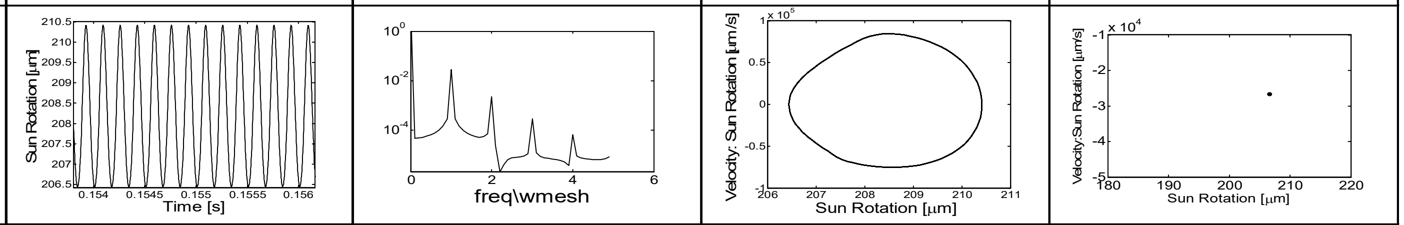
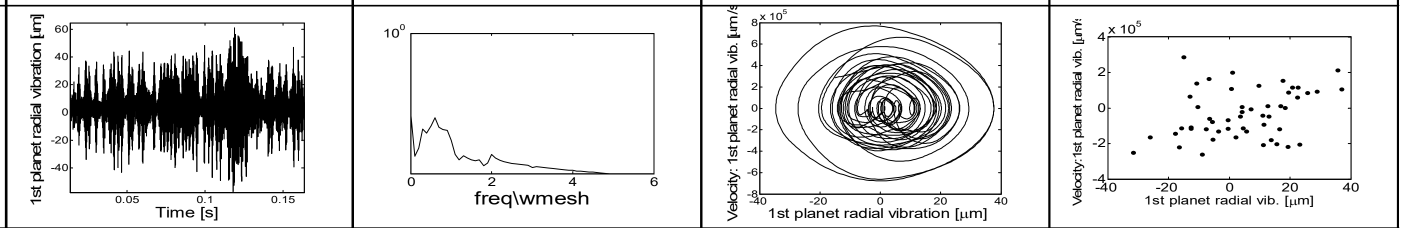
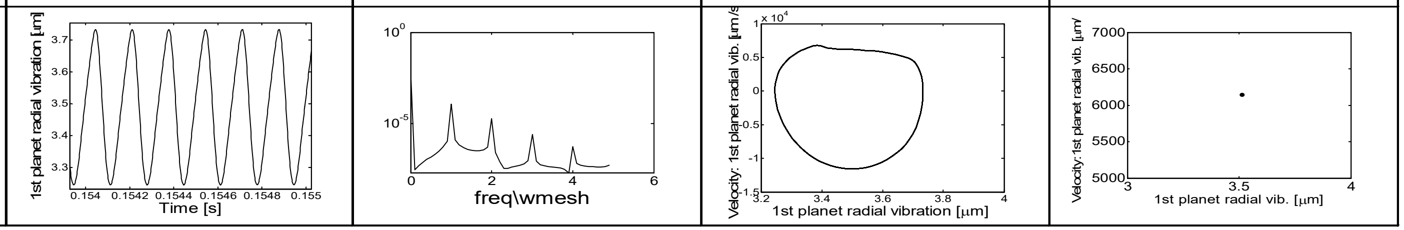
PLANETS RADIAL DISPLACEMENT



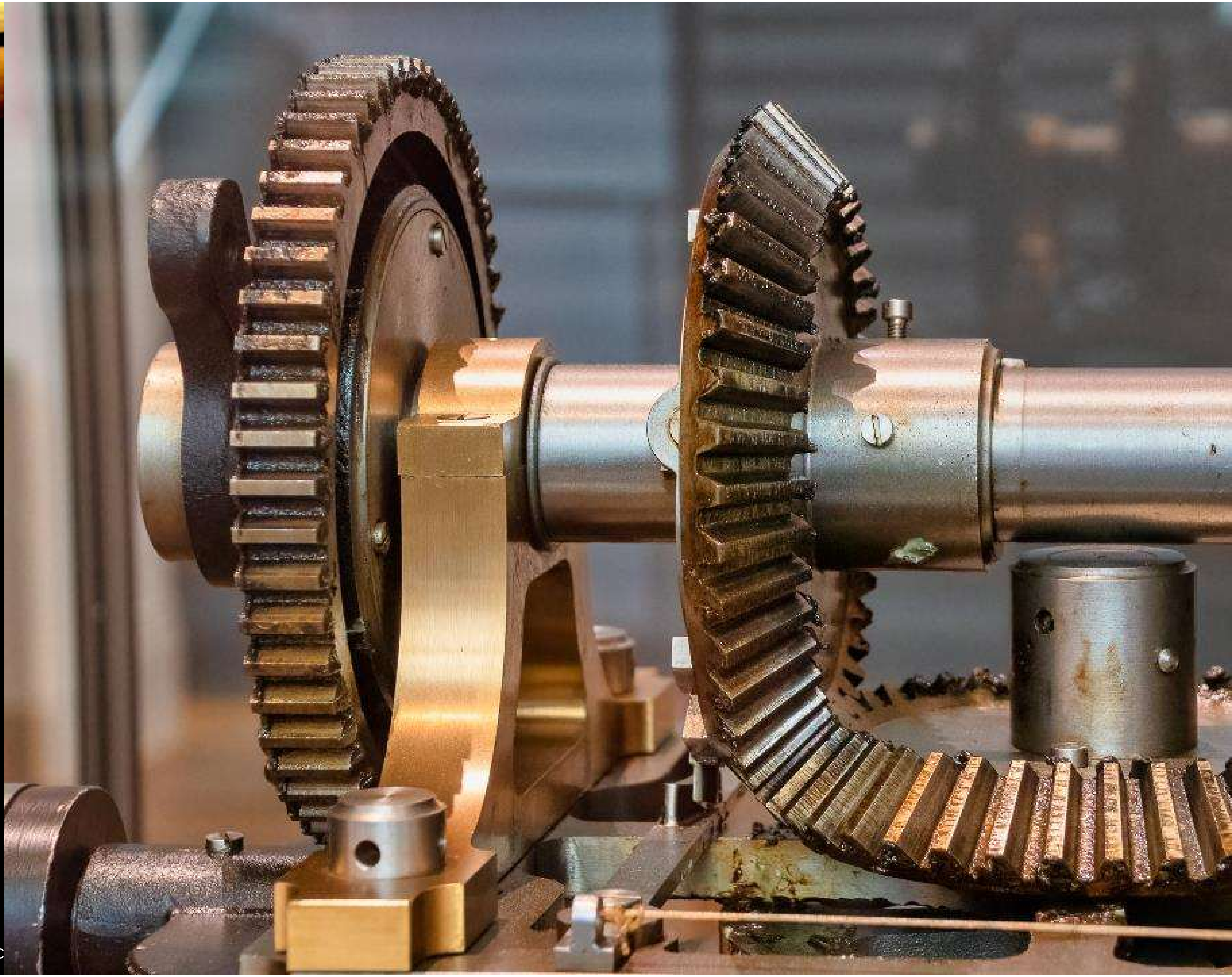




EFFECT OF MODIFICATION

Case 2: orig 6000 Hz Sun Rotation	
Case 2: opt 6000 Hz Sun Rotation	
Case 2: orig 6000 Hz 1st planet Radial	
Case 2: opt 6000 Hz 1st planet Radial	

TESTING FACILITIES

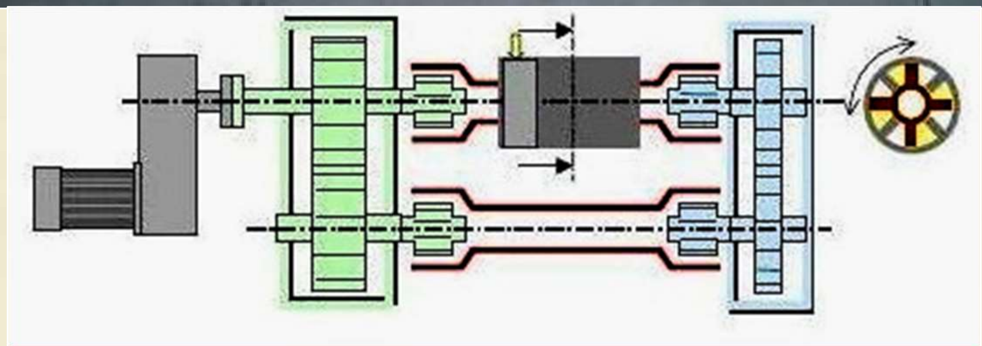


Test Rig 1:

Goal:
- Endurance Analysis
- Stress measurement

Software:
ANSOL. Transmission3D

System:
Back-to-back configuration

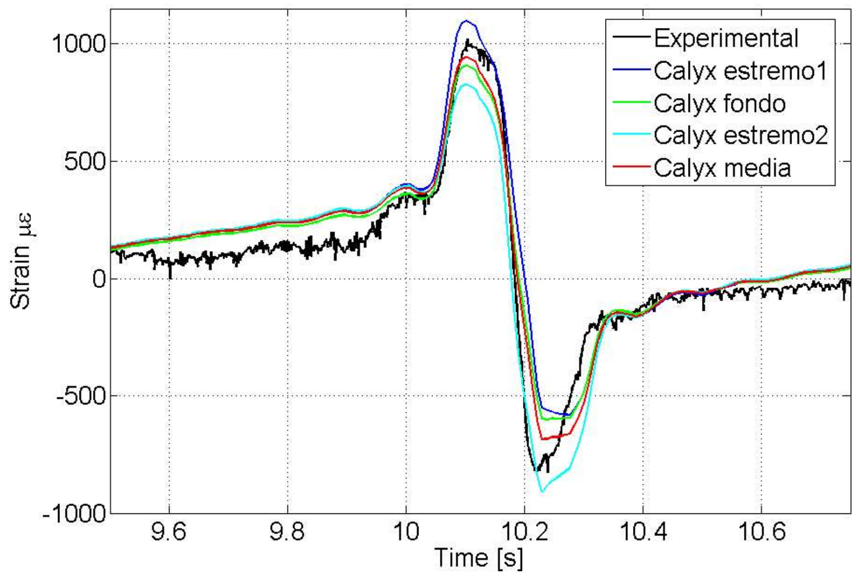


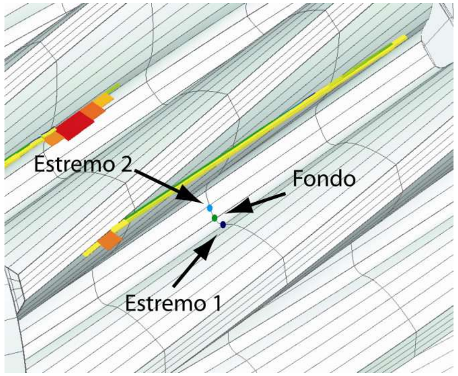
Test Rig 1: Validation

Conducted comparision:
- Simulation based on FEM
- Experimental test

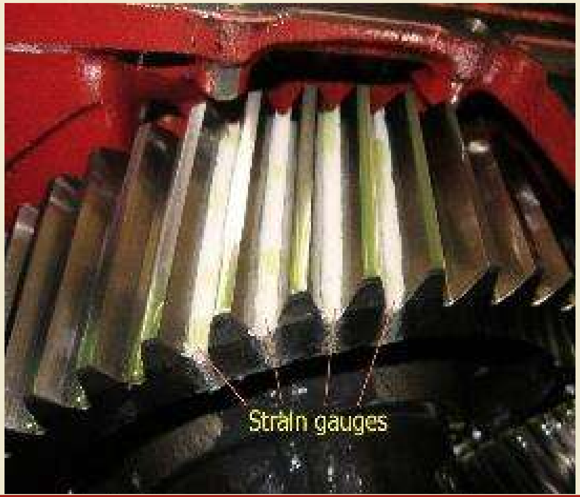
Goal:
Defining Accuracy

FEM-based Software:
ANSOL. Transmission3D





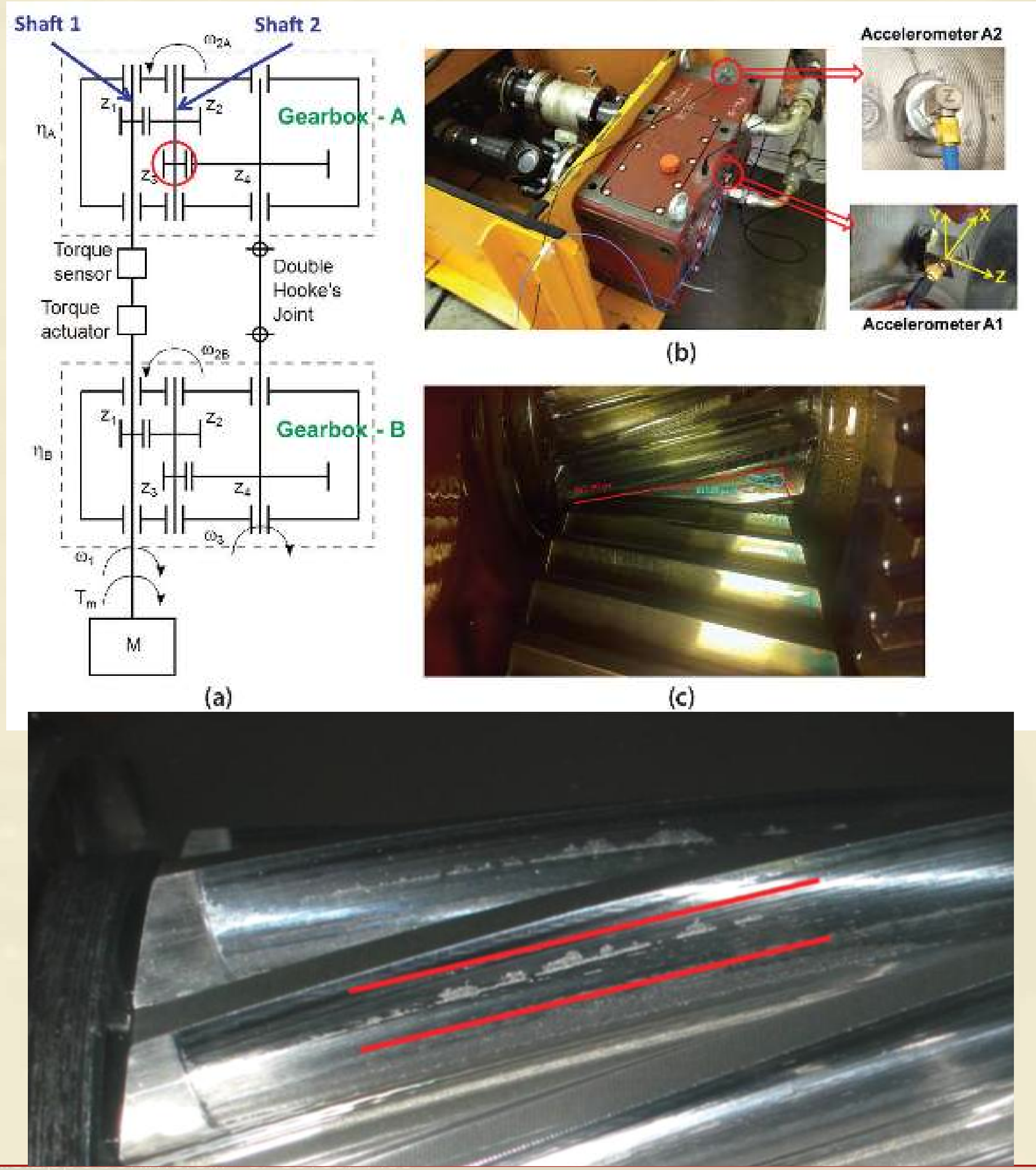
Error on peak max -5.7%
1000 microstrain vs 943 microstrain
(considering average



Test Rig 1: Test

Goal:
Endurance test

Results:
- Overload >> 100% nominal torque
- Pitting in reasonable time
- Vibration data recorded for validating condition monitoring techniques: 5% pitting detectable



(a)
(b)
(c)

Test Rig 1: Publication

Authors:
Gelman, Len, N. Harish Chandra, Rafal Kurosz, **Francesco Pellicano**, Marco Barbieri, **Antonio Zippo**.

Title:
Novel spectral kurtosis technology for adaptive vibration condition monitoring of multi-stage gearboxes.

Publisher:
The British Institute of Non-Destructive Testing

DOI:
10.1784/insi.2016.58.8.409

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VIBRATION ANALYSIS

Novel spectral kurtosis technology for adaptive vibration condition monitoring of multi-stage gearboxes

L Gelman, N Harish Chandra, R Kurosz, F Pellicano, M Barbieri and A Zippo

In this paper, the novel wavelet spectral kurtosis (WSK) technique is applied for the early diagnosis of gear tooth faults. Two variants of the wavelet spectral kurtosis technique, called variable resolution WSK and constant resolution WSK, are considered for the diagnosis of pitting gear faults. The gear residual signal, obtained by filtering the gear mesh frequencies, is used as the input to the SK algorithm. The advantages of using the wavelet-based SK techniques when compared to classical Fourier transform (FT)-based SK is confirmed by estimating the toothwise Fisher's criterion of diagnostic features. The final diagnosis decision is made by a three-stage decision-making technique based on the weighted majority rule. The probability of the correct diagnosis is estimated for each SK technique for comparison. An experimental study is presented in detail to test the performance of the wavelet spectral kurtosis techniques and the decision-making technique.

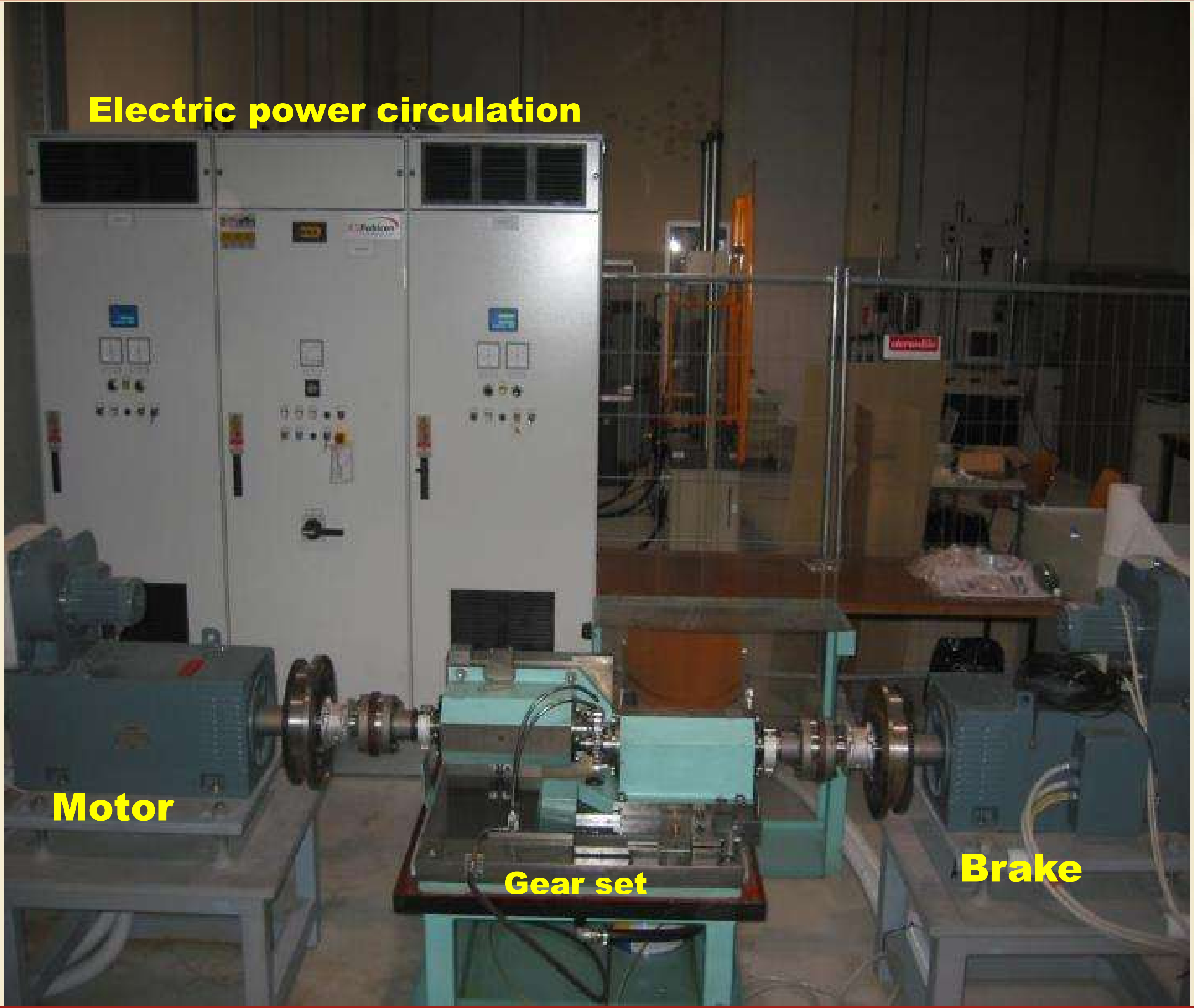
Test Rig 2

Colaboration:
cooperation with CNH

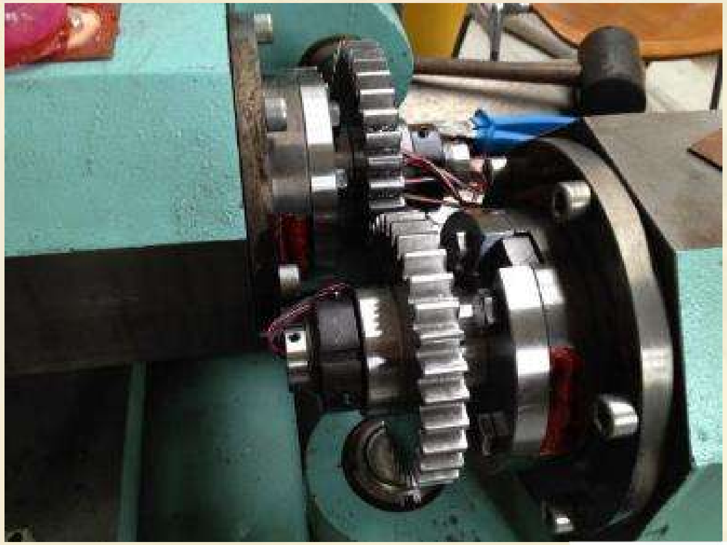
Goal:
Vibration measurement

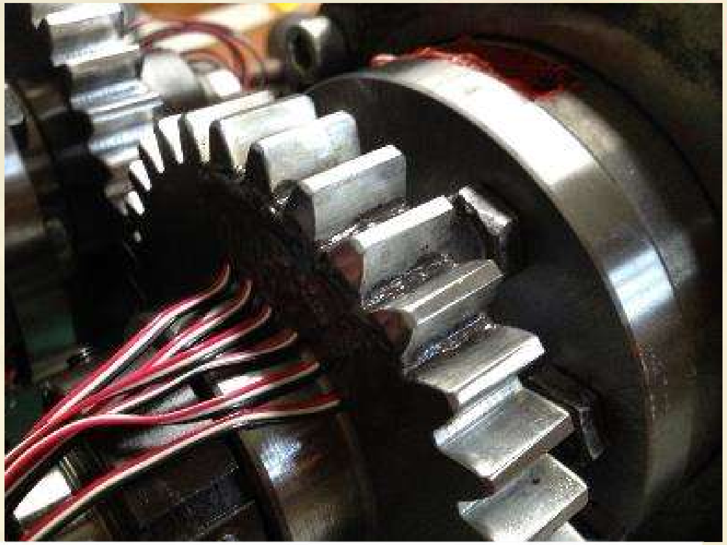
System:
Open loop configuration

Fully sensorized
- Torquemeters
- Tone wheel
- Accelerometers (on the shaft)
- Strain gauges
- Slip rings



Test Rig 2:

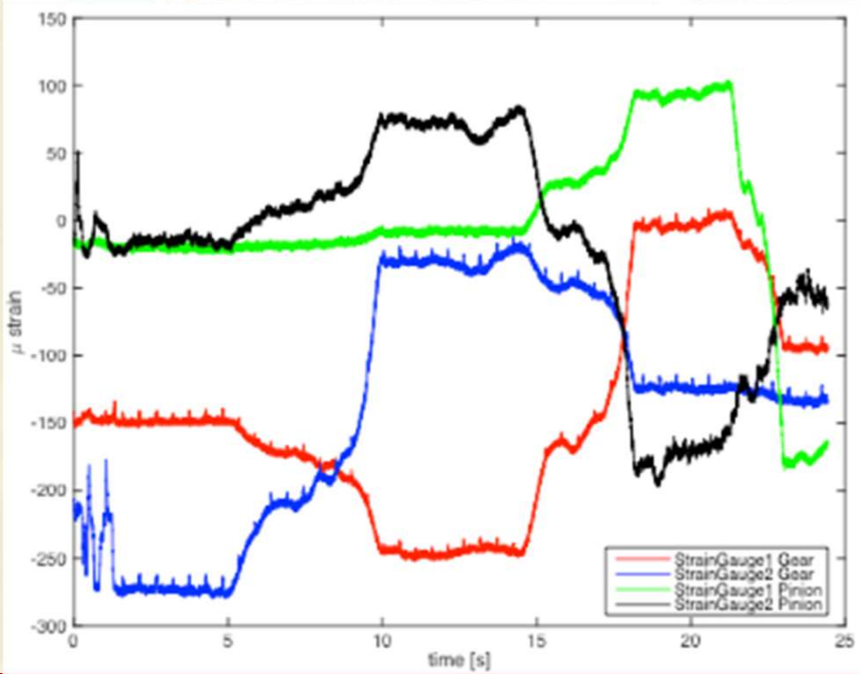




Goal:

Root stress measurement

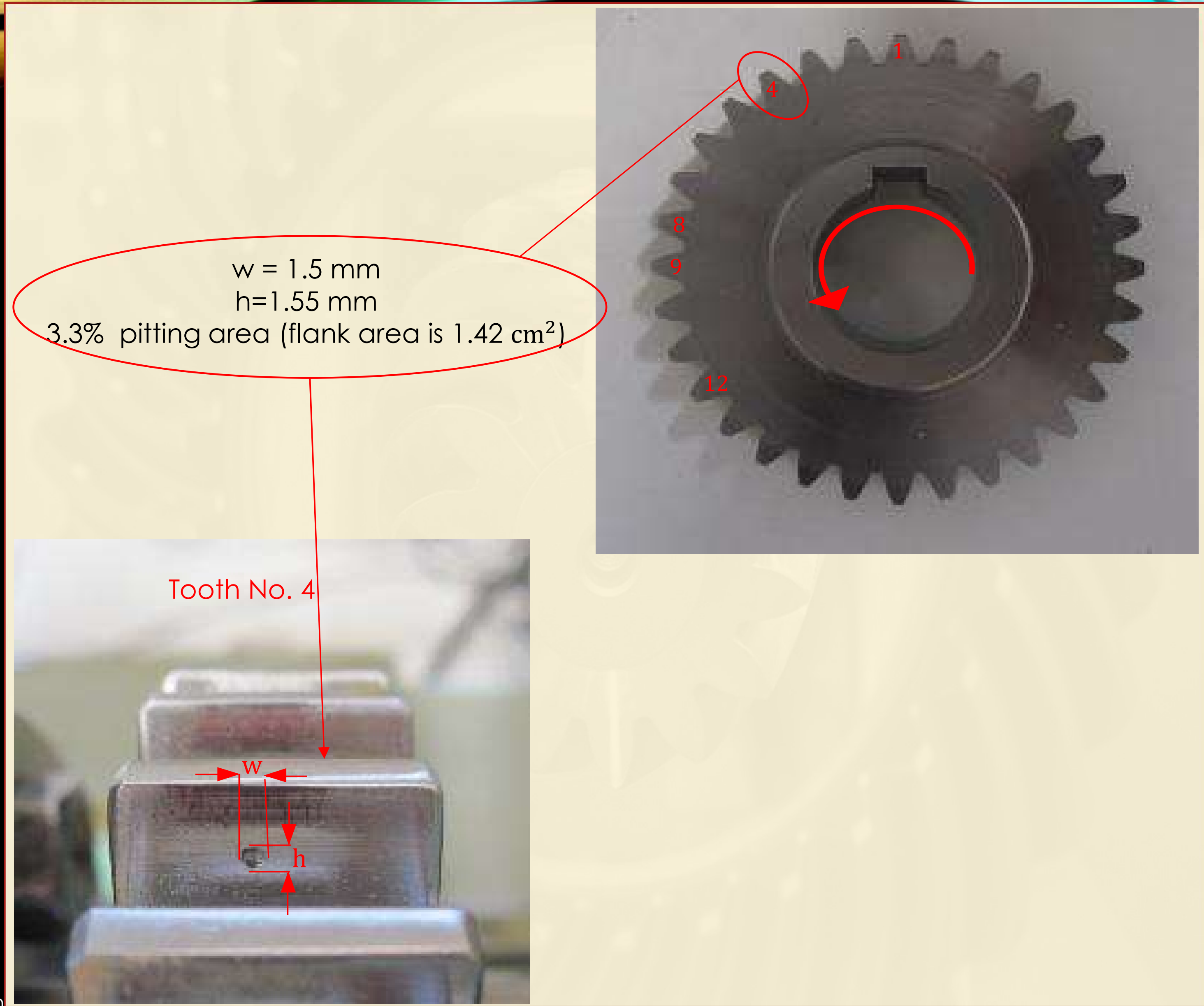
Four strain gauges are placed in the root fillet of each gear. The cables are connected to the slip ring through the hollow shaft



Collaboration:
cooperation with LUKA, INDGEAR industrial partner

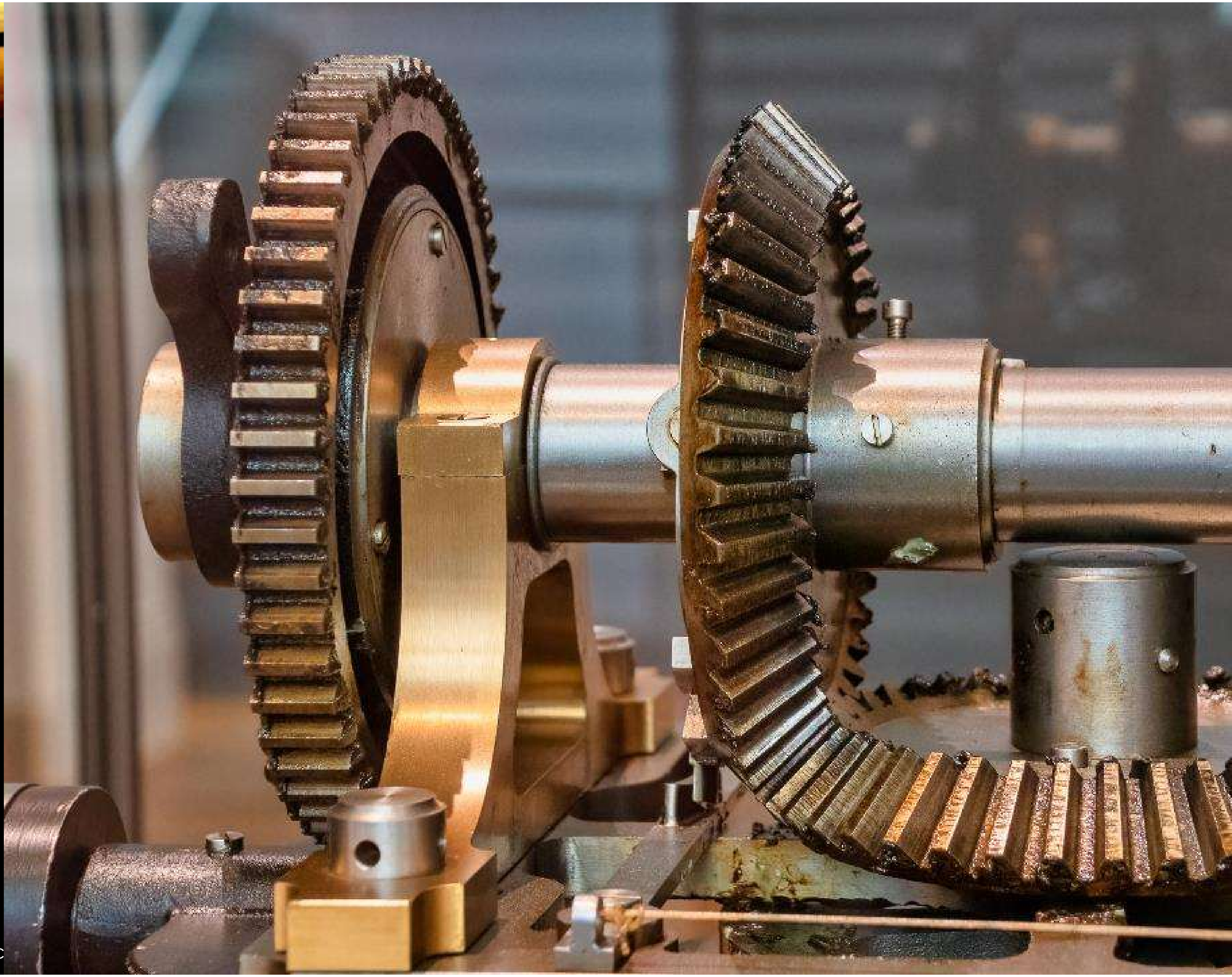
Goal:
Special design and production for fast pitting

Achievements:
- Accelerated endurance test (pitting)
- Vibration-stress dependencies
- Direct gear stress measurement



SOFTWARE

Simulations

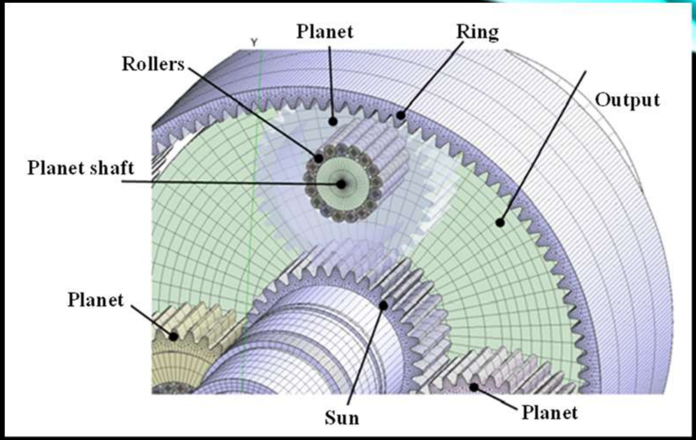


Type: Planetary gear

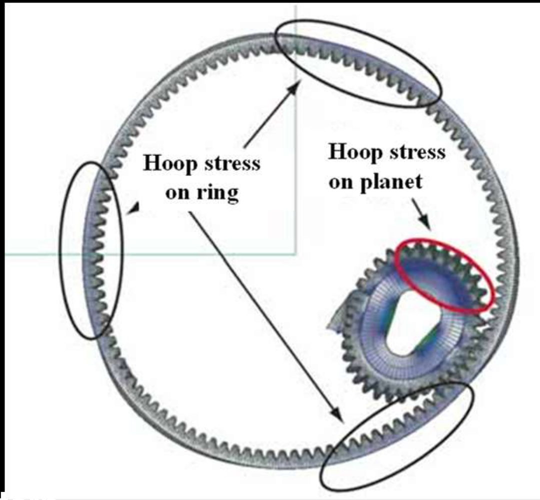
Goal:
FEM analysis of planetary gearbox for Robotics

Software:
ANSOL – Transmission3D

Detailed FEM modelling
- Housing, carrier and shaft elastic deformation
- Gears
  - Contact analysis
  - Misalignments
- Bearings
  - Cage deformation
  - Rollers: deformation and contact analysis



Hoop stresses are generally neglected (or unknown)
- Arise when the ring is thin
  - lightweight gearboxes
  - e.g., helicopter final reduction
- Can cause unexpected failures



The whole elasticity influences the contact pattern
- The carrier and shaft deformation induce planets misalignments
- Local stresses depend on:
  - load
  - teeth geometry (crowning)
  - global elasticity

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Type: Helical gear pair

Goal:

Hardness effects on gear-pair lifetime:

- Depth of hardening,
- Core hardness,
- Surface hardness.

Considering safety factors:

- root safety factor, ISO 6336-3,
- flank safety factor, ISO 6336-2,
- safety factor of hardened layer, DNV 41.2,
- safety factor against tooth flank fracture, ISO 6336-4,
- safety factor against scuffing, ISO 13989.

Software:

KISSsoft

Table 1. Effect of surface hardness on safety factors for gear-pair No.1

Depth [mm] =2						Core hardness [HRC] =34.6					
Surface hardness	HRC	52.4		54.4		56.4		58.4		60.4	
Safety factors Against tooth flank fracture		1.432	1.450	1.458	1.477	1.481	1.500	1.502	1.521	1.520	1.539

Table 2. Effect of core hardness on safety factors for gear-pair No.1

Depth [mm] =3							Surface hardness [HRC] =61				
Core hardness	HRC	29.8		40.8		49.1		55.2		56.3	
Life time [h]		20,000		20,000		20,000		20,000		20,000	
Safety factors	Root	2.406	2.221	2.406	2.221	2.406	2.221	2.406	2.221	2.406	2.221
	Flank	1.481	1.574	1.481	1.574	1.481	1.574	1.481	1.574	1.481	1.574
	hardened layer	1.948		1.948		1.948		1.948		1.948	
	Against tooth flank fracture	1.821	1.842	2.069	2.090	2.125	2.147	1.515	1.534	1.429	1.447
	Against scuffing	3.609		3.609		3.609		3.609		3.609	

Table 3. Effect of hardness depth on safety factors for gear-pair No.1

Core hardness [HRC] =34.6							Surface hardness [HRC] =52.4				
Depth [mm]		2-2.1		2.5-2.6		3-3.1		3.5-3.6		4-4.1	
Life time [h]		20,000		20,000		20,000		20,000		20,000	
Safety factor	Root	2.406	2.221	2.406	2.221	2.406	2.221	2.406	2.221	2.406	2.221
	Flank	1.481	1.574	1.481	1.574	1.481	1.574	1.481	1.574	1.481	1.574
	hardened layer	1.696		1.826		1.948		2.064		2.173	
	Against tooth flank fracture	1.432	1.450	1.586	1.606	1.774	1.794	1.727	1.747	1.892	1.913
	Against scuffing	3.609		3.609		3.609		3.609		3.609	

Type: Helical gear pair

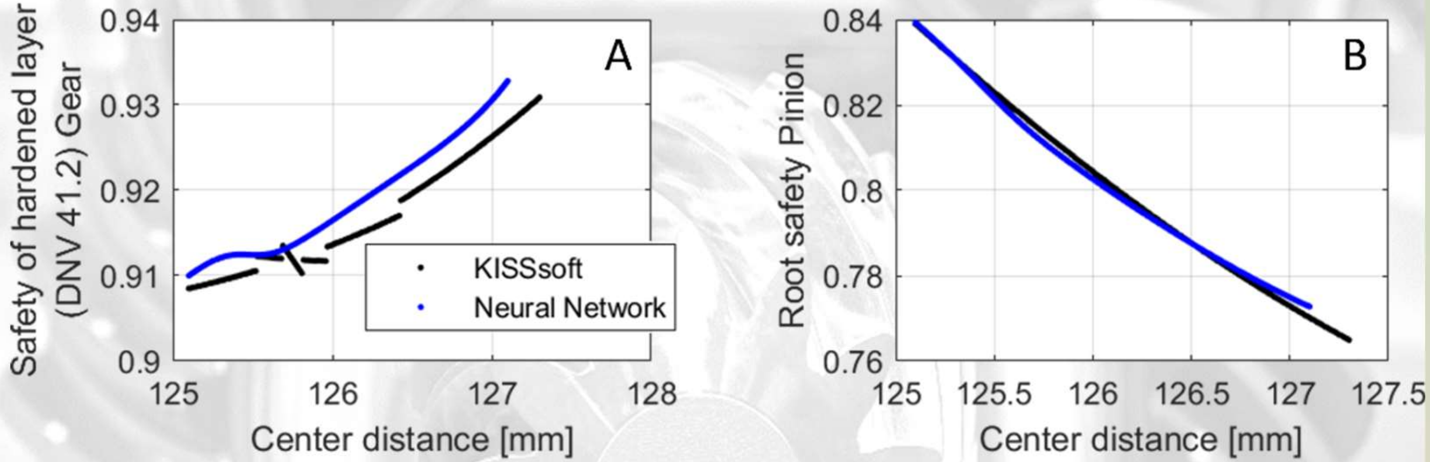
Goal:

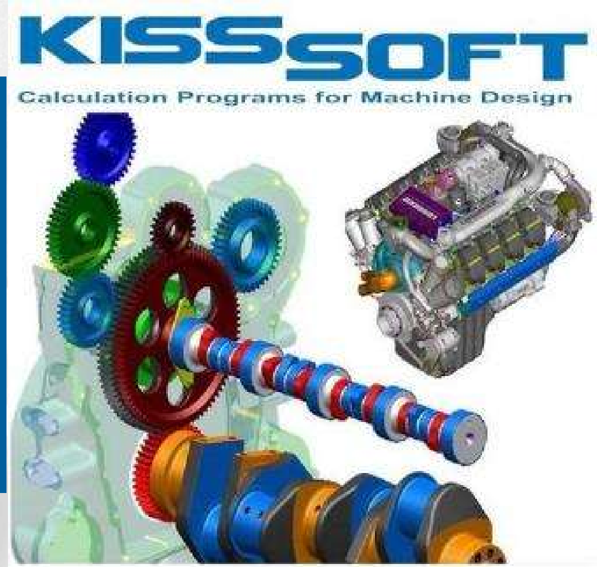
Train the algorithm to predict the life-time

Software:

MATLAB and KISSsoft

Neural Network Algorithm





Type: Spiral Bevel Gear

Goal:

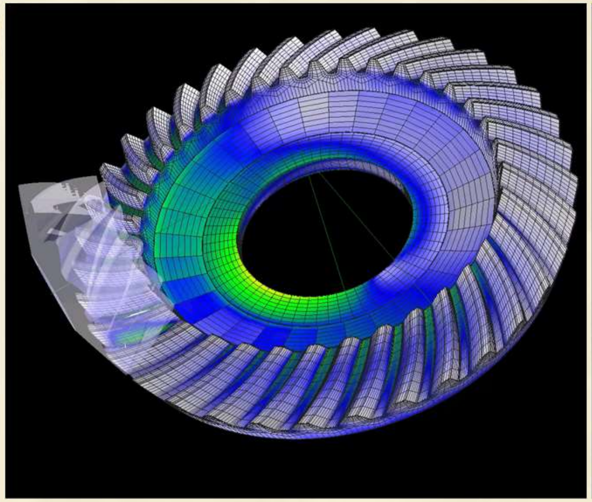
- Loaded and Unloaded tooth contact analysis (LTCA and UTCA)
- Mesh stiffness calculation

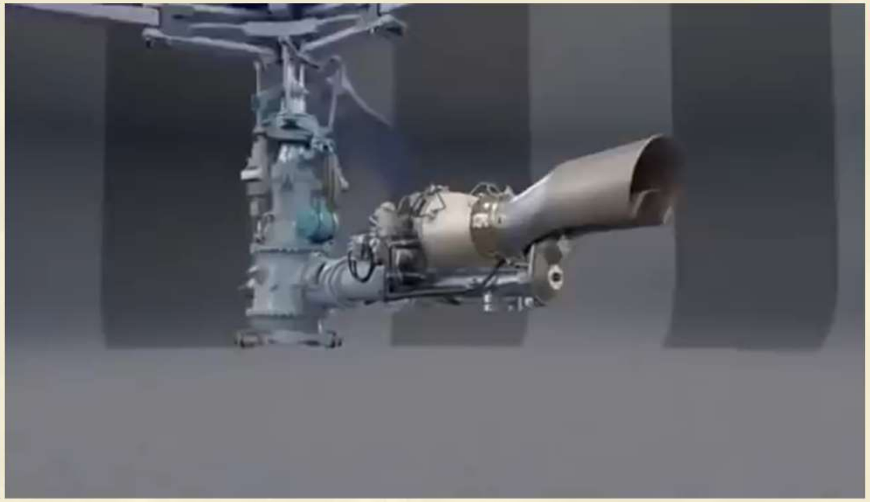
Application:

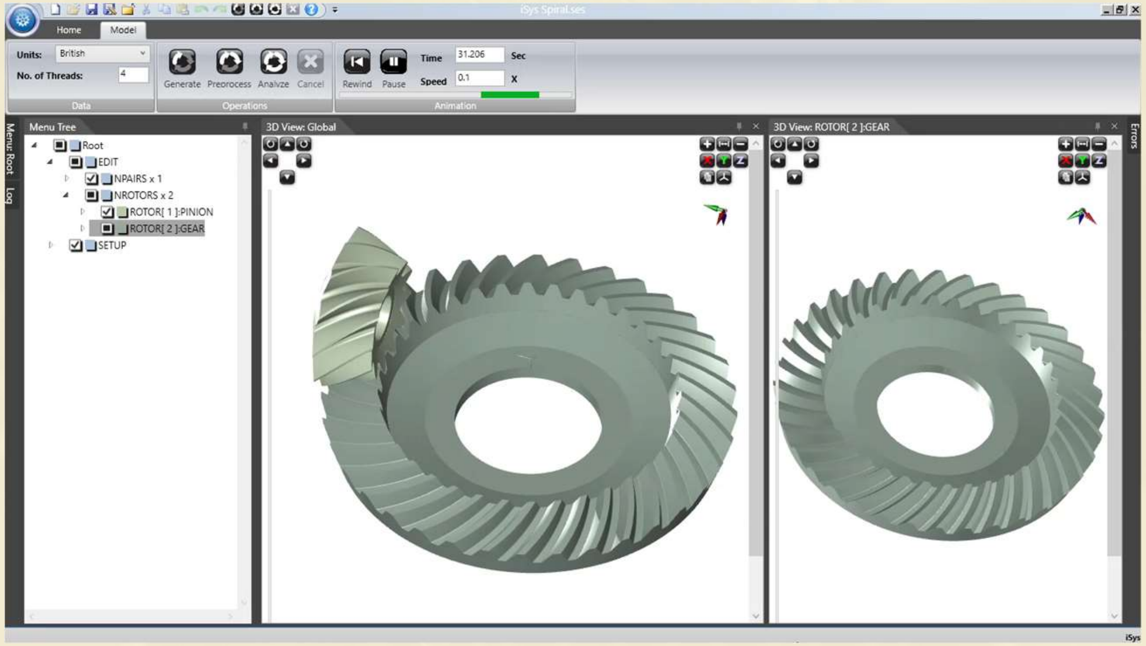
Helicopter transmission system

Software:

ANSOL – Transmission3D





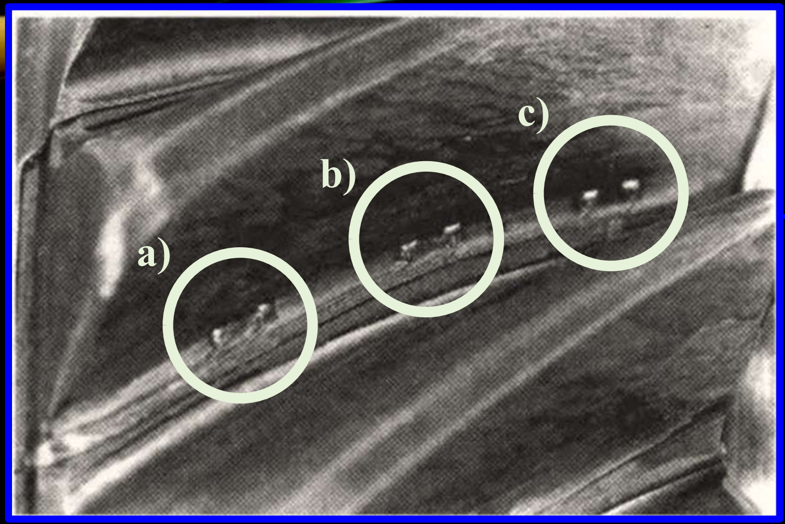


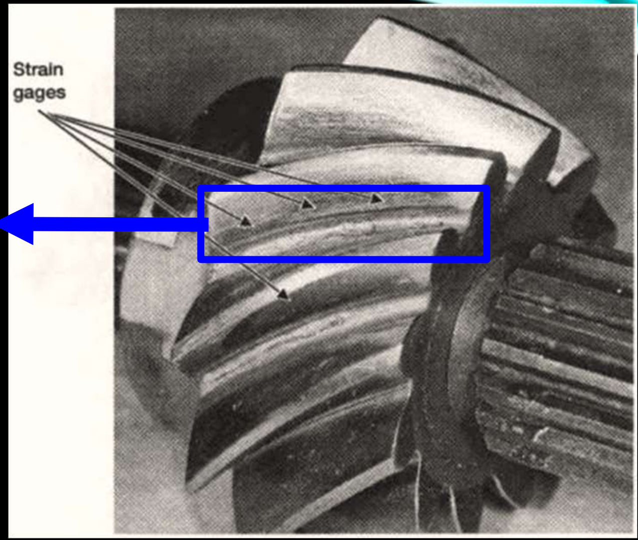
Type: Spiral Bevel Gear

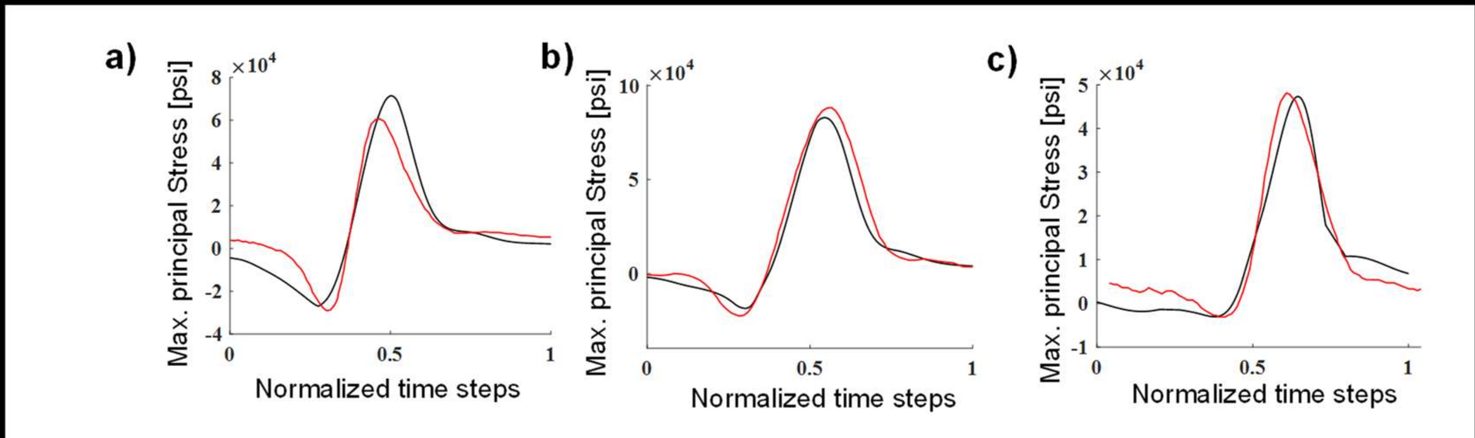
Goal:
Part 1:
Validation numerical results

Comparison with:
Experimental data, done by NASA group.

Comparison parameters:
Max. principal stress on the root of pinion tooth

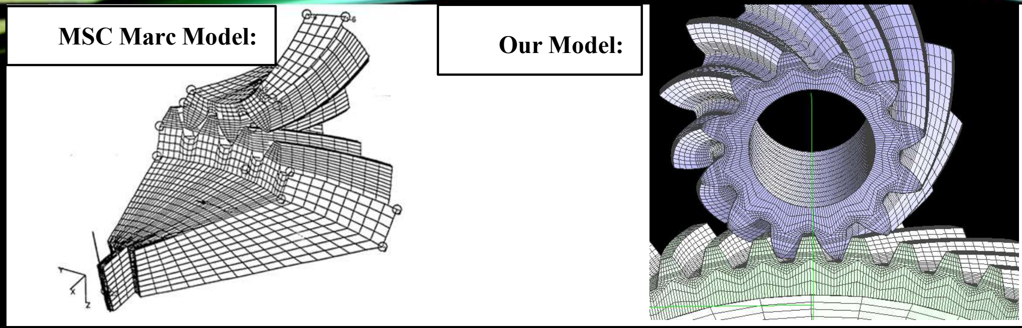






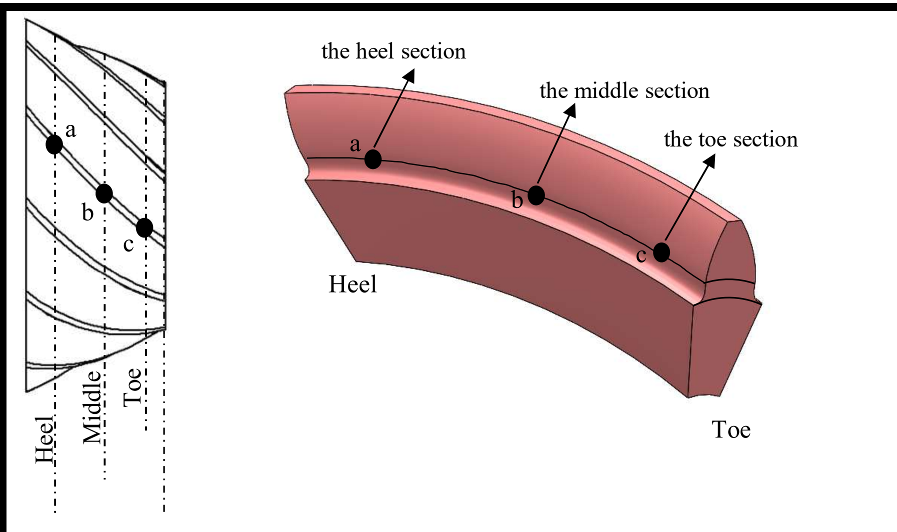
Maximum principal stress on the root of the tooth,
— FEM (Calyx) simulation, — Experimental results.

Type: Spiral Bevel Gear



Goal:

Part 2:
Validation numerical results

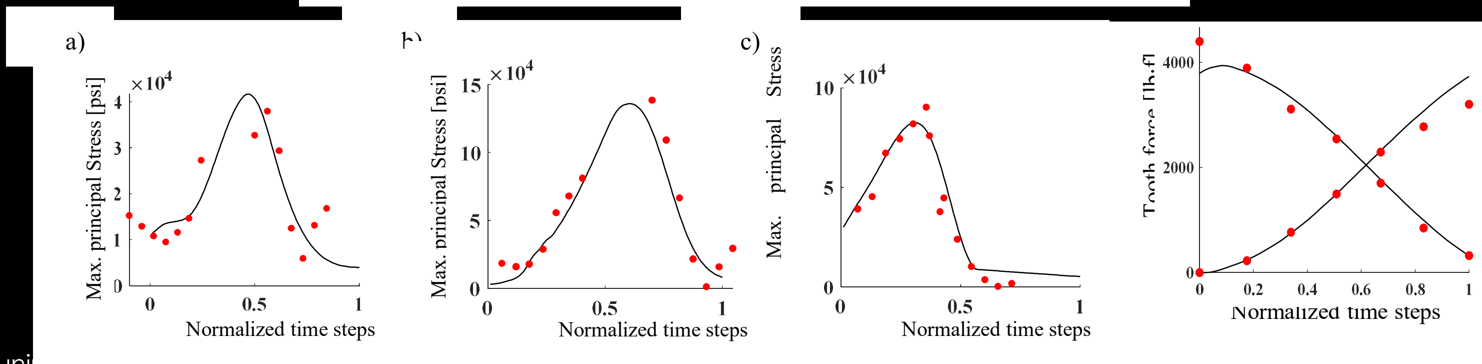


Comparison with:

Numerical results, FEM (MSC Marc); done by NASA group.

Comparison parameters:

- Max. principal stress on the root of pinion tooth
- Force between two mated teeth



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Type: Spiral Bevel Gear

Goal:

FEM grid sensitivity analysis and find the suitable mesh size

Considered parameter:

Average rotational deflection of gear

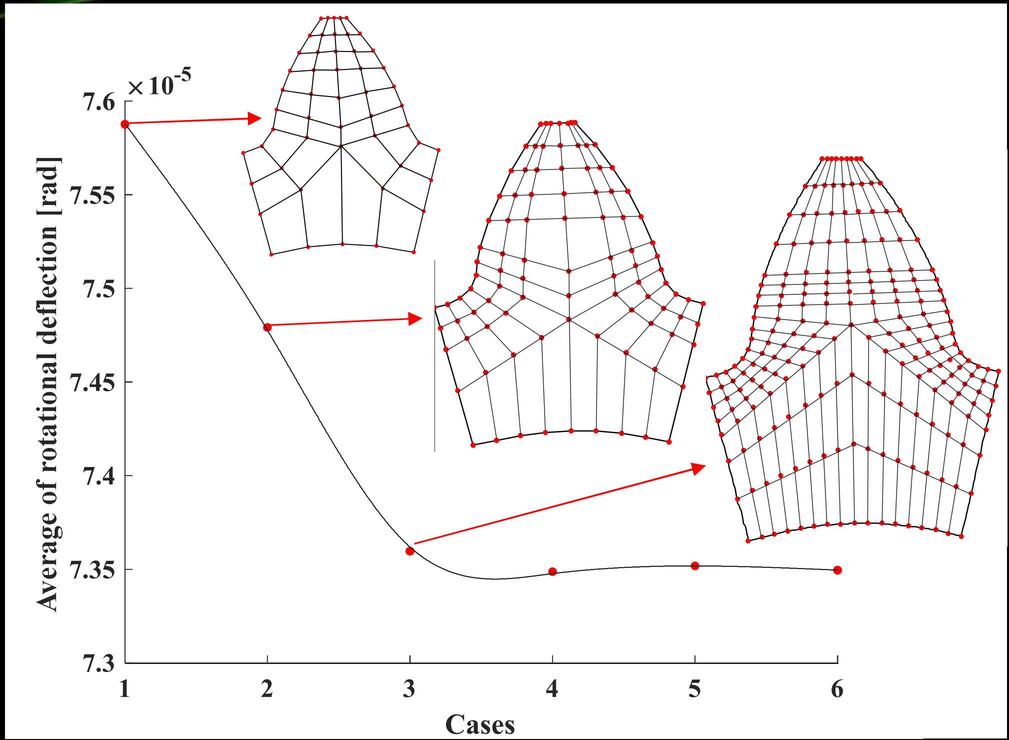

Table 1. Grid sensitivity analysis

Case	1	2	3	4	5	6
Average deflection of gear [µrad]	75.86	74.77	73.60	73.49	73.52	73.49
Number of sections along the tooth width	3	8	13	14	15	16
Number of elements on the tooth section	36	74	176	176	176	176
Computation time [minute]	260	286	398	420	437	454

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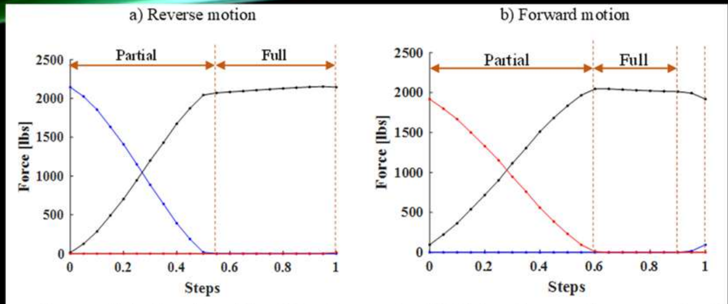
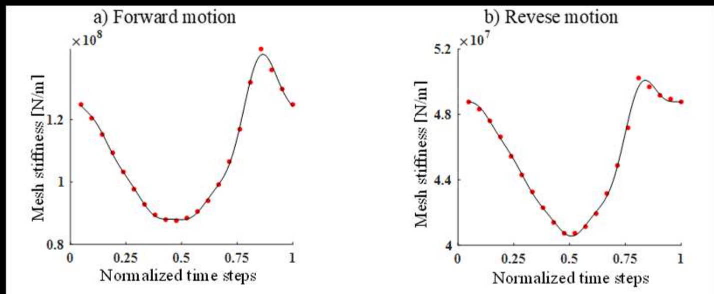
Type: Spiral Bevel Gear



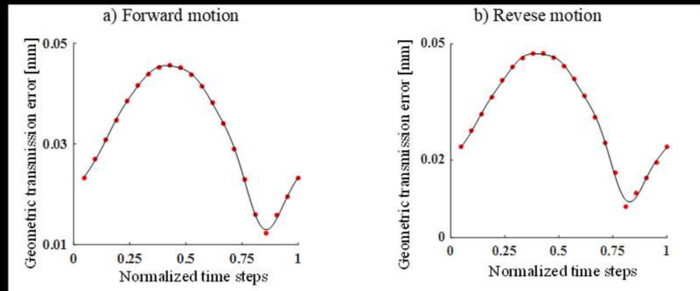
Force on the tooth surface of the pinion - torque value 4754 lb.in on the gear; a) reverse motion, b) forward motion; tooth No. 1 (red line), tooth No. 2 (black line), and tooth No. 3 (blue line)

Goal:

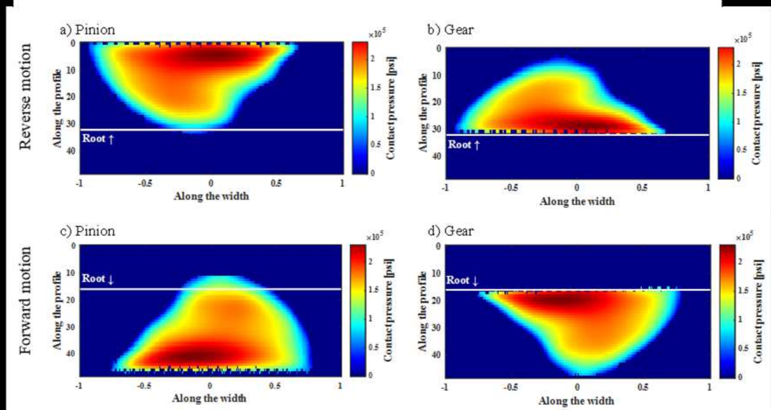
- Force on the tooth surface of the pinion
- Contact pressure on the tooth surface
- Mesh stiffness diagram
- Geometric transmission error



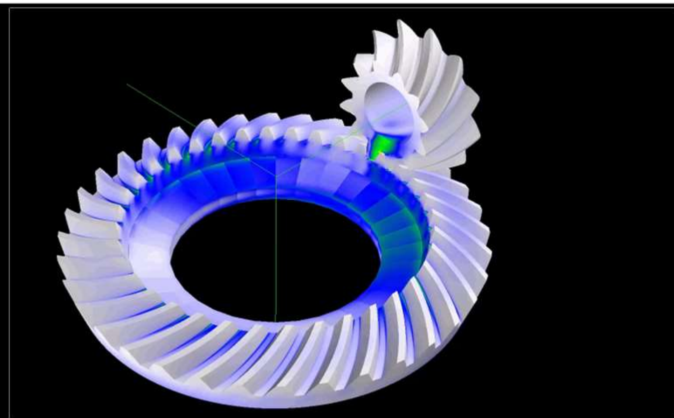
Mesh stiffness diagram:
a) forward motion, b) reverse motion; — Fitted curve by Fourier series, ● FEM results.



Geometric transmission error on the line of action:
a) forward motion, b) reverse motion; — Fitted curve by Fourier series, ● FEM results.



Contact pressure on the tooth surface - torque value is 4754 lb.in on the gear;



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SBG: Publication

Authors:
Moslem Molaie, Farhad S. Samani, Antonio Zippo, Francesco Pellicano.

Title:
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Journal:
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Spiral Bevel Gears: nonlinear dynamic model based on accurate static stiffness evaluation

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ARTICLE INFO

Keywords:
nonlinear vibration
spiral bevel gear
gear mesh stiffness
forward and reverse motions
backside contact

SCAN ME

ABSTRACT

In the present paper non-linear dynamics of a spiral bevel gear pair with backlash are investigated in order to clarify the internal excitations of major importance from the vibration point of view: manufacturing errors in the teeth profile, teeth spacing errors, and elastic deformation of the teeth. In some conditions, like in the case of backside contact, the destructive effect of internal excitations can be intensified leading to complex dynamics; for such reasons here backside contacts and reverse rotation are investigated in detail using a nonlinear time-varying model. The effect of damping is investigated as well. A one-DOF model is developed in order to study the dynamic behavior; the resulting a nonlinear differential equation with time-varying mesh stiffness is solved via numerical integration based on an adaptive step-size implicit Runge-Kutta scheme. The dynamic response of the system is analyzed through time histories, phase portraits, bifurcation diagrams, and Poincaré maps. Results show that for small backlash values, the possibility of backside contact increases. Meanwhile, by increasing the backlash value, the amplitude vibration of the gear rotation rises as well. By comparing the dynamic response of the system with different damping ratios, the results show that higher damping effectively reduces gear vibration resonance, although the probability of unsteady response still exists.

THE VIBRATION AND POWERTRAIN LAB

Not only Gears

- NVH testing: vibration and noise measurements, modal analysis, materials analysis
- Mechanical and NVH modelling: FEA and Multibody
- Environmental and mechanical tests on shaking table
- Active and passive vibration control
- Problem solving

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RETE ALTA TECNOLOGIA EMILIA-ROMAGNA HIGH TECHNOLOGY NETWORK
TECNOPOLO MODENA

THANK YOU FOR YOUR ATTENTION

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