



PhD Final Hearings XXXIII Cycle

Torino, 17th Dec. 2020



PhD Thesis

Power transmission systems: from mechanical to magnetic gearboxes

PhD funded by: Interdepartmental Centre for Automotive
Research and Sustainable mobility (CARS@PoliTO)

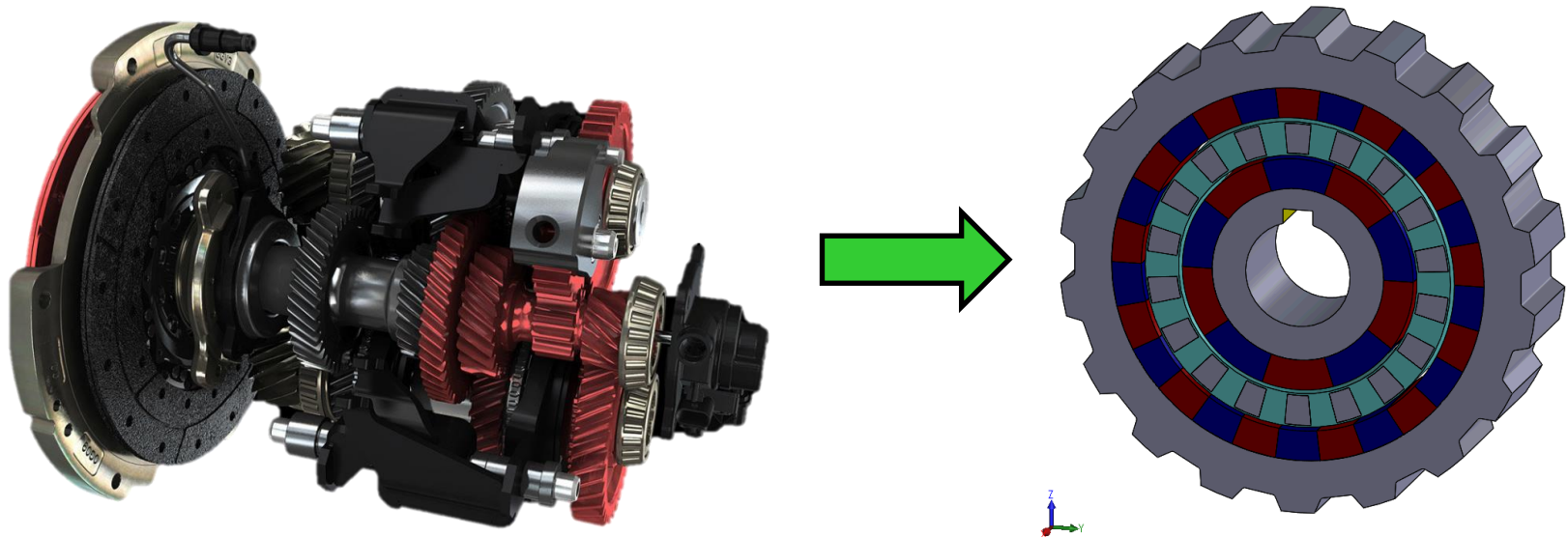
Supervisor:

Prof. Elvio BONISOLI

PhD Candidate:

Luca DIMAURO

Aim: design and analysis of a magnetic gearbox prototype starting from issues of traditional mechanical gearbox



How: using a methodology for the objective evaluation of gear shift induced vibration in a mechanical transmission and magneto/dynamic simulations of magnetic transmission with experimental validation

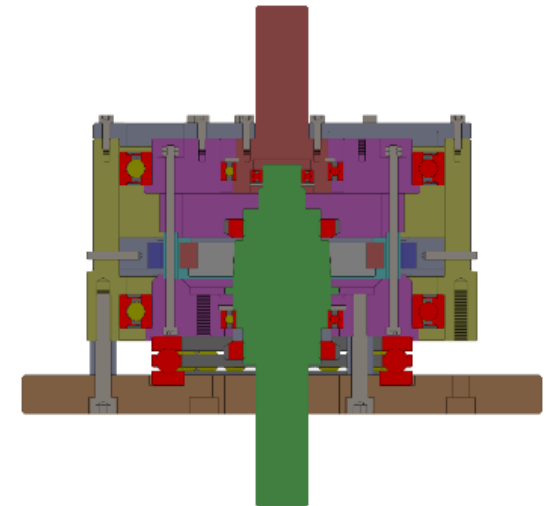
Mechanical Transmissions:

- Introduction to transmission noise issues
- Methodology for gearshift vibration assessment in automotive transmission
- Simulation results and possible improvements



Magnetic Transmissions:

- Advantages and drawbacks of magnetic gears
- Design, optimisation, dynamics and simulations of a PMG for powertrain applications
- Magnetic gearbox solution
- Conclusions and future developments



Typical transmission noise



Type

Reason

Shudder

Torsional oscillations during **clutch** engagement process influenced by **frictional characteristics**.
The frequency range is $5 \div 20$ Hz.

Whine

It is an **acoustic** issue due to transmission error and to variation of gears **meshing stiffness**.
The frequency range is $2 \div 15$ kHz.

Clunk

Metallic noise under **tip-in/out** condition due to impacts in gears teeth when engine torque is suddenly applied. Broad band frequency range.

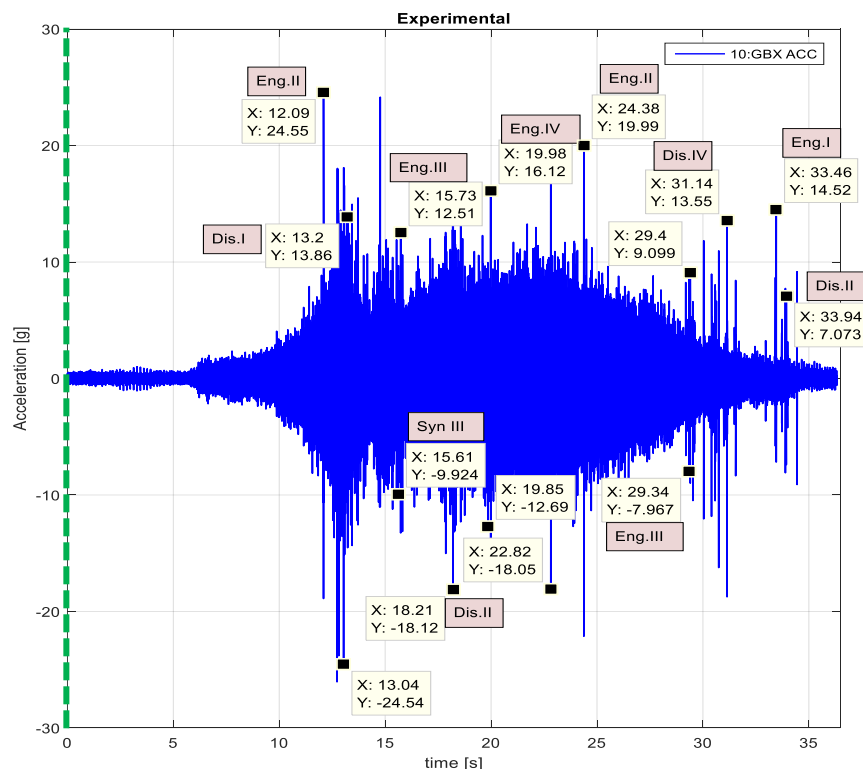
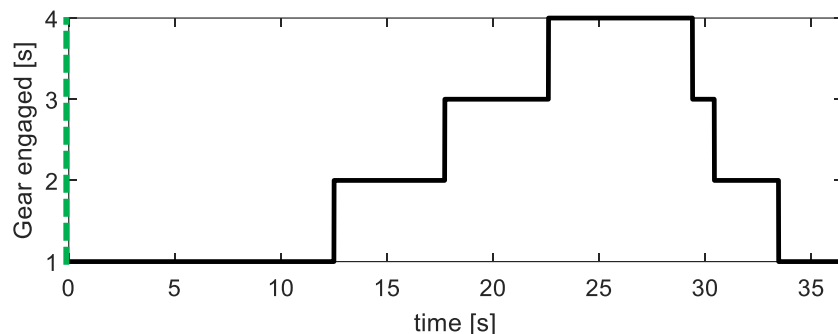
Rattle

Torsional vibrations due to the **backlashes** between **unloaded** gear pairs. Gear rattle intensity is directly related to **engine angular acceleration**.

Gear-shift

It is due to axial and rotational impacts between teeth flanks of different **synchronizer elements** during **gear engagement**.

Gearshift events producing acceleration peaks



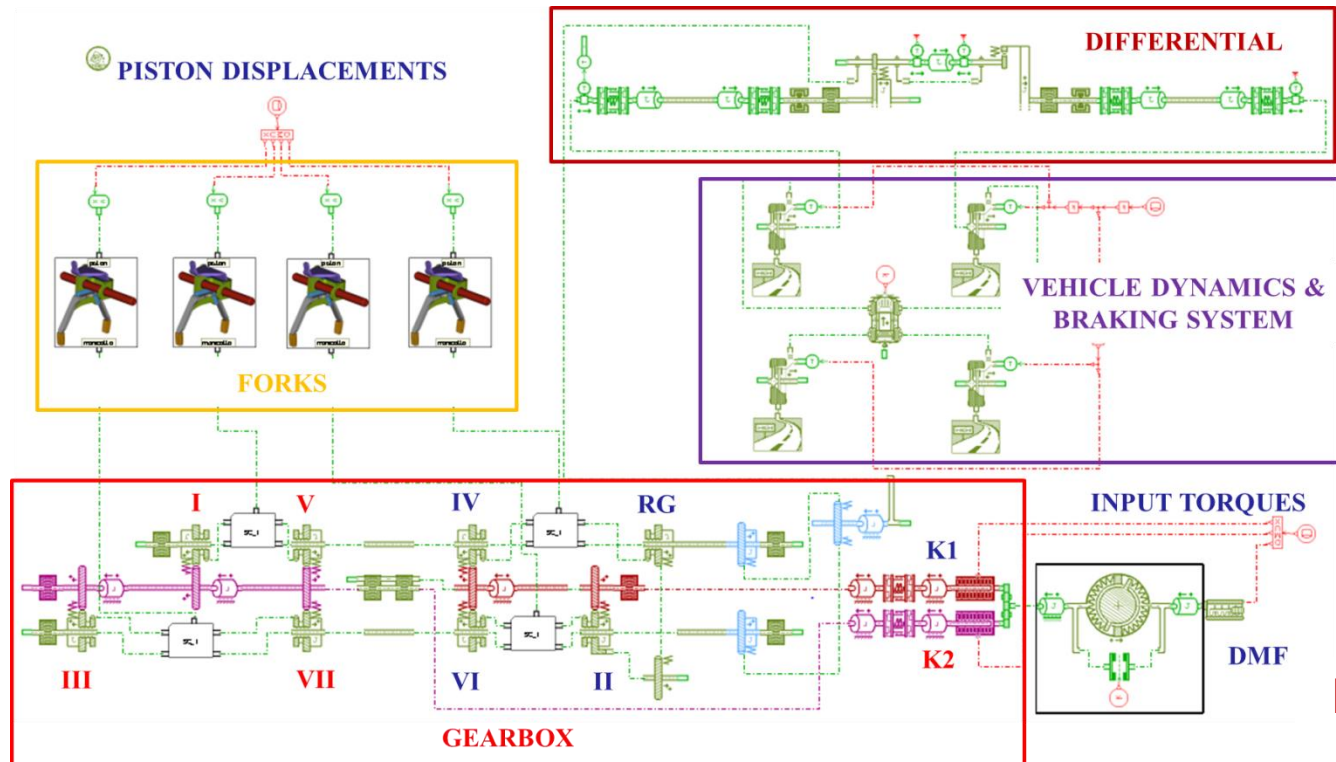
Higher acceleration peaks due to:

- Gear engagement
- Gear disengagement

Others acceleration peaks due to:

- Oncoming clutch start of engagement during cross-shift
- Outgoing clutch end of engagement during cross-shift
- Zero-crossing of torque on an active (engaged) transmission path (kiss/touch point).

Detailed transmission model layout

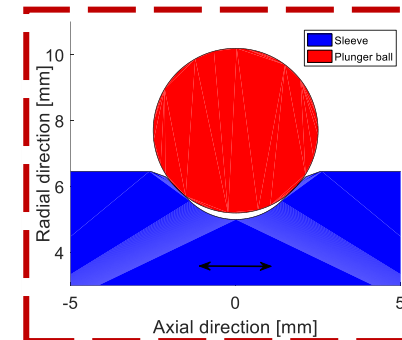
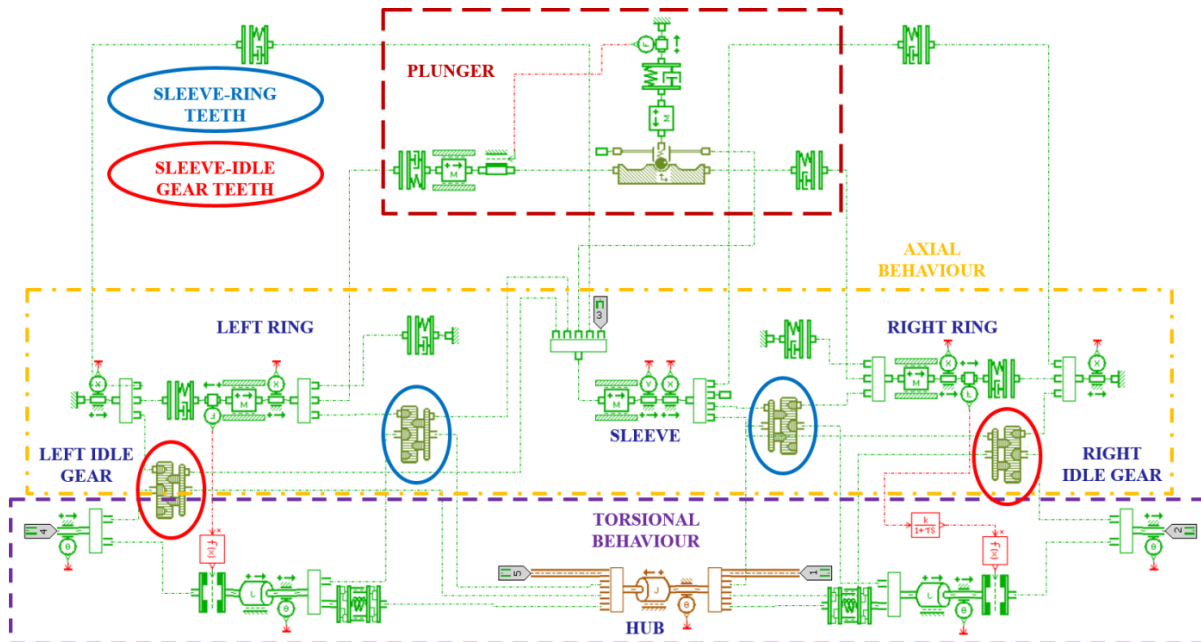


**More than
300 DoFs**

Model requirements:

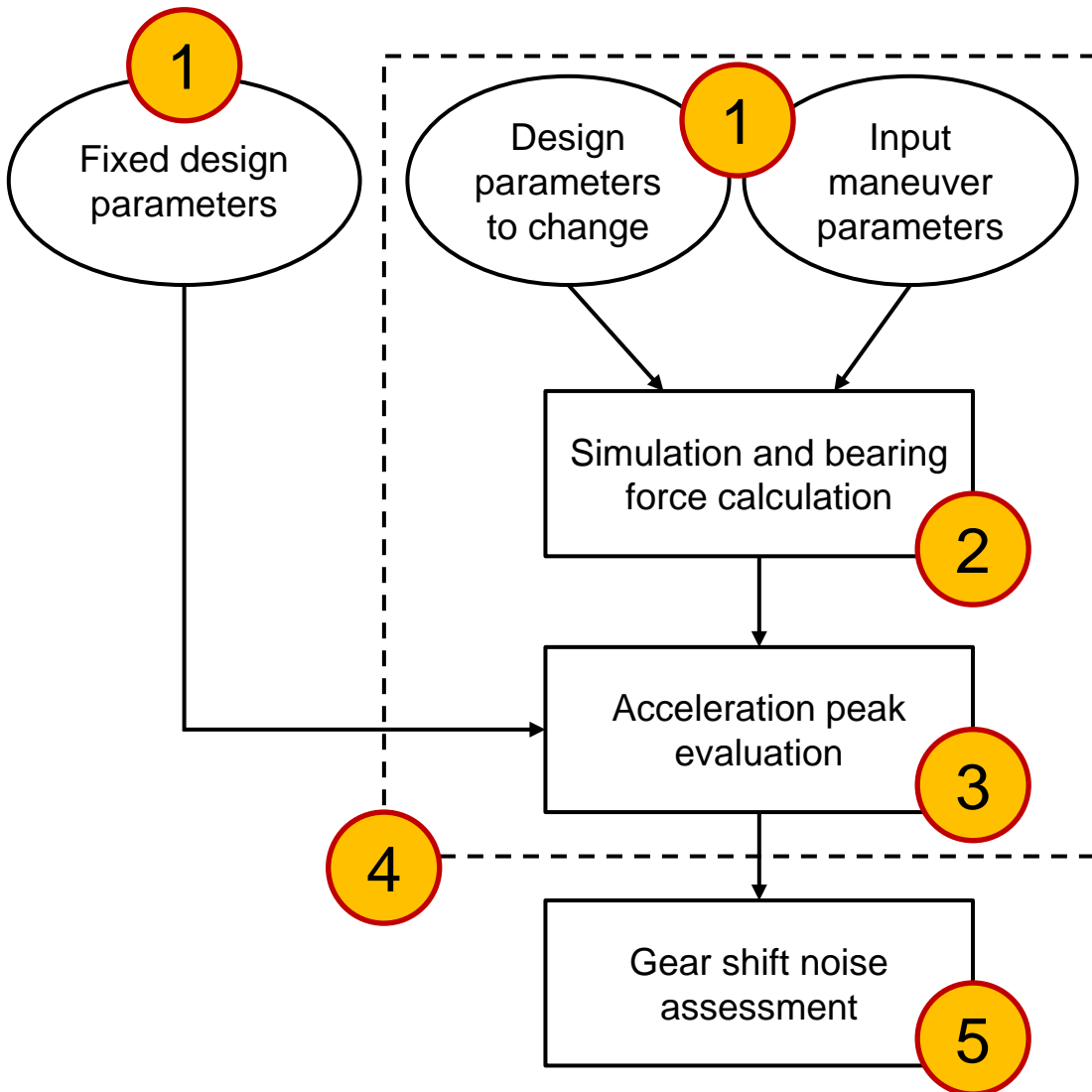
- Capability to predict the synchronization dynamics
- Model inputs are the gear shift forks displacements, engine torque, clutches transmissible torques, and pressure of the breaking system
- Used to simulate manoeuvres with one or two synchronizers engaged at the same time

Synchronizer customised model



Characteristics:

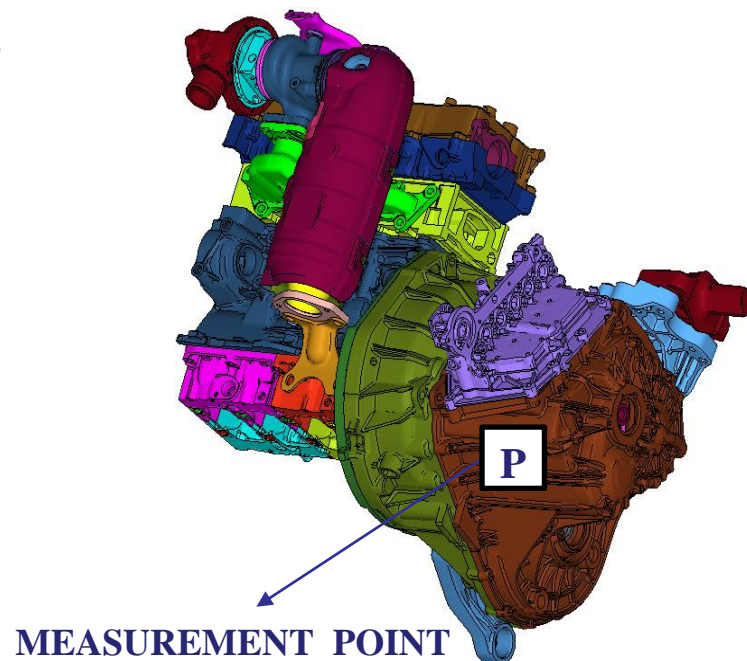
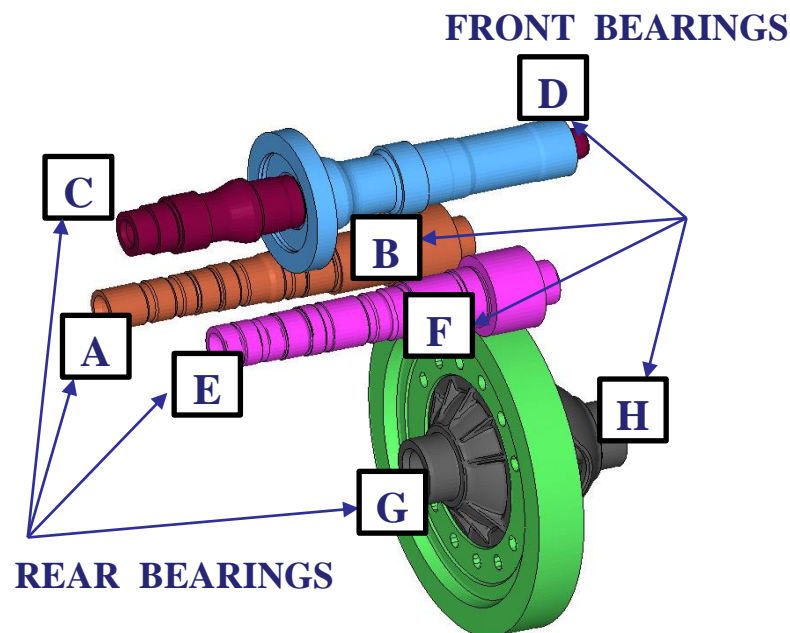
- Synchronizer model considers both **torsional** and **axial behaviour** of its inner elements. It describes, with high level of detail, the dynamics during **synchronization phase** and **gear engagement**, source of the **gearshift noise**. The following DOFs with corresponding backlashes are assumed for each synchronizer:
 - Axial displacement of sleeve (1);
 - Radial and axial displacements of the plunger ball (2);
 - Axial and rotational displacements of left and right rings (4);
 - Rotation of the two idle gears (2)



Steps for NVH assessment :

1. Define the transmission design parameters and the maneuver to be tested
2. Simulation: evaluate whole transmission inner dynamics and bearing force
3. Calculate the gearbox housing acceleration time history
4. Repeat steps 1, 2 and 3 for different values of the design parameter
5. Perform comparative gear-shift noise assessment using indices based on gearbox acceleration

Gearbox housing acceleration computation

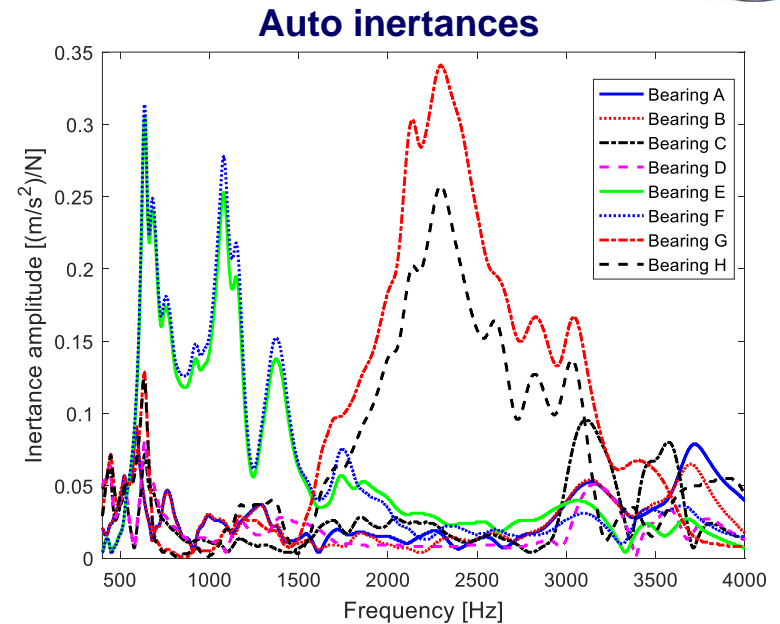


Assumptions:

- Gearbox is considered a **MDOFs** linear system, with **free-free boundary conditions**, to reproduce engine mounts
- The dynamics of the transmission and the gearbox **are decoupled**
- It is possible compute the acceleration of the gearbox at point (P) using a post-processing tool: the **bearing forces** are only **due to torsional dynamics**

Procedure:

- Export inertance FRF from FEA model data
- Perform the IFFT of the inertance FRF, i.e., get the impulse acceleration response of the system
- Perform time-domain convolution between the excitation and the impulse acceleration response



$$\ddot{h}(t) = IFFT\left(H_{j,k}(\omega)\right) = \frac{1}{2\pi} \int_{-\infty}^{+\infty} H_{j,k}(\omega) e^{i\omega t} d\omega$$

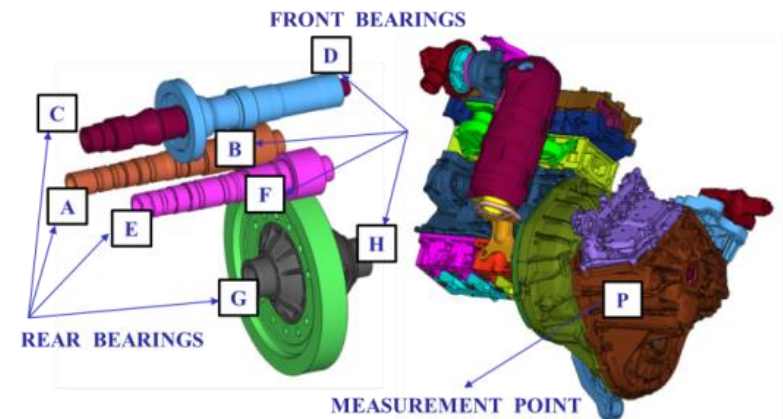
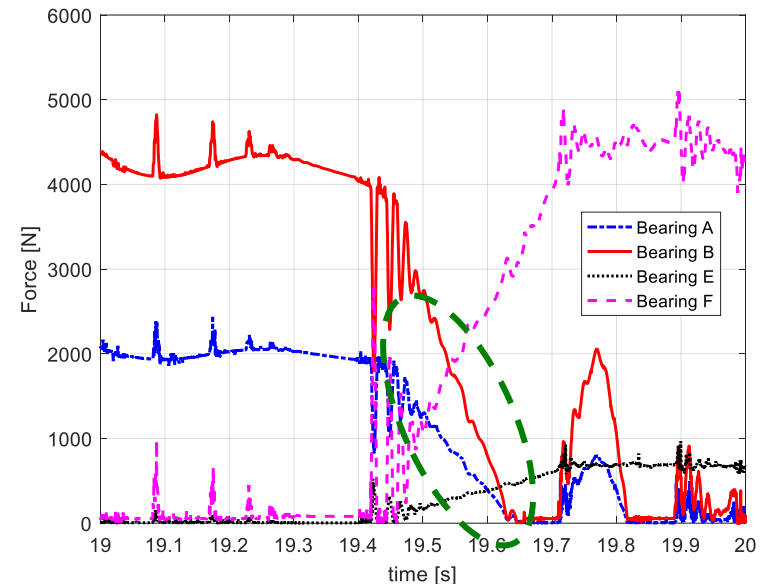


$$a(t) = \ddot{h}(t) * F(t) = \int_0^{+\infty} \ddot{h}(\tau) \cdot F(t - \tau) d\tau$$

Procedure:

- Export inertance FRF from FEA model data
- Perform the IFFT of the inertance FRF, i.e., get the impulse acceleration response of the system
- Perform time-domain convolution between the excitation and the impulse acceleration response
- Repeat the previous 2 steps for each inertance FRF between the single component of force (X,Y,Z) applied in all bearings (A - H) and the single component of acceleration (P)

Gearshift 1st to 2nd



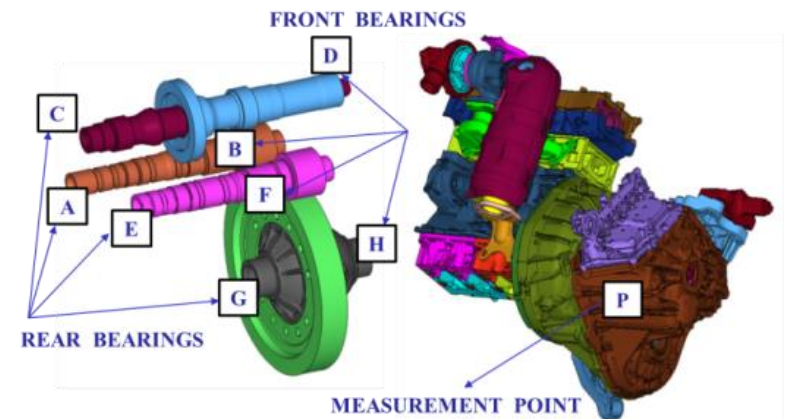
Gearbox housing acceleration computation



Procedure:

$$\begin{Bmatrix} a_X \\ a_Y \\ a_Z \end{Bmatrix} = \begin{bmatrix} a_X/F_{X_A} & \cdots & a_X/F_{X_H} & a_X/F_{Y_A} & \cdots & a_X/F_{Y_H} & a_X/F_{Z_A} & \cdots & a_X/F_{Z_H} \\ a_Y/F_{X_A} & \cdots & a_Y/F_{X_H} & a_Y/F_{Y_A} & \cdots & a_Y/F_{Y_H} & a_Y/F_{Z_A} & \cdots & a_Y/F_{Z_H} \\ a_Z/F_{X_A} & \cdots & a_Z/F_{X_H} & a_Z/F_{Y_A} & \cdots & a_Z/F_{Y_H} & a_Z/F_{Z_A} & \cdots & a_Z/F_{Z_H} \end{bmatrix} \begin{Bmatrix} F_{X_A} \\ \vdots \\ F_{X_H} \\ F_{Y_A} \\ \vdots \\ F_{Y_H} \\ F_{Z_A} \\ \vdots \\ F_{Z_H} \end{Bmatrix}$$

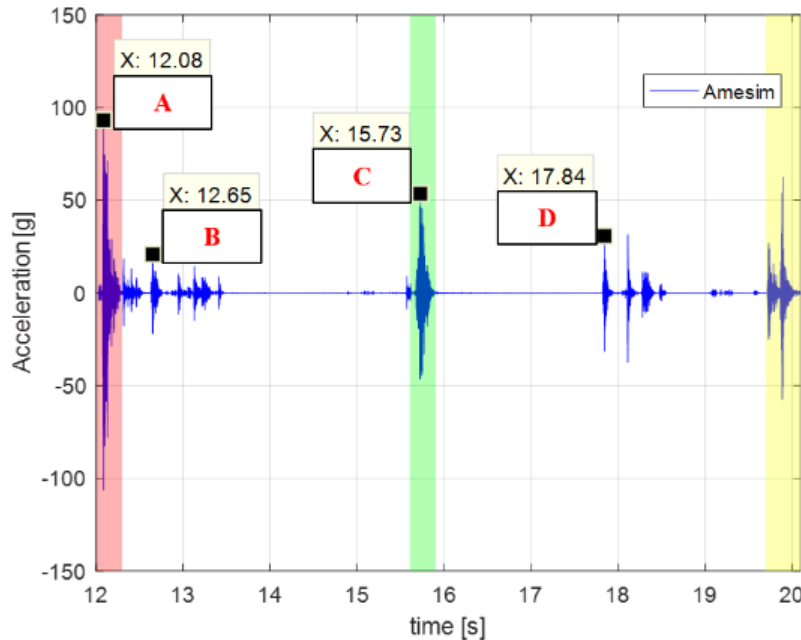
- Repeat the previous 2 steps for each **inertance FRF** between the single component of force (X,Y,Z) applied in all bearings (A - H) and the single component of acceleration (P)



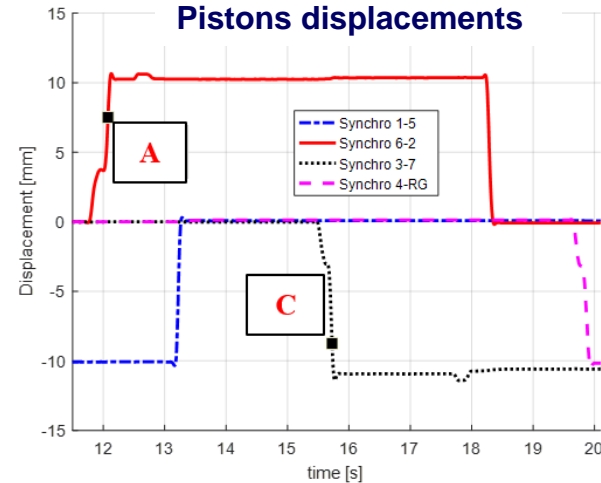
Typical simulation results



Acceleration



Pistons displacements

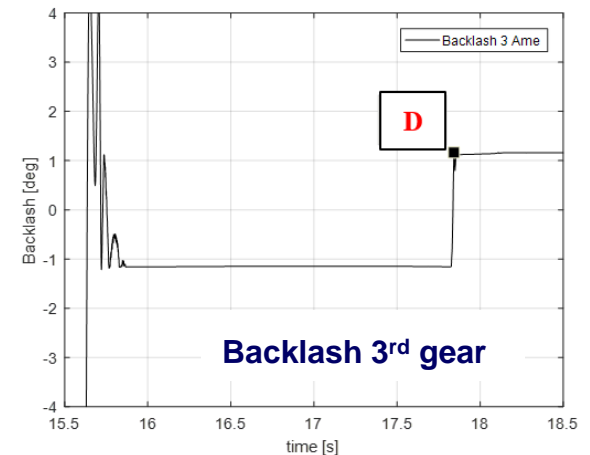
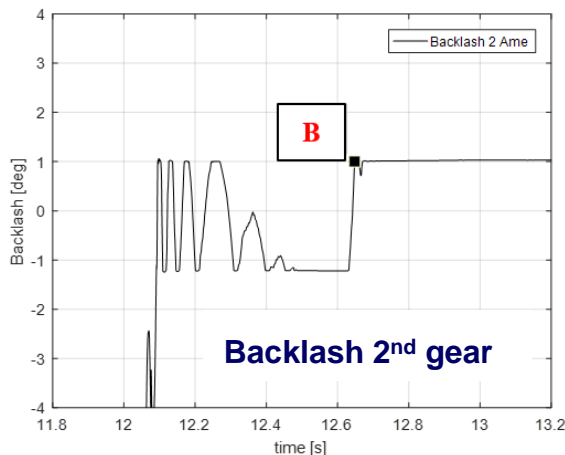


A and C due to:

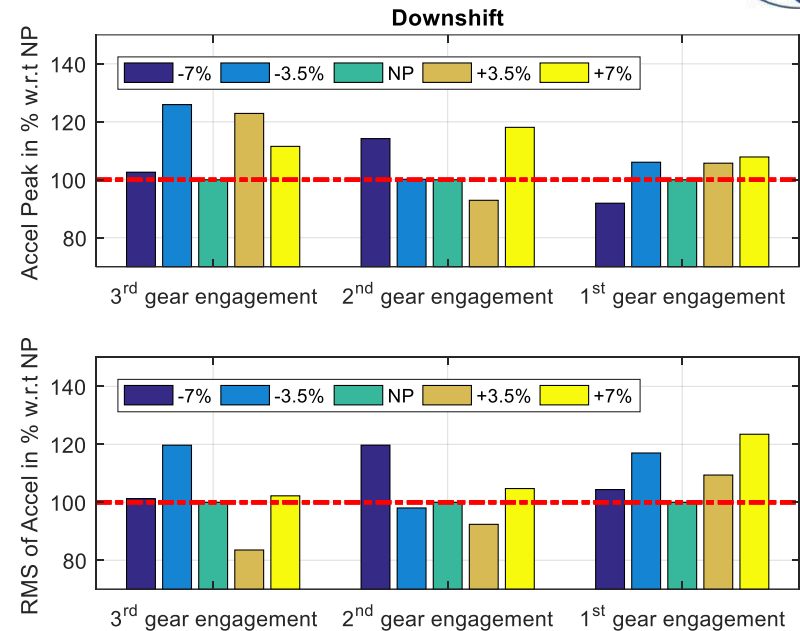
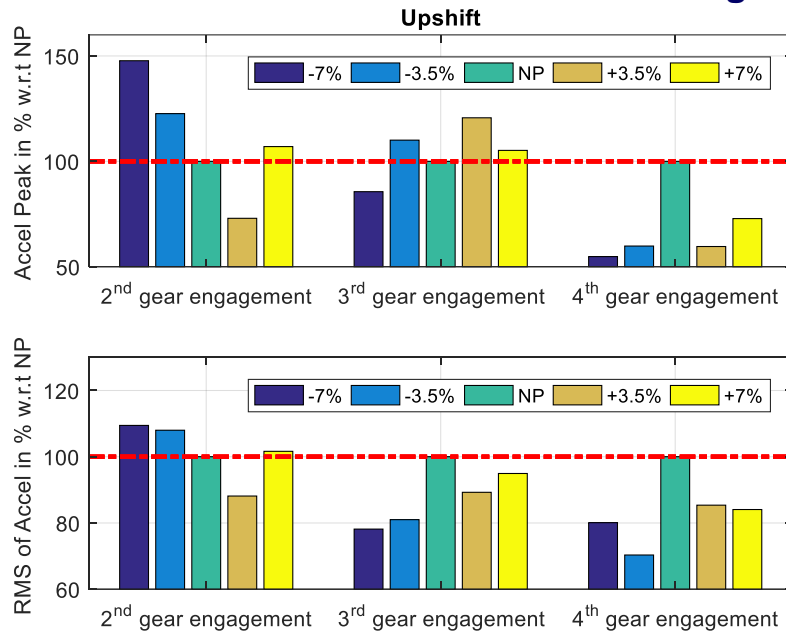
- Gear shift forks displacements during the gear engagement (**axial impacts**)

B and D due to:

- Recovery of the torsional backlashes in the synchronizer sleeve during the cross-shift phase, when a positive torque is applied (**torsional impacts**)



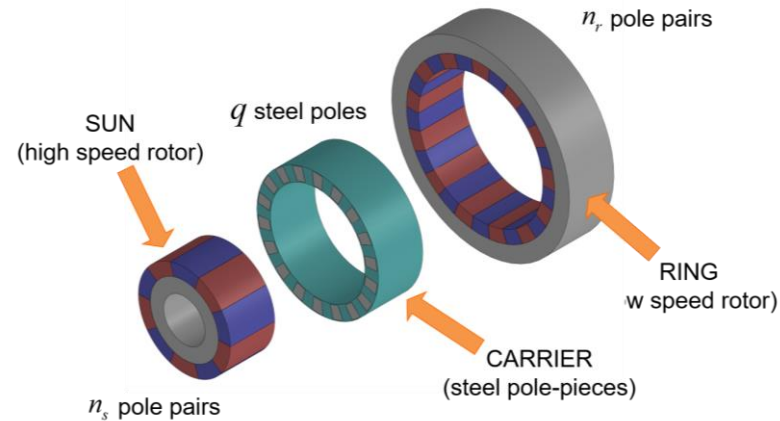
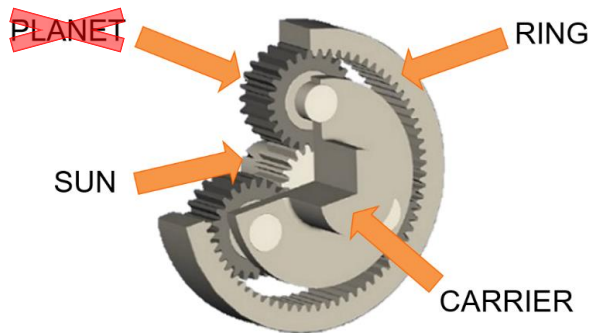
2nd idle gear inertia modification



Results:

- Two different indices are used to assess clunk severity: **peak to peak** amplitude and **RMS** of the gearbox housing acceleration
- Only the **higher peak** inside the colored are is considered for the first index
- Improvements for 1st to 2nd gearshift, that is the most critical, with an increase of inertia of 3.5% for the 2nd idle gear

Magnetic Gears: One Technology, Endless Possibilities



Mechanical Gears:

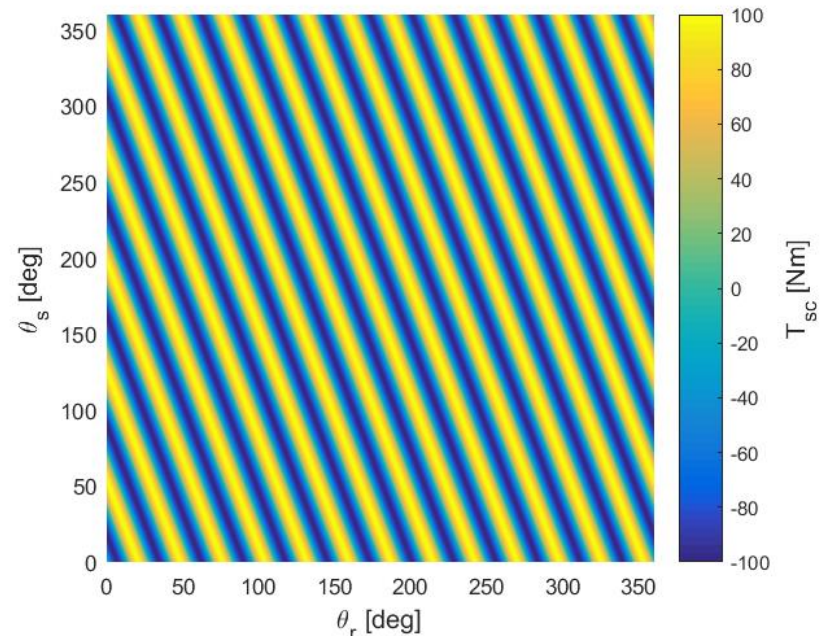
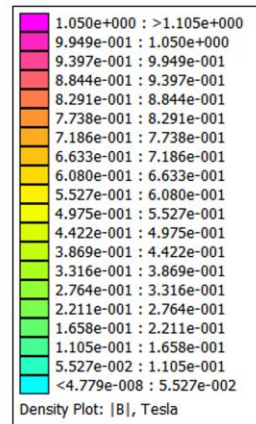
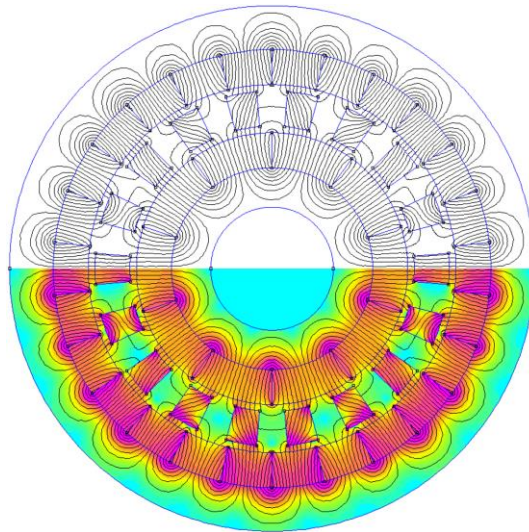
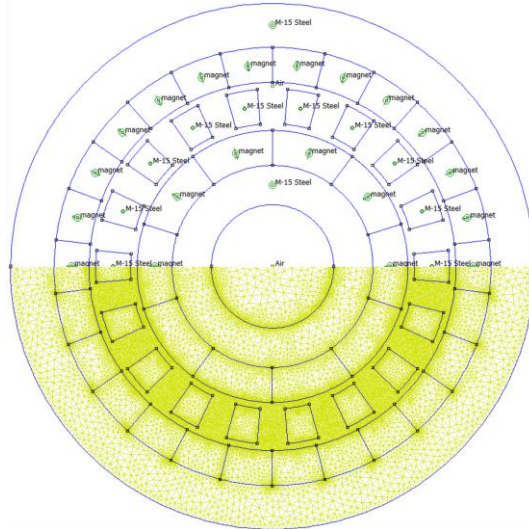
- They are affected by:
 - power losses for sliding friction
 - low efficiency and wear
 - periodic maintenance
- Critical effect on machine reliability
- Complementary devices:
 - to limit torque transmission
 - to ensure gearshifts (use of clutch in the driveline)

Magnetic Gears:

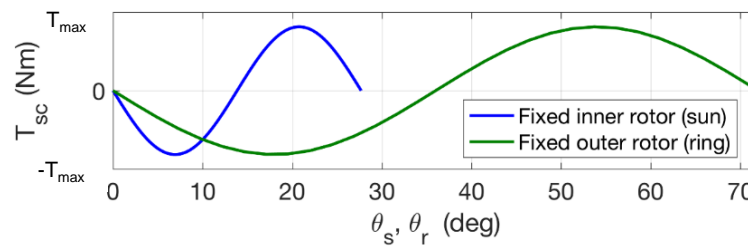
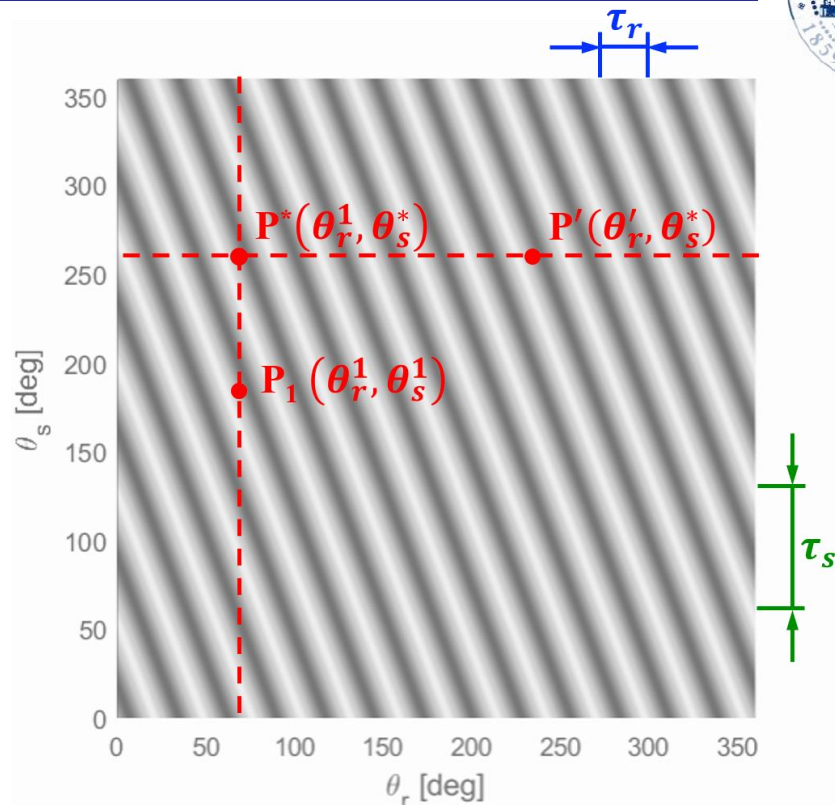
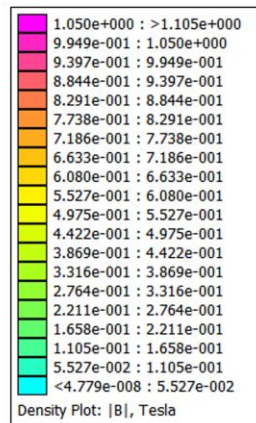
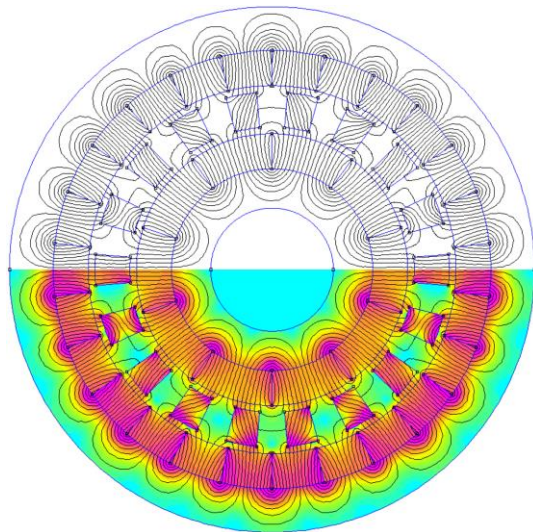
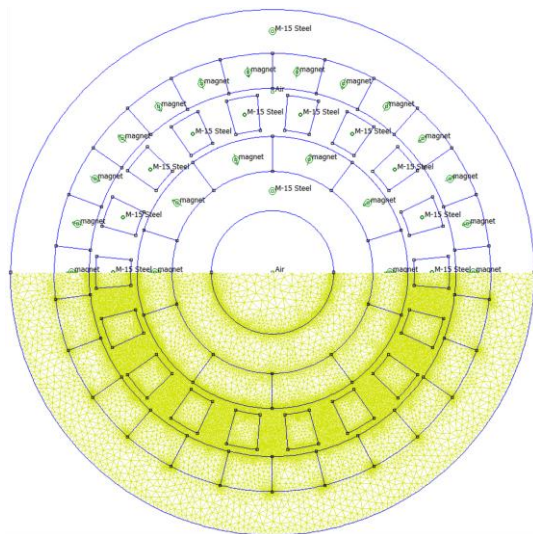
- Power transmission:
 - contactless
 - with environmental benefit (no lubrication)
 - with economic benefit (reducing maintenance)
- Much more reliable and efficient
- Torque limiter included (no clutch)

Procedure:

- Design of PMG with assignment of material properties and magnetisation direction for PMs
- Meshing and evaluation of magnetic field distribution
- Evaluation of transmissible torque for each different angle position of input/output rotors



Structural & magnetic FEM analyses



Constraints:

- External radius
- Axial length
- Fixed number of steel poles and PMs pole pairs

Goals:

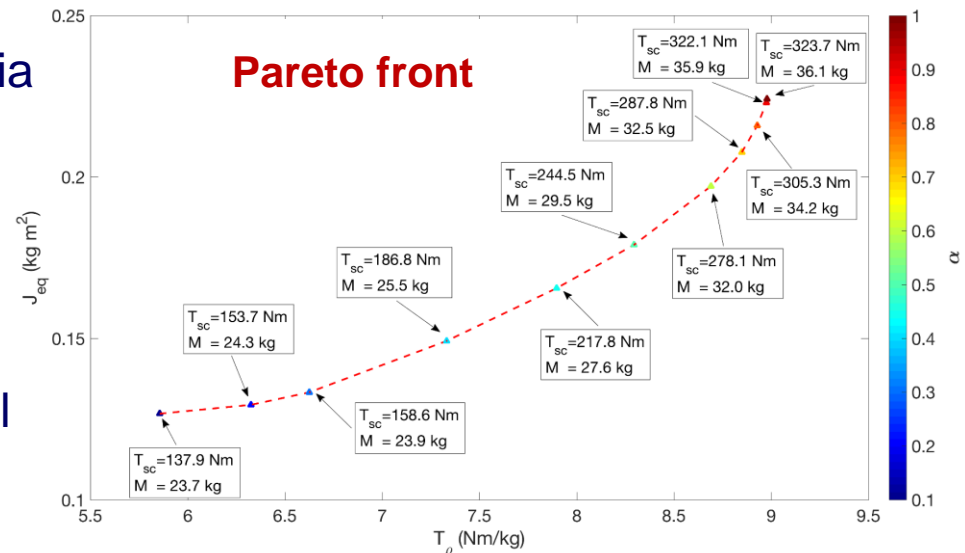
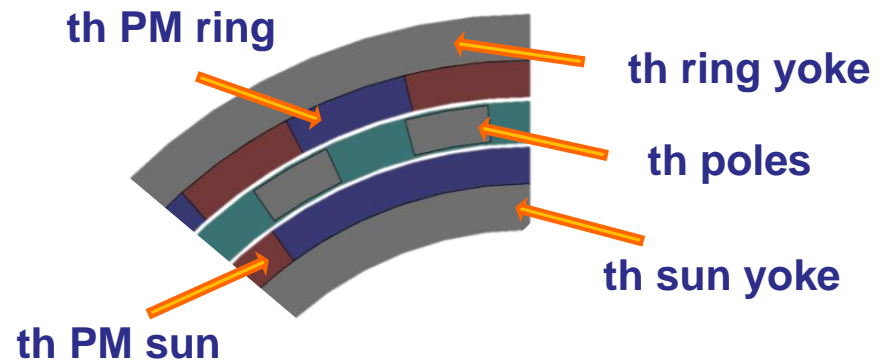
- Maximisation of the torque density
- Minimisation of the moment of inertia

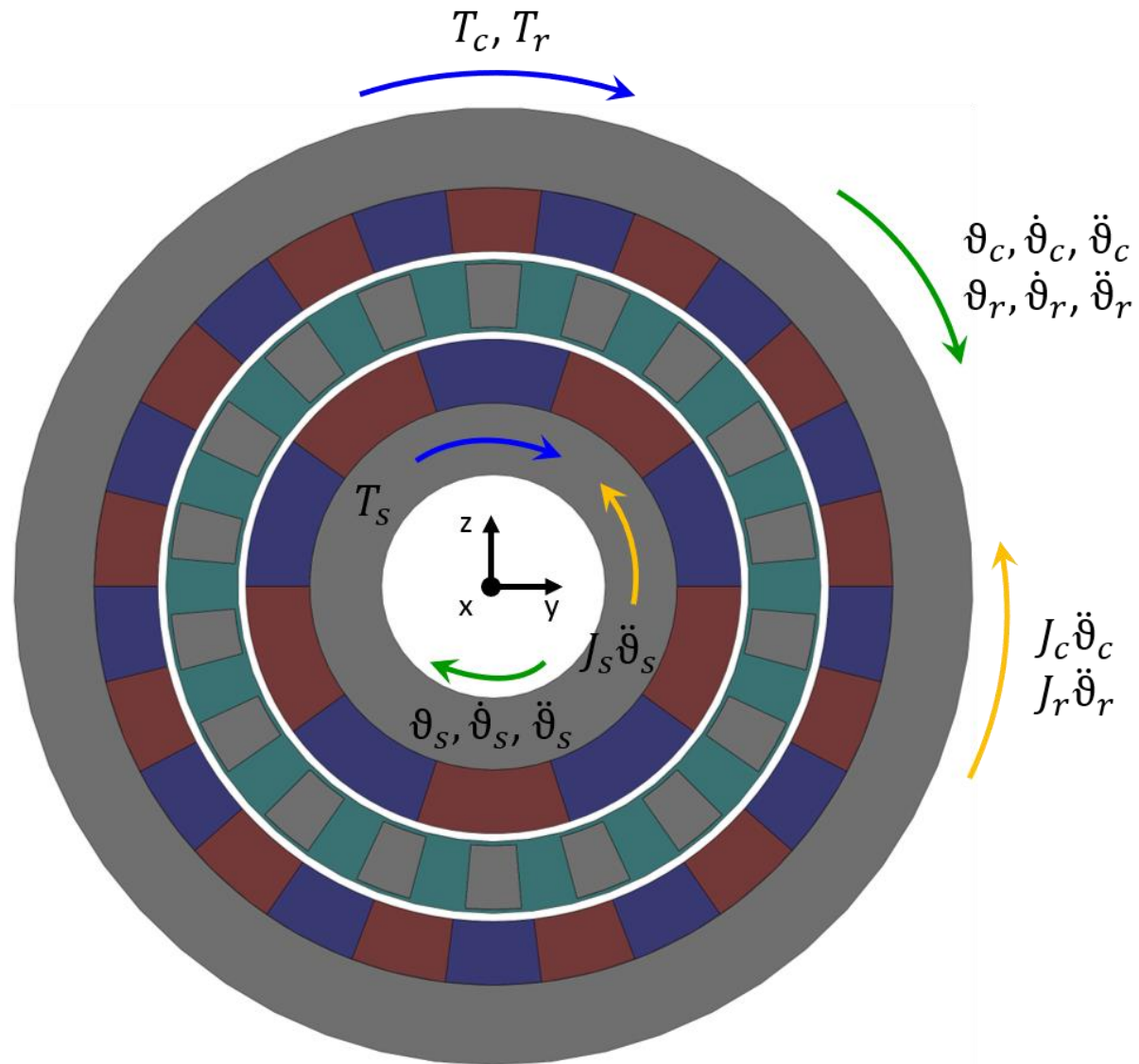
$$of (T_{\rho}, J_{eq}) = \max \left[\alpha \frac{T_{\rho}}{T_{\rho,ref}} - (1 - \alpha) \frac{J_{eq}}{J_{eq,ref}} \right]$$

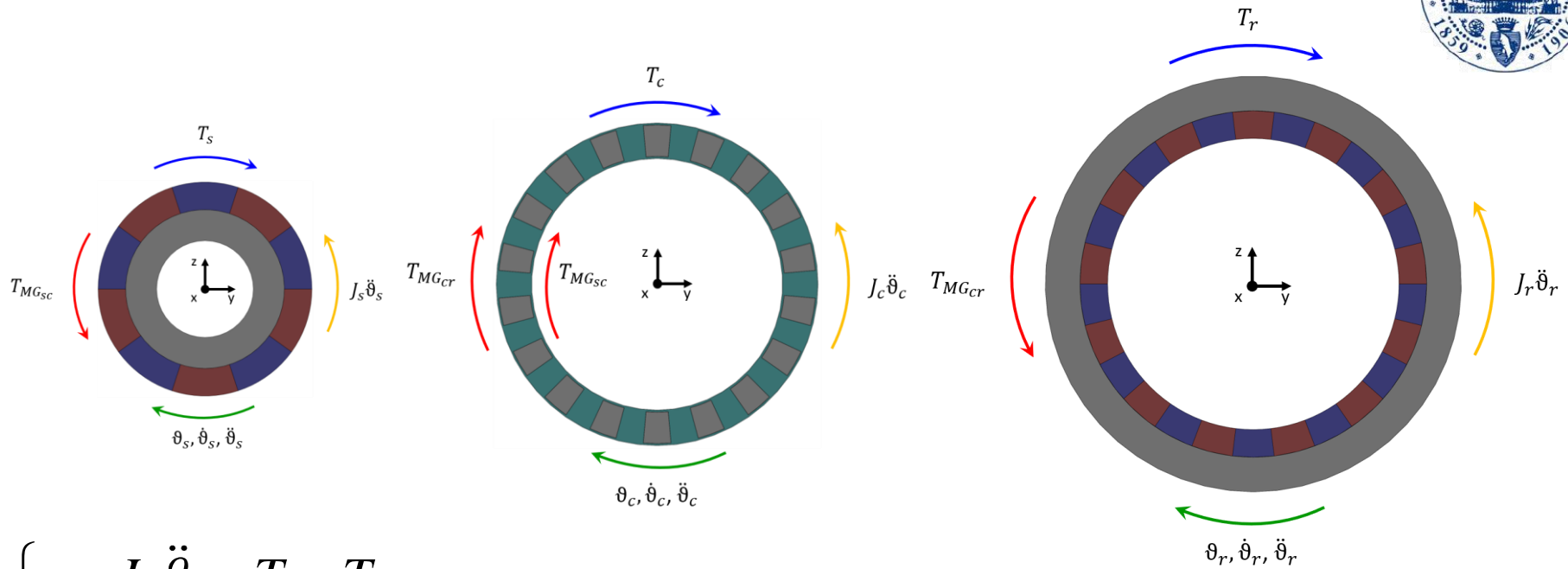
Limit:

- To transmit the same torque, overall dimensions of a PMG higher than a mechanical gear

5 DOFs







$$\begin{cases} J_s \ddot{\theta}_s = T_s - T_{MG_{sc}} \\ J_c \ddot{\theta}_c = T_c + T_{MG_{sc}} + T_{MG_{cr}} \\ J_r \ddot{\theta}_r = T_r - T_{MG_{cr}} \end{cases}$$

$$\frac{\dot{\theta}_s}{\dot{\theta}_r} = G_{s/r} = -\frac{n_r}{n_s}$$



➤ External torques (input or output)

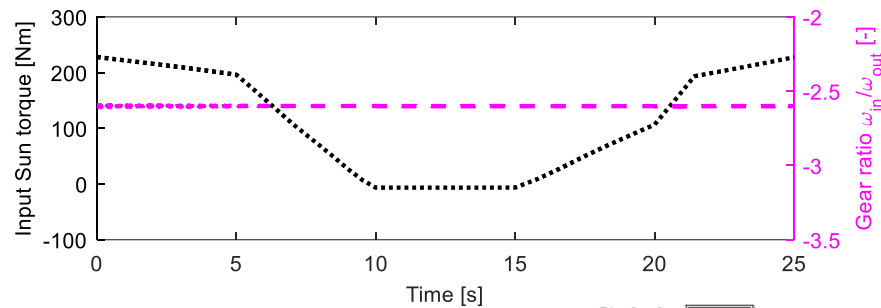
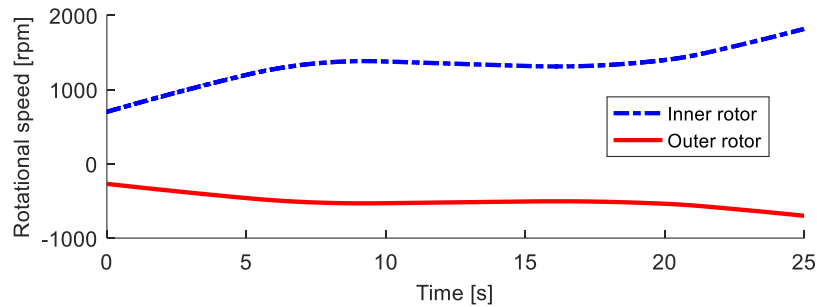


➤ Magnetic torques



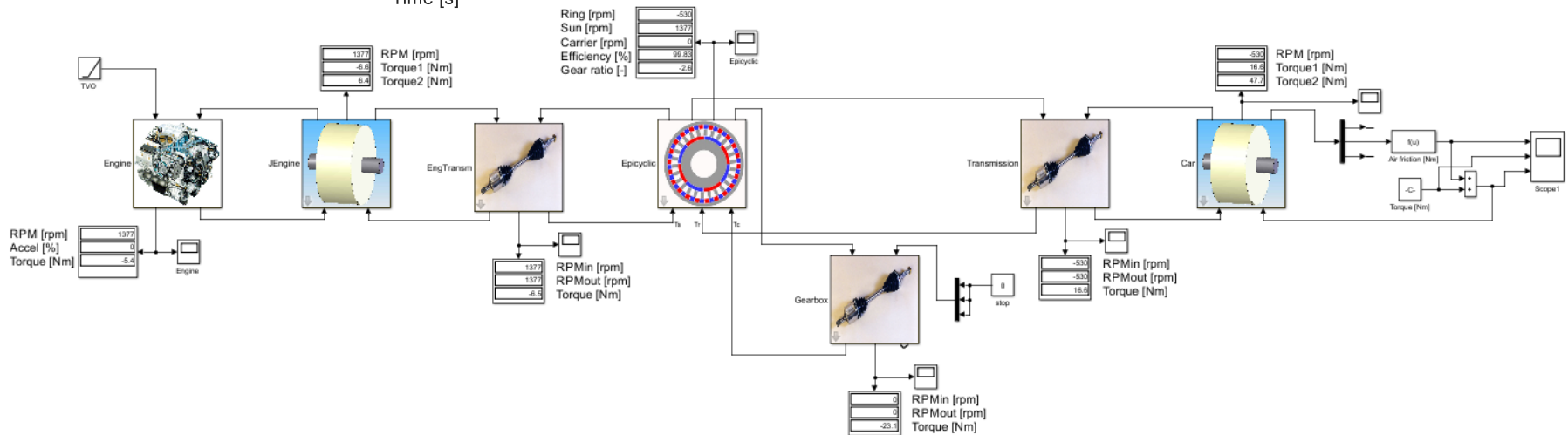
➤ Inertial torques

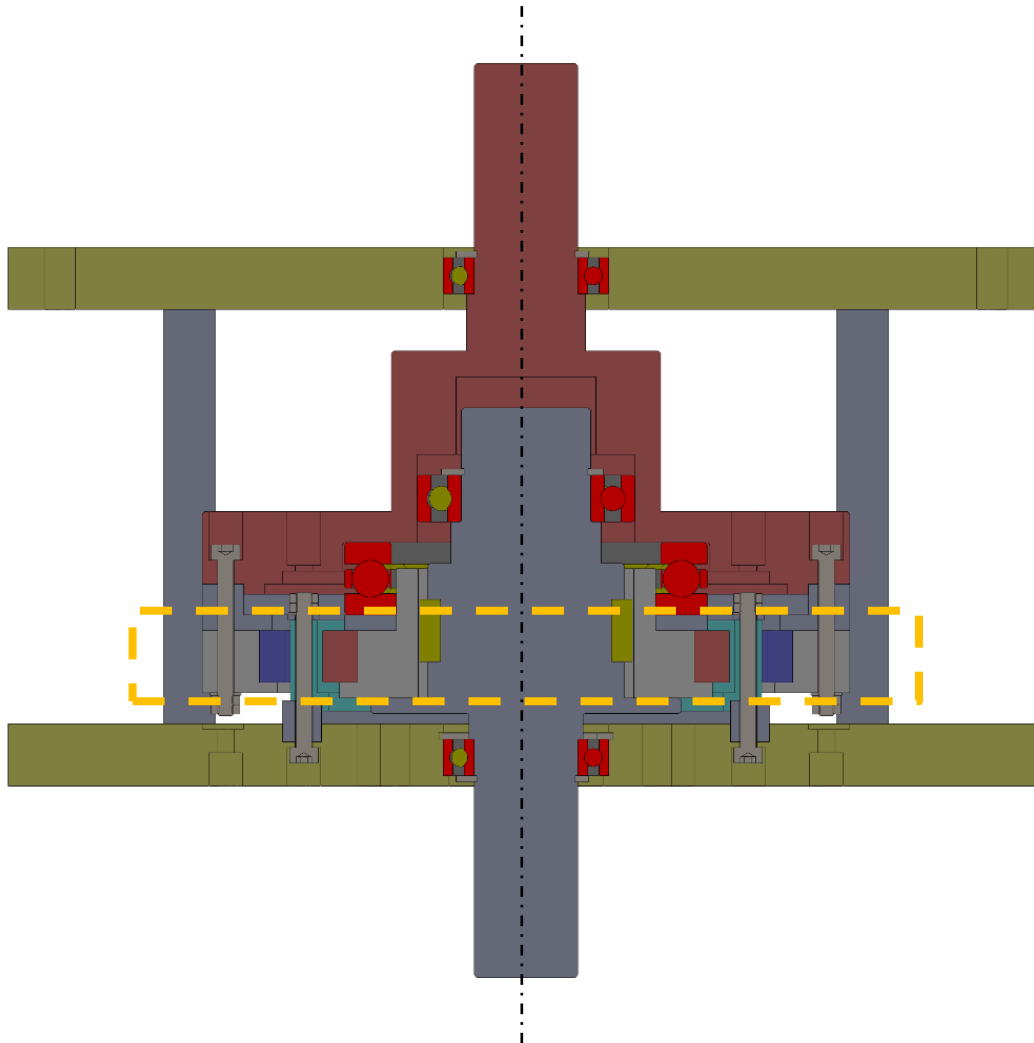
Transmission with a PMG



Model characteristics:

- Engine connected to inner rotor
- Resistive load connected to outer rotor
- The driver act on the GPP accelerating or decelerating





Advantages:

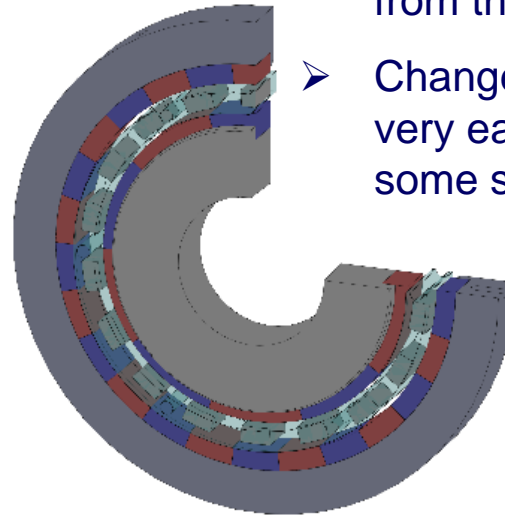
- The number of components is small
- The system weight is limited

Drawbacks:

- Difficult to assembly
- It is possible to change the configuration, but in a very invasive way
- The concentricity of magnetic gear rotors is very difficult to be replicated in case of disassembly

Advantages:

- The device is independent from the test bench
- Change of configuration is very easy, switching only some screws



Drawbacks:

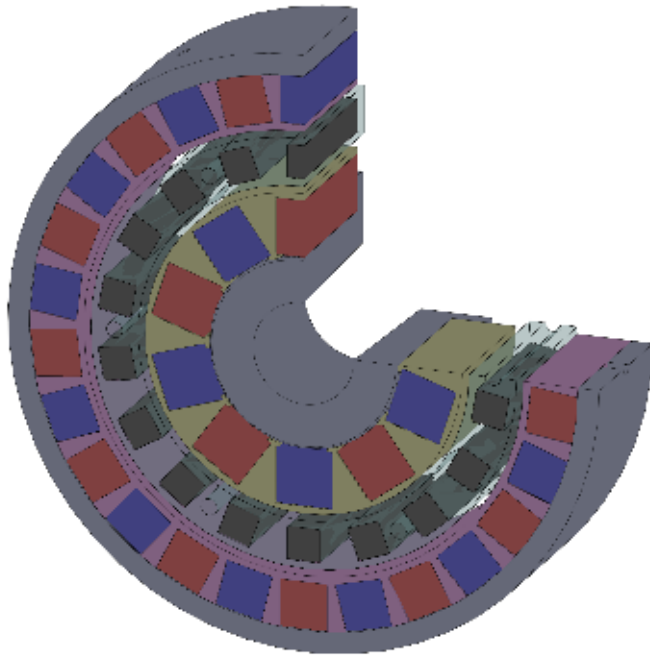
- 6 bearings instead of 4 (more dissipations)
- Expensive rare-earth customised permanent magnets (angular sectors and segmentation) and iron poles
- Expensive adoption of titanium components (paramagnetic and non-conductive material)

Advantages:

- The device is independent from the test bench
- Change of configuration is very easy, switching only some screws
- Low cost solution with commercial rare-earth permanent magnets and commercial ferrite poles
- Possible adoption of plastic material for magnetic gear and external components to realise with 3D printing

Drawbacks:

- Larger air-gaps between rotors
- Smaller torque density w.r.t previous solutions



System Complexity Reduction



Clutch



Mechanical gearbox



Torque limiter

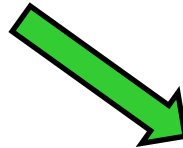


Main aims:

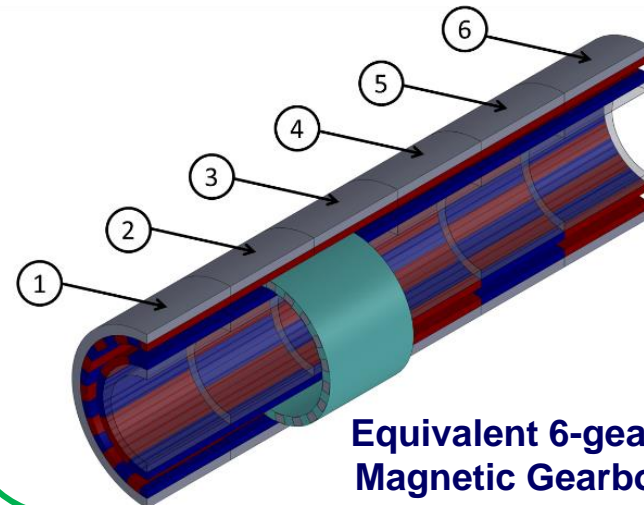
- Eliminate or reduce maintenance costs
- Reduce overall system complexity
- Enhance failure resistance

Customer advantages:

- Reduction of overall costs
- Improvement of product longevity



ALL IN ONE



**Equivalent 6-gears
Magnetic Gearbox**

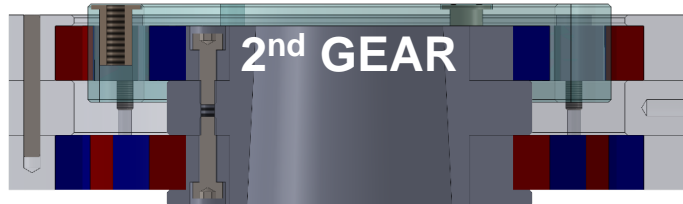
Dynamic simulation



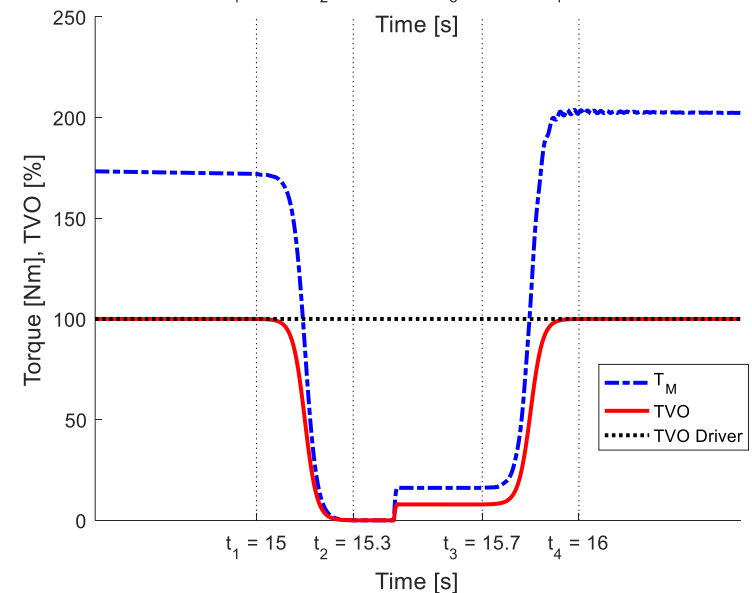
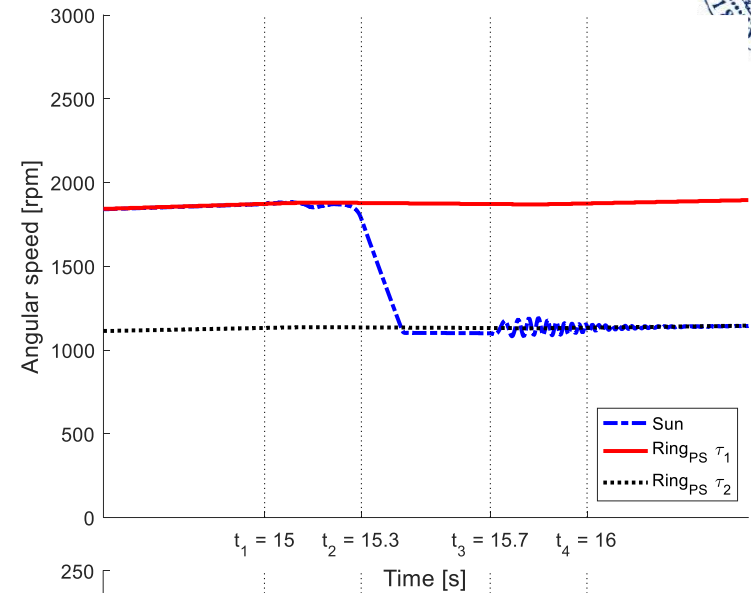
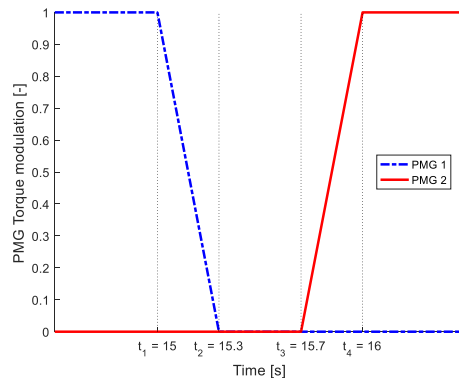
$$t < t_1$$

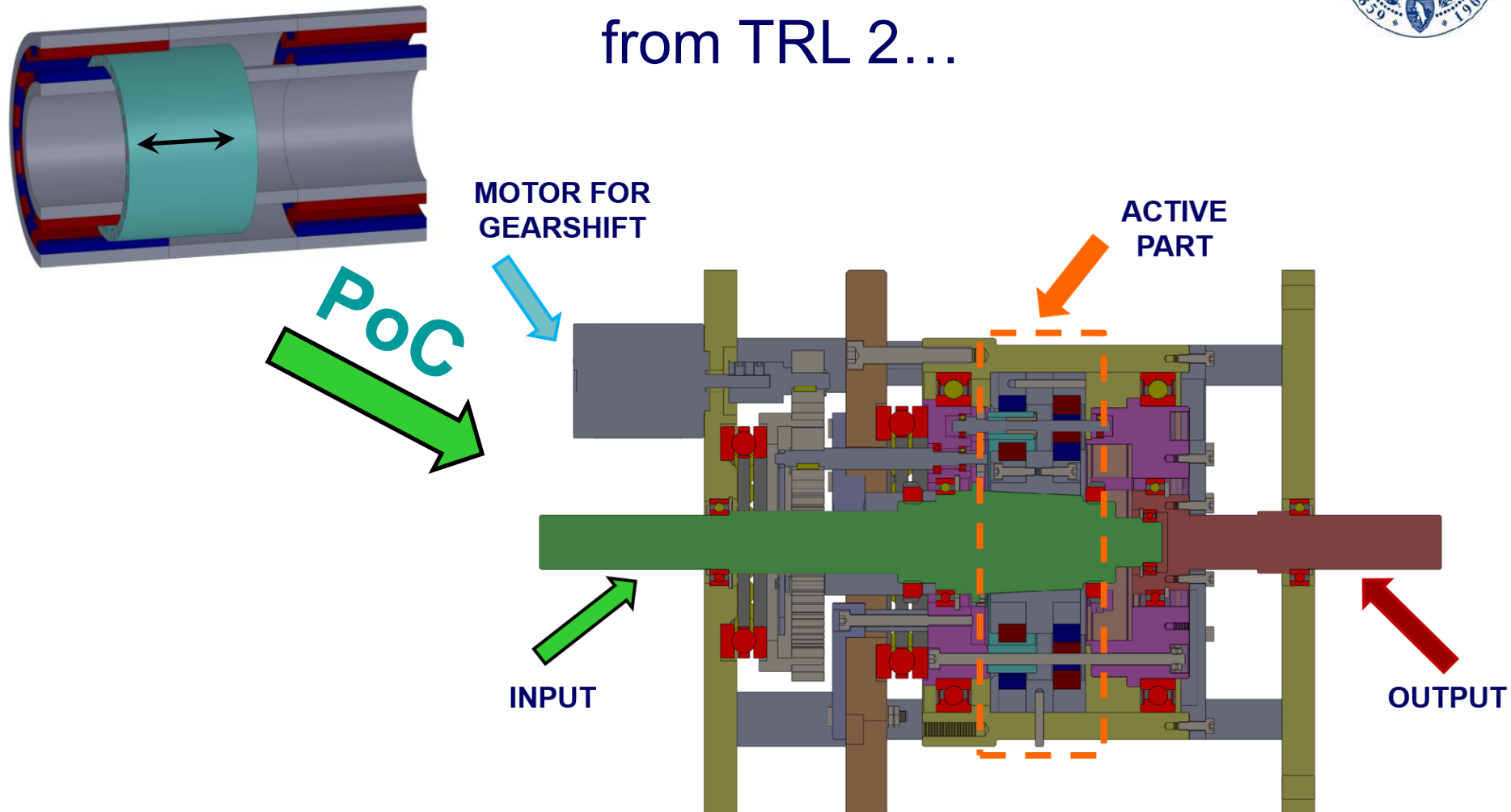


$$t_2 < t < t_3$$



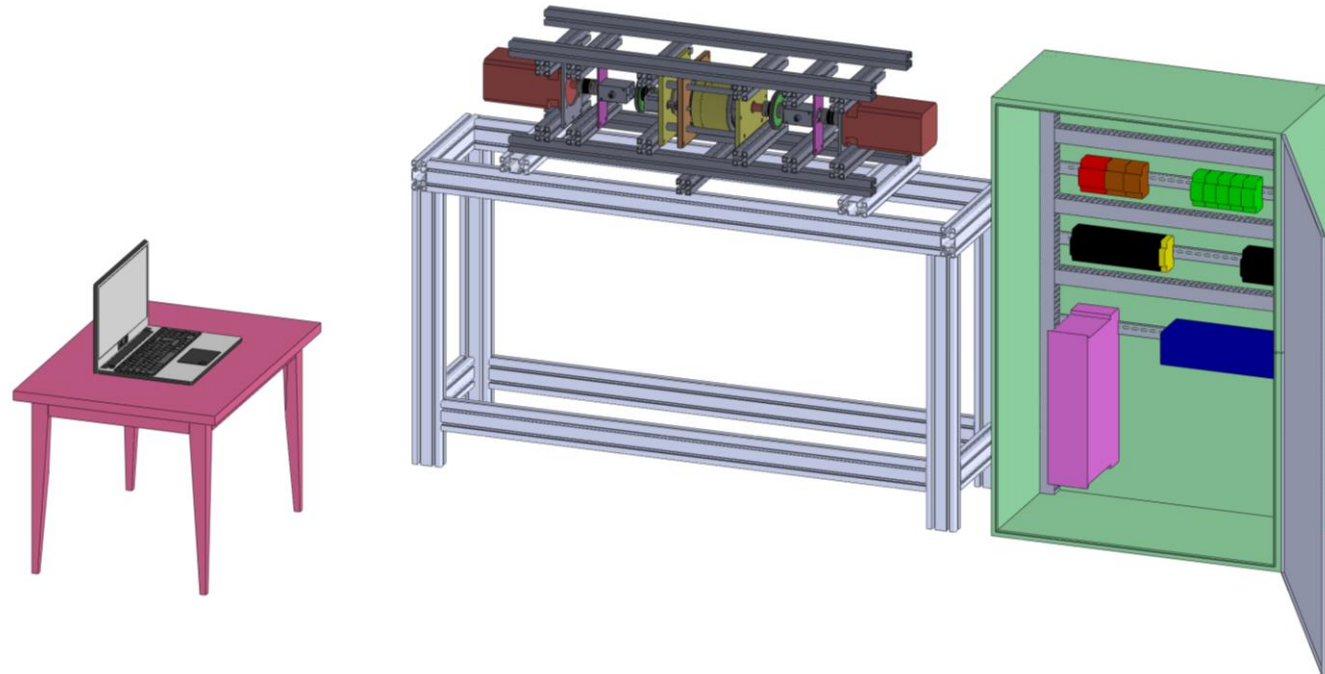
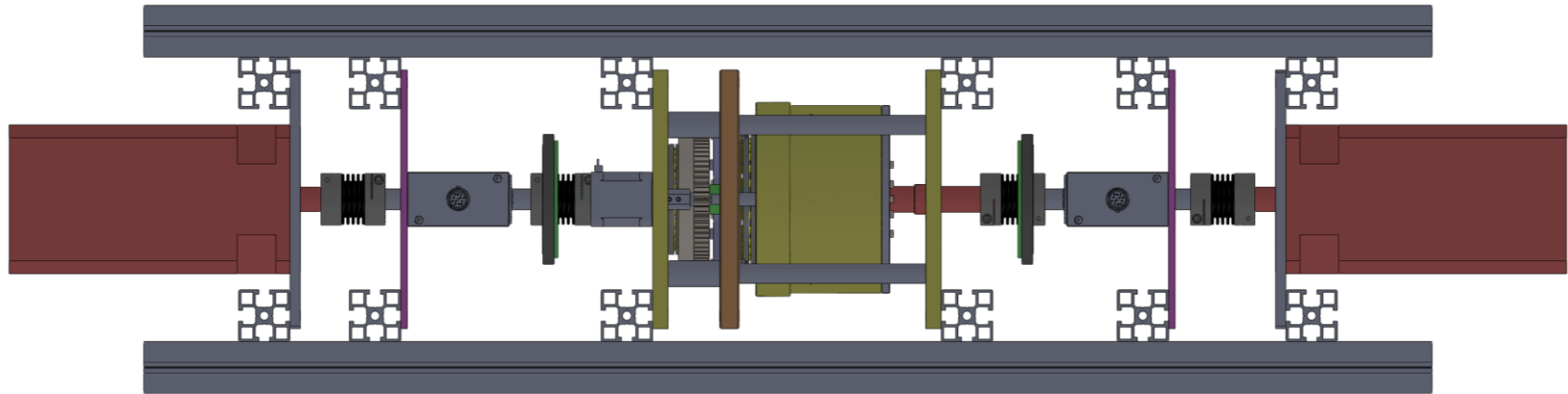
$$t > t_4$$





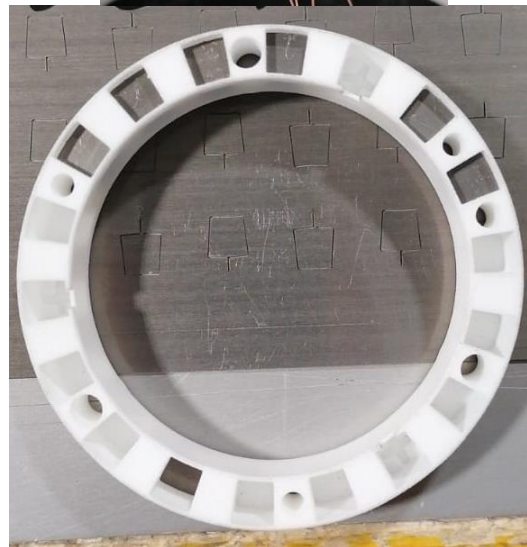
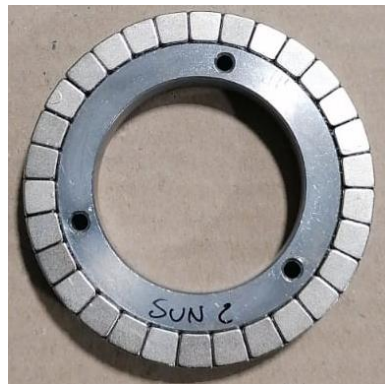
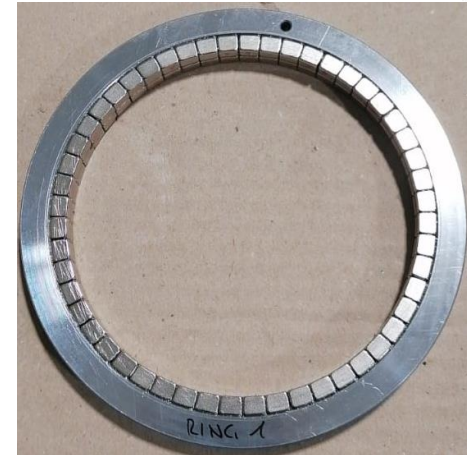
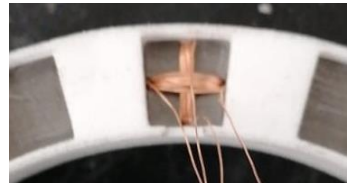
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Proof of Concept – Magnetic Gearbox



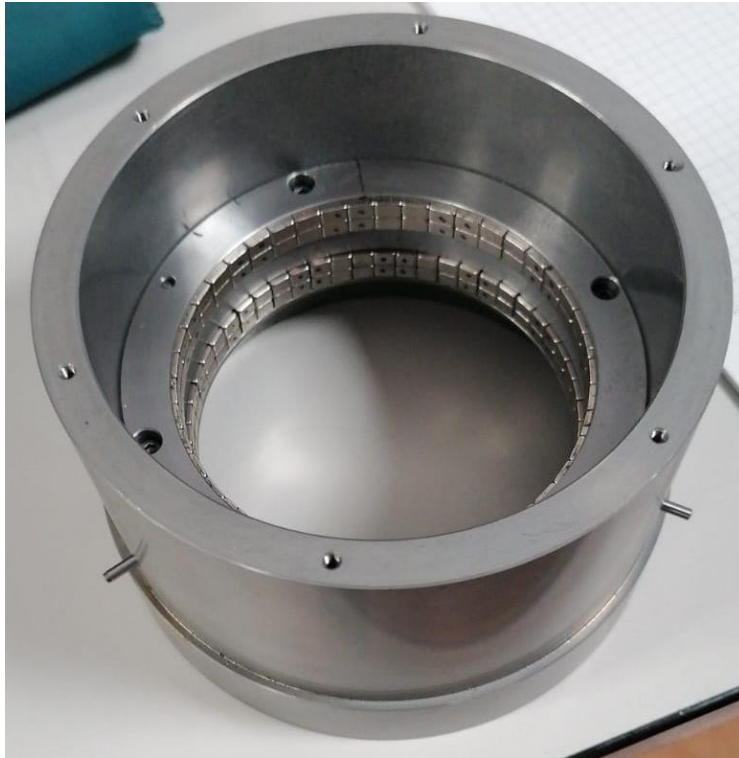


1st GEAR

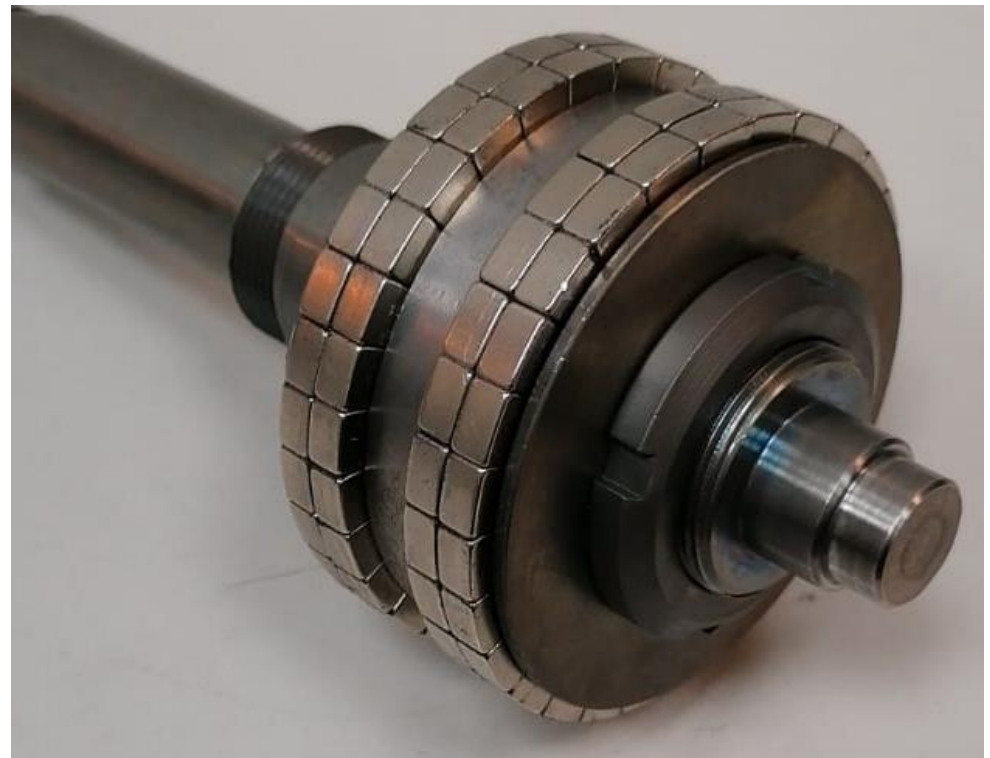


2nd GEAR





Outer rotor



Input shaft & inner rotor

Conclusions and future development



- A **nonlinear lumped parameters model** and a post processing tool for the evaluation of the **gearbox housing acceleration** starting from the bearing forces has been developed.
- A novel methodology for the **assessment of the gear shift noise** in a DCT gearbox has been proposed and proven to be effective to compare design solutions in the **transmission NVH perspective**.
- An **electro-mechanical approach** has been used for the design of a planetary magnetic gear; dynamic simulations have proven the **possible adoption of a PMG** in an automotive driveline.
- Several release of a PMG have been proposed and a **magnetic gearbox prototype** together with a **test bench** for **power-transmission application** has been designed and it is actually under construction.
- The magnetic gearbox will be **tested in stationary, transition and overload condition** for an **experimental validation** of the numerical and simulative results achieved so far.



International Journal Papers:

- Filippini M., Alotto P., Cirimele V., Repetto M., Ragusa C., **Dimauro L.**, Bonisoli E., “Magnetic loss analysis in coaxial magnetic gears”, *Electronics*, Vol. 8, Issue 11, 2019, paper n° 1320, pp. 1-15.
- Cirimele V., **Dimauro L.**, Repetto M., Bonisoli E., “Multi-objective Optimization of a Magnetic Gear for Powertrain Applications”, *International Journal of Applied Electromagnetics and Mechanics*, Vol. 60, Issue S1, 2019, pp. S25-S34.

Book Chapters:

- Bonisoli E., Lisitano D., **Dimauro L.**, Peroni L., “A proposal of dynamic behaviour design based on mode shape tracing: numerical application to a motorbike frame”, *Dynamic Substructures*, Vol. 4, *Proceedings of the 37th IMAC*, A Conference and Exposition on Structural Dynamics, Conference Proceedings of the Society for Experimental Mechanics Series, 186 pp., Ch. 14, 2020, Springer.
- Bonisoli E., Casazza M., Lisitano D., **Dimauro L.**, “Parametric experimental modal analysis of a modern violin based on a Guarneri del Gesù model”, *Rotating Machinery, Vibro-Acoustics & Laser Vibrometry*, Vol. 7, *Proceedings of the 36th IMAC*, A Conference and Exposition on Structural Dynamics, Conference Proceedings of the Society for Experimental Mechanics Series, 244 pp., Ch. 21, 2019.



International Conference Papers:

- Galvagno, E., **Dimauro, L.**, Mari, G., Velardocchia, M. et al., “Dual Clutch Transmission Vibrations during Gear Shift: A Simulation-Based Approach for Clunking Noise Assessment”, *SAE Technical Paper*, 2019-01-1553, 2019, DOI: 10.4271/2019-01-1553, pp. 1-12.
- Bonisoli E., Lisitano D., **Dimauro L.**, “Experimental and numerical mode shape tracing from components to whole motorbike chassis”, *International Conference on Noise and Vibration Engineering, ISMA*, 2018, Leuven, Belgium, September 17-19, pp. 3597-3604.



PhD Final Hearings XXXIII Cycle

Torino, 17th Dec. 2020



Thanks for Your attention!

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Supervisor:

Prof. Elvio BONISOLI

PhD Candidate:

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