

TUNED HARMONIC ABSORBERS FOR ENGINE TORSIONAL VIBRATIONS



RESEARCH & DEVELOPMENT

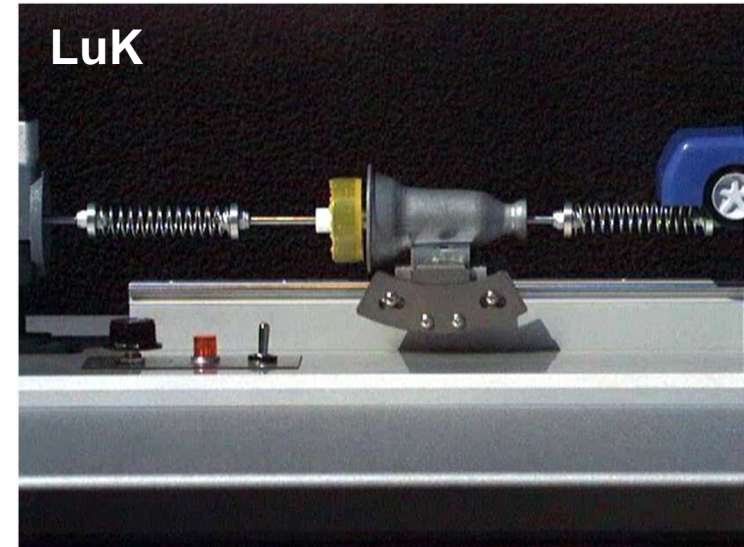
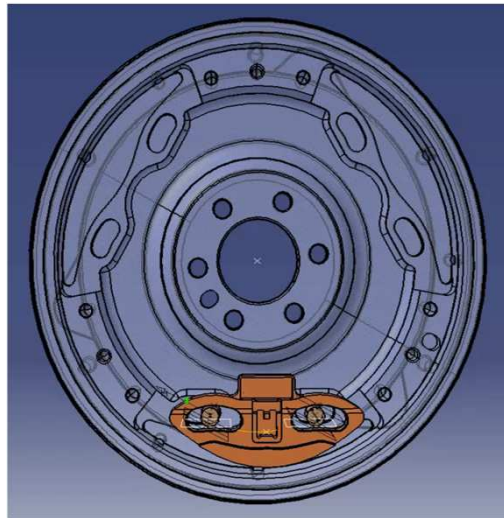
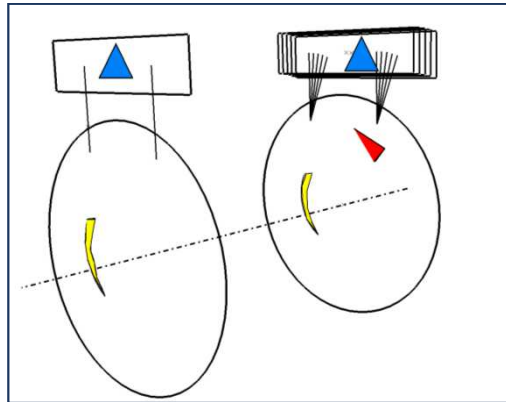
10 octobre 2016
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DRD/DRIA/DSTF

Zoran DIMITRIJEVIC

CONTEXT AND ISSUES

- Torsional vibrations absorption using pendular systems



- Limiting harmonics on clutches, flywheels, cankshafts, camshafts :
 - H2 on 4 cylinders engine.
 - H1,5 and H3 on 3 cylinders engine.
 - Here we want to prevent h0,75 on a 3 cylinders engine.

CONTEXT AND ISSUES

History and bibliography

- Since the early 1910's (Kutzbach 1911) multiple applications of the oscillating pendular system for damping torsional vibrations:

(12) NACH DEM VERTRAG ÜBER DIE INTERNATIONALE PATENTWESEN (PCT) VERÖFFENTLICHT

(19) Weltorganisation für geistiges Eigentum International

(43) International 15. September 2002

(51) International F16F

(21) International

(22) International

(25) Einreichungsdatum

(26) Veröffentlichungsdatum

(30) Angaben zum Anmelder

2002

EXPERIMENTAL INVESTIGATION OF CIRCULAR PATH CENTRIFUGAL PENDULUM VIBRATION ABSORBERS

By Tyler Monroe Nester

A THESIS Submitted to Michigan State University in partial fulfillment of the requirements for the degree of MASTER OF SCIENCE Department of Mechanical Engineering 2002

Figure 3.2: Schematic diagram of the experimental apparatus.

The remaining components shown in Figure 3.2 are a frequency to voltage converter (C), a computer (G), and a low pass filter (A). The frequency to voltage converter (C) is connected to the motor (D) and the computer (G). The low pass filter (A) is connected to the computer (G) and the motor (D). The pendulum flange (B) is mounted on the motor (D) and the pendulum masses (E, F) are attached to the flange (B). The pendulum masses (E, F) are arranged symmetrically about the vertical axis of the motor (D). The pendulum masses (E, F) are connected to the flange (B) by means of bolts (26) on both sides of the flange (22) and are guided by rollers and can be pivoted relative to the flange (22) to a limited extent, wherein one pendulum mass (24) has a lateral surface (40) that faces the lateral surface (40) of the other pendulum mass (24) and wherein the lateral surface (40) has a substantial angle (42).

1992

1992

Figure 3.3: Absorber arc-length amplitude versus non-dimensional torque, Γ_θ , for forcing order 1.2987.

Riga 2013

(57) Abstract: The invention relates to a centrifugal pendulum mechanism (20), comprising a pendulum flange (22) and pendulum masses (24) which are fastened by means of bolts (26) on both sides of the flange (22) and are guided by rollers and can be pivoted relative to the flange (22) to a limited extent, wherein one pendulum mass (24) has a lateral surface (40) that faces the lateral surface (40) of the other pendulum mass (24) and wherein the lateral surface (40) has a substantial angle (42).

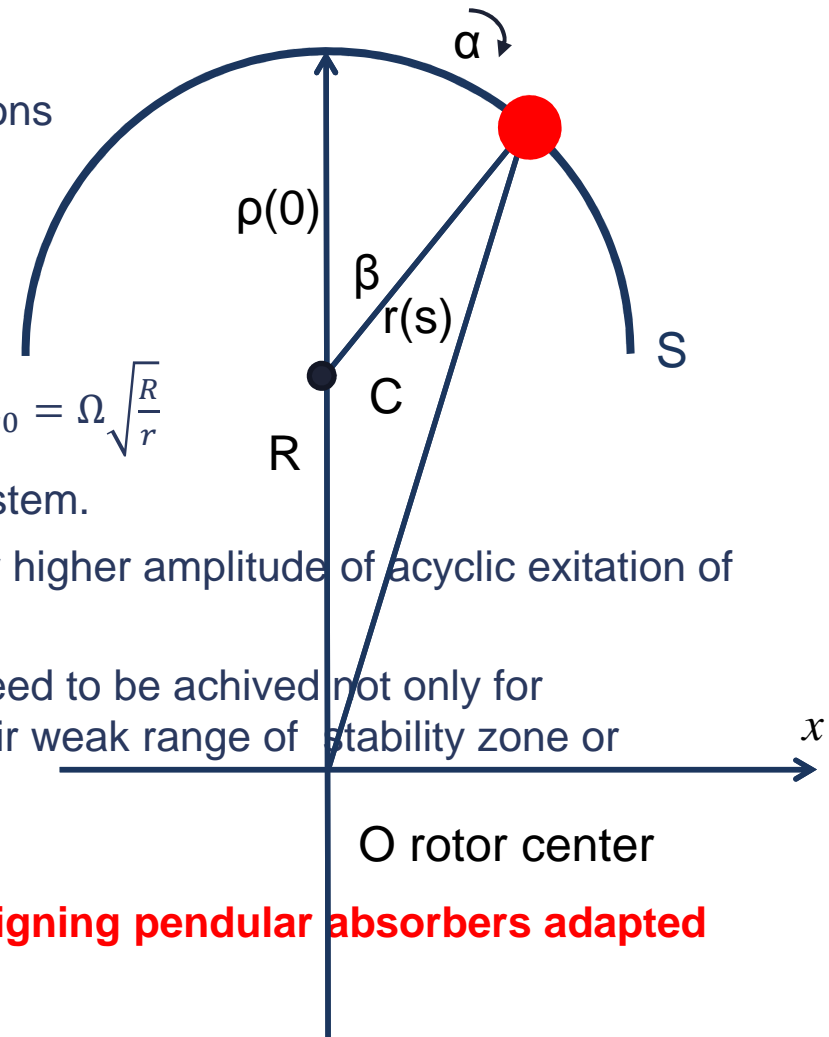
Zusammenfassung: Die Erfindung betrifft eine raftpendeleinrichtung (20) mit einem flansch (22) und mit beidseitig des flansches (22) mittel Bolzen (26) befestigten und geföhren geföhren und gegenüber dem flansch (22) begrenzt verschwenkbare pendelmassen (24) vorgeschlagen, wobei wenigstens zwei pendelmassen (24) an dem flansch (22) gegenseitig benachbart angeordnet sind.

CONTEXT AND ISSUES

- Using « classical » linearized Lagrangian equations

$$\begin{aligned} [I + J + m(R + r)^2] \ddot{\alpha} + [mr(R + r)] \ddot{\beta} &= \gamma \\ mr(R + r) \ddot{\alpha} + mr^2 \ddot{\beta} + mRr\Omega^2 \beta &= 0 \end{aligned}$$

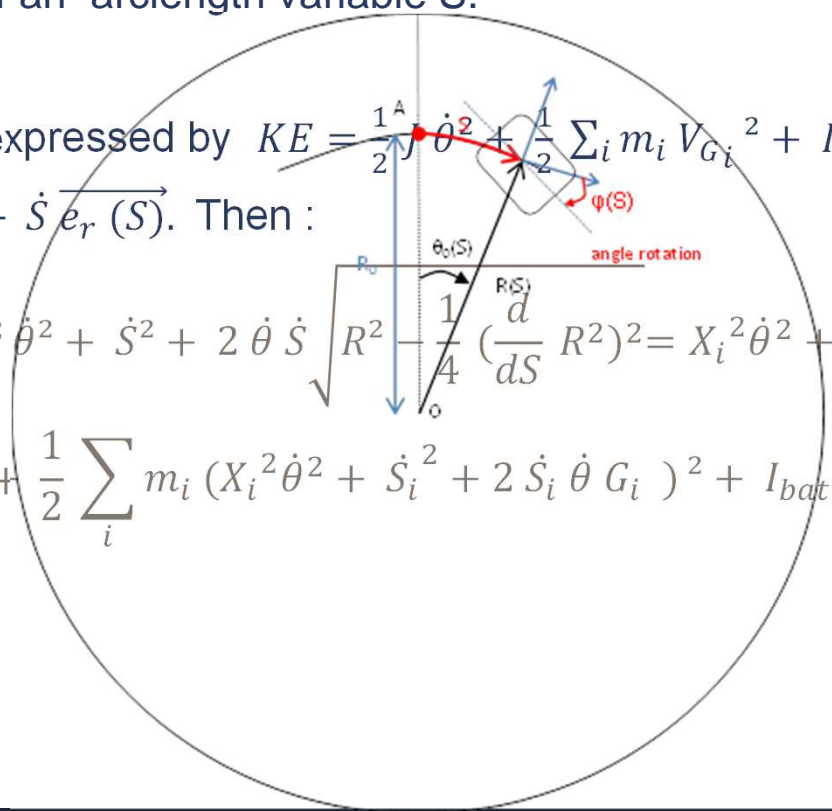
- Then the absorbers are tuned to the frequency $\Omega_0 = \Omega \sqrt{\frac{R}{r}}$
- This is true only for weak perturbations of the system.
- In the industrial context we need robust paths for higher amplitude of acyclic excitation of the rotor.
- The nonlinear analyzes of the coupled system need to be achieved not only for epicycloidal or tautochrone paths because of their weak range of stability zone or efficiency.
- The main aim is to produce a tool for pre-designing pendular absorbers adapted on the flywheel of a 3 cylinders engine.**



NONLINEAR DYNAMIC EQUATIONS OF THE COUPLED SYSTEM ENGINE FLYWHEEL WITH PENDULAR ABSORBERS

- We try to find the steady responses in 1:1 resonance of the coupled system rotor with N pendulars when pendulars are in a synchronous motion.
- The pendulars can rotate, and the path is represented by the distance from the center of rotation, function of an arclength variable S.

- Kinetic energy is expressed by $KE = \frac{1}{2} J \dot{\theta}^2 + \frac{1}{2} \sum_i m_i V_{G_i}^2 + I_{batteur_i} (\dot{S}_i \frac{d\varphi_i}{dS_i})^2$ with $\vec{V}_G = R(S)\dot{\theta} \vec{e}_\theta(S) + \dot{S} \vec{e}_r(S)$. Then :



$$V_G^2 = R^2 \dot{\theta}^2 + \dot{S}^2 + 2 \dot{\theta} \dot{S} \sqrt{R^2 - \frac{1}{4} \left(\frac{d}{dS} R^2\right)^2} = X_i^2 \dot{\theta}^2 + \dot{S}_i^2 + 2 \dot{S}_i \dot{\theta} G_i$$

$$KE = \frac{1}{2} J \dot{\theta}^2 + \frac{1}{2} \sum_i m_i (X_i^2 \dot{\theta}^2 + \dot{S}_i^2 + 2 \dot{S}_i \dot{\theta} G_i)^2 + I_{batteur_i} (\dot{S}_i \frac{d\varphi_i}{dS_i})^2$$

$T = T_0 + T(\theta)$
 couple

NONLINEAR DYNAMIC EQUATIONS OF THE COUPLED SYSTEM ENGINE FLYWHEEL WITH PENDULAR ABSORBERS

- Lagrangian equations of motion for the coupled system are written :

$$J\ddot{\theta} + \sum_j m_j \left\{ \frac{dX_j}{ds_j} \dot{s}_j \dot{\theta} + X_j(s_j)\ddot{\theta} + G_j(s_j)\ddot{s}_j + \frac{dG_j}{ds_j} \dot{s}_j^2 \right\} + I_{batteur_j} \left(\frac{d\varphi_j}{ds_j} \ddot{s}_j + \frac{d^2\varphi_j}{ds_j^2} \dot{s}_j^2 \right) = \sum_j c_{a_j} G_j(s_j)\dot{s}_j - c_0 \dot{\theta} + T_0 + T(\theta)$$

$$-\frac{1}{2} m_i \frac{dX_i}{ds_i} \dot{\theta}^2 + m_i (\ddot{s}_i + G_i(s_i)\ddot{\theta}) + I_{batteur_i} \left(\left(\frac{d\varphi_i}{ds_i} \right)^2 \ddot{s}_i + \frac{d\varphi_i}{ds_i} s_i \ddot{\theta} \right) + c_{a_i} s_i \dot{s}_i = 0$$

- Adimensionnalization using $s_i = \frac{S_i}{R_i(0)}$, $X_i(S_i) = R_i(0)^2 x(s_i)$, $G(S_i) = R_i(0)g(s_i)$ is done:

$$\begin{aligned} \ddot{\theta} + \sum_j b_j \left\{ \frac{dx_j}{ds_j} \dot{s}_j \dot{\theta} + x_j(s_j)\ddot{\theta} + g_j(s_j)\ddot{s}_j + \frac{dg_j}{ds_j} \dot{s}_j^2 \right\} + b_j K_j \left\{ \frac{d\varphi_j}{ds_j} \ddot{s}_j + \frac{d^2\varphi_j}{ds_j^2} \dot{s}_j^2 \right\} \\ = \sum_j \frac{c_{a_j} R_j(0)^2}{J} g_j(s_j)\dot{s}_j - \frac{c_0}{J} \dot{\theta} + \frac{T_0}{J} + \frac{T(\theta)}{J} \end{aligned}$$

$$-\frac{1}{2} b_i \frac{dx_i}{ds_i} \dot{\theta}^2 + b_i \{ \dot{s}_i + g_i(s_i)\ddot{\theta} \} + b_i K_i \left\{ \left(\frac{d\varphi_i}{ds_i} \right)^2 \ddot{s}_i + \frac{d\varphi_i}{ds_i} s_i \ddot{\theta} \right\} = -\frac{c_{a_i} R_i(0)^2}{J} \dot{s}_i$$

$$\text{With } b_i = \frac{m_i R_i(0)^2}{J}, I_{batteur_i} = m_i \rho_i^2, K_i = \left(\frac{\rho_i}{R_i(0)} \right)^2$$

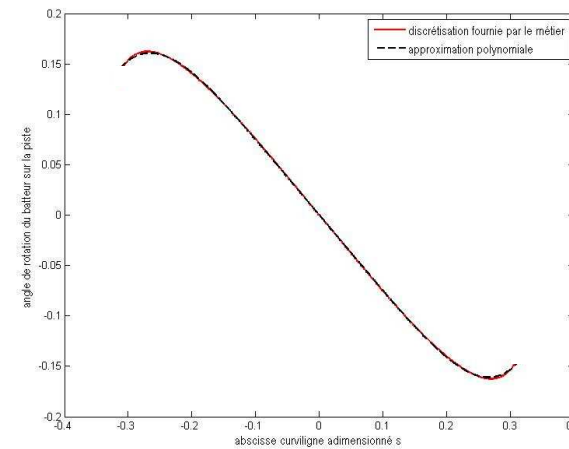
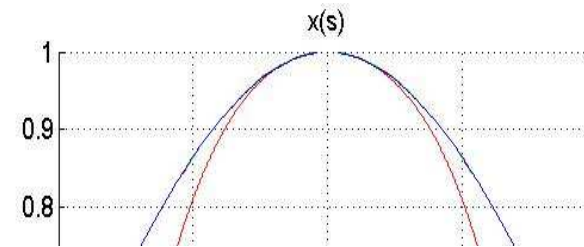
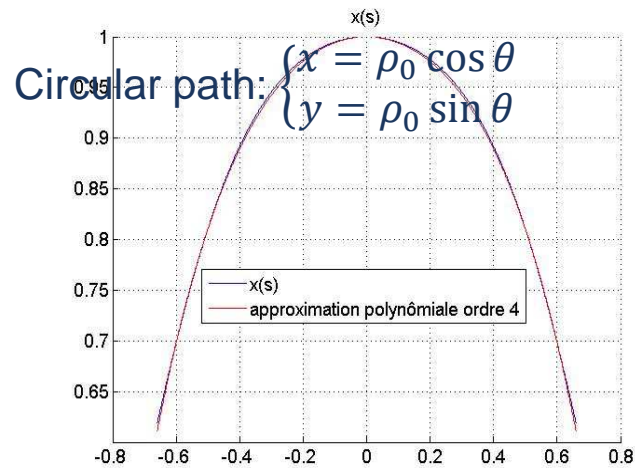
NONLINEAR DYNAMIC EQUATIONS OF THE COUPLED SYSTEM ENGINE FLYWHEEL AND PENDULAR ABSORBERS

■ Approximations of the path curvature:

- $x(s) = 1 - \tilde{n}^2 s^2 + \gamma s^4 + o(s^4)$
- $g(s) = 1 - \frac{1}{2}(\tilde{n}^2 + \tilde{n}^4)s^2 + \frac{1}{4}\left(2\gamma(1 + 4\tilde{n}^2) + \frac{1}{2}(\tilde{n}^2 + \tilde{n}^4)^2\right)s^4 + o(s^4)$
- $\varphi(s) = \alpha s + \beta s^2 + \delta s^3 + o(s^3)$

"Hardening" path

$$\begin{cases} x = b(1 + k_1\theta^2 + k_2\theta^4) \cos \theta \\ y = b(1 + k_1\theta^2 + k_2\theta^4) \sin \theta \end{cases}$$



NONLINEAR DYNAMIC EQUATIONS OF THE COUPLED SYSTEM ENGINE FLYWHEEL AND PENDULAR ABSORBERS

- We define the scale factor $\varepsilon = \frac{N m_i R_i(0)^2}{J}$
- Considering the next variable changes and introducing the detuning parameter σ :

$$\left\{ \begin{array}{l} v = \frac{\dot{\theta}}{\Omega} \\ \ddot{\theta} = \Omega^2 v v' \\ (\dot{\quad}) = \Omega v (\quad)' \\ (\ddot{\quad}) = \Omega^2 v v' (\quad)' + \Omega^2 v^2 (\quad)'' \end{array} \right.$$

- $v = 1 + \varepsilon^{3/2} w + o(\varepsilon^{3/2})$
- $s_i = \sqrt{\varepsilon} z_i, \tilde{\mu}_i = \varepsilon \mu_i, \tilde{\Gamma} = \varepsilon^{3/2} \Gamma, \tilde{n} = n(1 + \varepsilon\sigma)$

- The motion equations become:

$$v' = \varepsilon^{3/2} \left\{ \tilde{\Gamma} + \frac{n^2}{N} \sum_j \frac{z_j}{(1 + K_j \alpha_j^2)} \right\}$$

$$\begin{aligned} & z_i'' (1 + K_i \alpha_i^2) + n^2 z_i \\ & = \sqrt{\varepsilon} \{ -4 K_i \alpha_i \beta_i z_i z_i'' \} \\ & + \varepsilon \left\{ 2 \gamma z_i^3 - \tilde{\mu}_{a,i} z_i' - \tilde{\Gamma} - 2 n^2 \sigma z_i - 4 K_i \beta_i^2 z_i^2 z_i'' - 6 K_i \alpha_i \delta_i z_i^2 z_i'' - \frac{n^2}{N} \sum_j \frac{z_j}{(1 + K_j \alpha_j^2)} \right\} \end{aligned}$$

NONLINEAR DYNAMIC EQUATIONS OF THE COUPLED SYSTEM ENGINE FLYWHEEL AND PENDULAR ABSORBERS

- By the hypothesis of a weak nonlinearity, we look for solutions of the type $z_i = a_i \sin(\omega_{0i}\theta + \psi_i)$. The motion of the pendular absorbers are then solution of:

$$\begin{pmatrix} a_i' \\ a_i \varphi_i' \end{pmatrix} = \begin{pmatrix} \sin(\omega_{0i}\theta + \psi_i) & \frac{1}{A} \cos(\omega_{0i}\theta + \psi_i) \\ \cos(\omega_{0i}\theta + \psi_i) & -\frac{1}{A} \sin(\omega_{0i}\theta + \psi_i) \end{pmatrix} \begin{pmatrix} 0 \\ \sqrt{\varepsilon} f_1 + \varepsilon g_1 \end{pmatrix}$$

- The averaged equations are then:

$$\begin{pmatrix} \bar{a}_i' \\ \bar{a}_i \bar{\psi}_i' \end{pmatrix} = \begin{pmatrix} \int_0^{2\pi} \frac{\cos(\omega_{0i}\theta + \bar{\psi}_i)}{\omega_{0i}} (\varepsilon \bar{g}_1 + \varepsilon \bar{f}_1 \bar{h}_1) d\theta \\ \int_0^{2\pi} -\frac{\sin(\omega_{0i}\theta + \bar{\psi}_i)}{\omega_{0i}} (\varepsilon \bar{g}_1 + \varepsilon \bar{f}_1 \bar{h}_1) d\theta \end{pmatrix}$$

- Synchronous steady responses are solved for $\bar{a}_i' = 0$ and $\bar{\varphi}_i' = 0$. Let be $\bar{a}_i = \bar{a}_j = r$, $\bar{\psi}_i = \bar{\psi}_j = \psi$, $\tilde{\mu}_{a,i} = \tilde{\mu}_{a,j} = \tilde{\mu}$, $\omega_{0i} = \omega_{0j} = \omega_0$, we finally solve (r, ψ) :

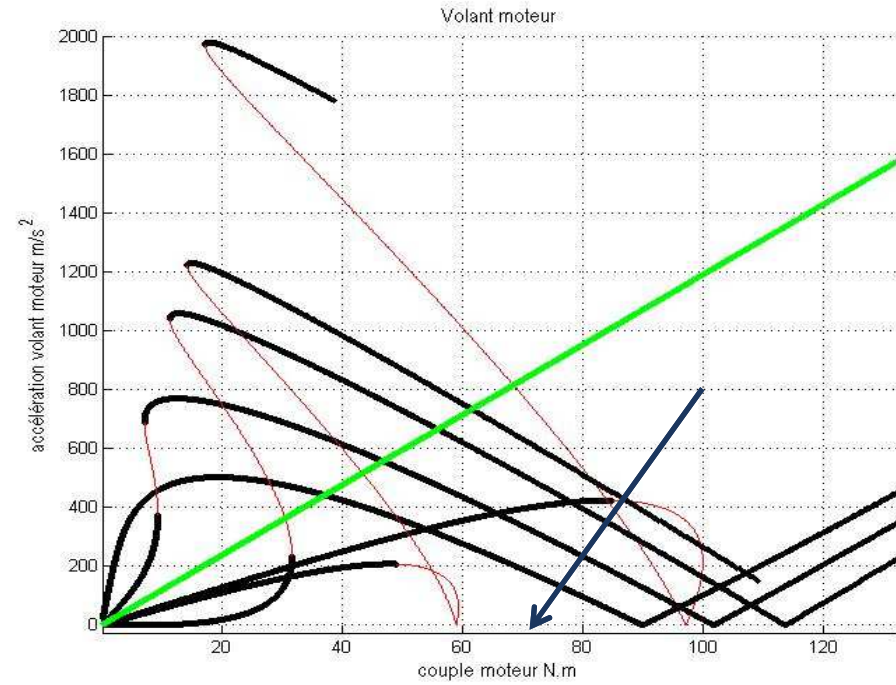
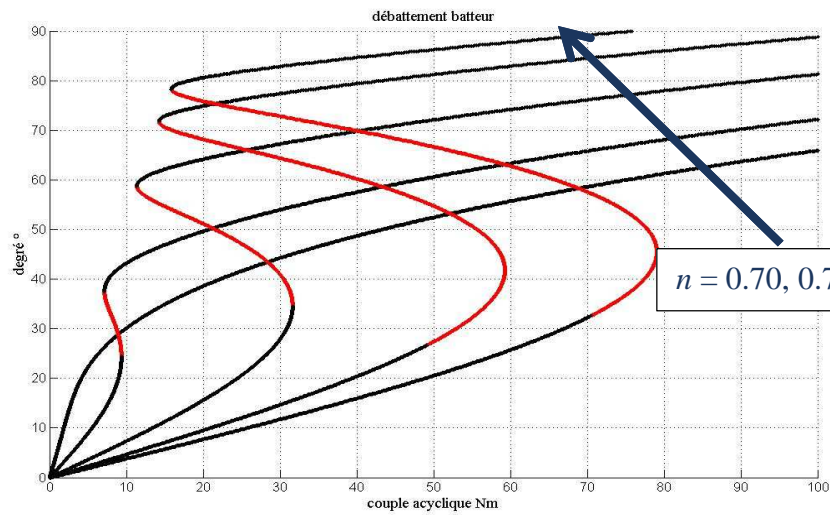
$$r \pi \tilde{\mu} = \frac{\tilde{\Gamma}}{(n^2 - \omega_0^2)} \left\{ -2 n \cos(\varphi) \sin^2 \left(\frac{n \pi}{\omega_0} \right) + \omega_0 \sin(\varphi) \sin \left(\frac{2 \pi n}{\omega_0} \right) \right\}$$

- We can obtain acyclic torque expression of the flywheel using (r, ψ) :

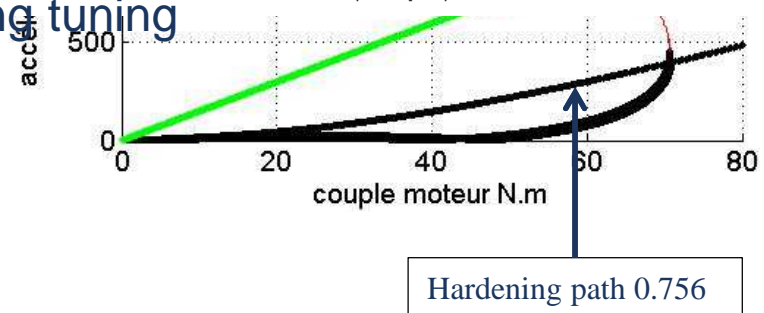
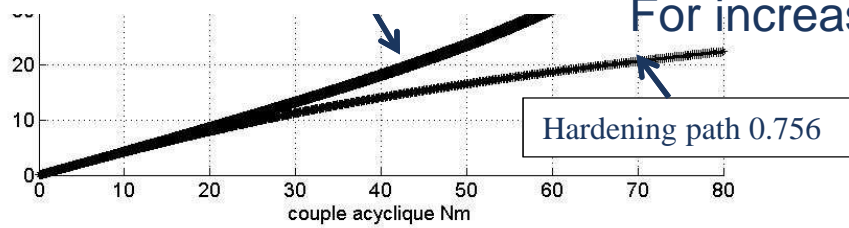
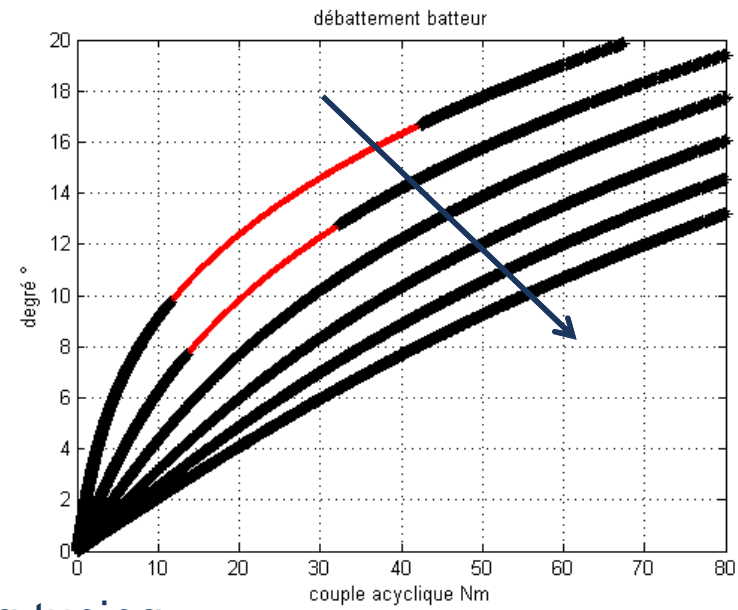
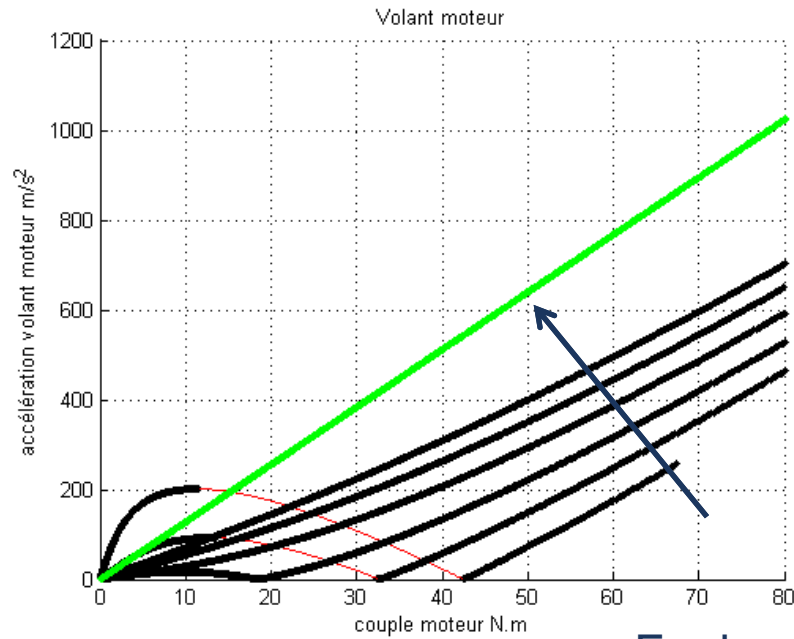
$$\begin{aligned} \tilde{\Gamma}^2 = & \frac{(\omega_0^2 - n^2)^2}{2 \omega_0} \left\{ \frac{\pi}{4 n^2} \sin^4 \left(\frac{n \pi}{\omega_0} \right) K \omega_0^2 \sin^2 \left(\frac{2 n \pi}{\omega_0} \right) \left[\gamma + K (2 \beta^2 + 3 \alpha \delta) \omega_0^2 \right] - 2 \omega_0^2 r \right\} \\ & + \frac{\pi^2}{4 \omega_0^2 (n^2 - \omega_0^2)} \left\{ \left[-4 r (1 + K \omega_0^2 \sin^2(\psi)) \sin^3 \left(\frac{n \pi}{\omega_0} \right) + K (2 \beta_0^2 + 3 \alpha \delta) \omega_0^2 \left(\frac{2 \pi n}{\omega_0} \right)^2 \right]^2 \right\} \end{aligned}$$

DESIGN OF THE HARMONIC ABSORBERS

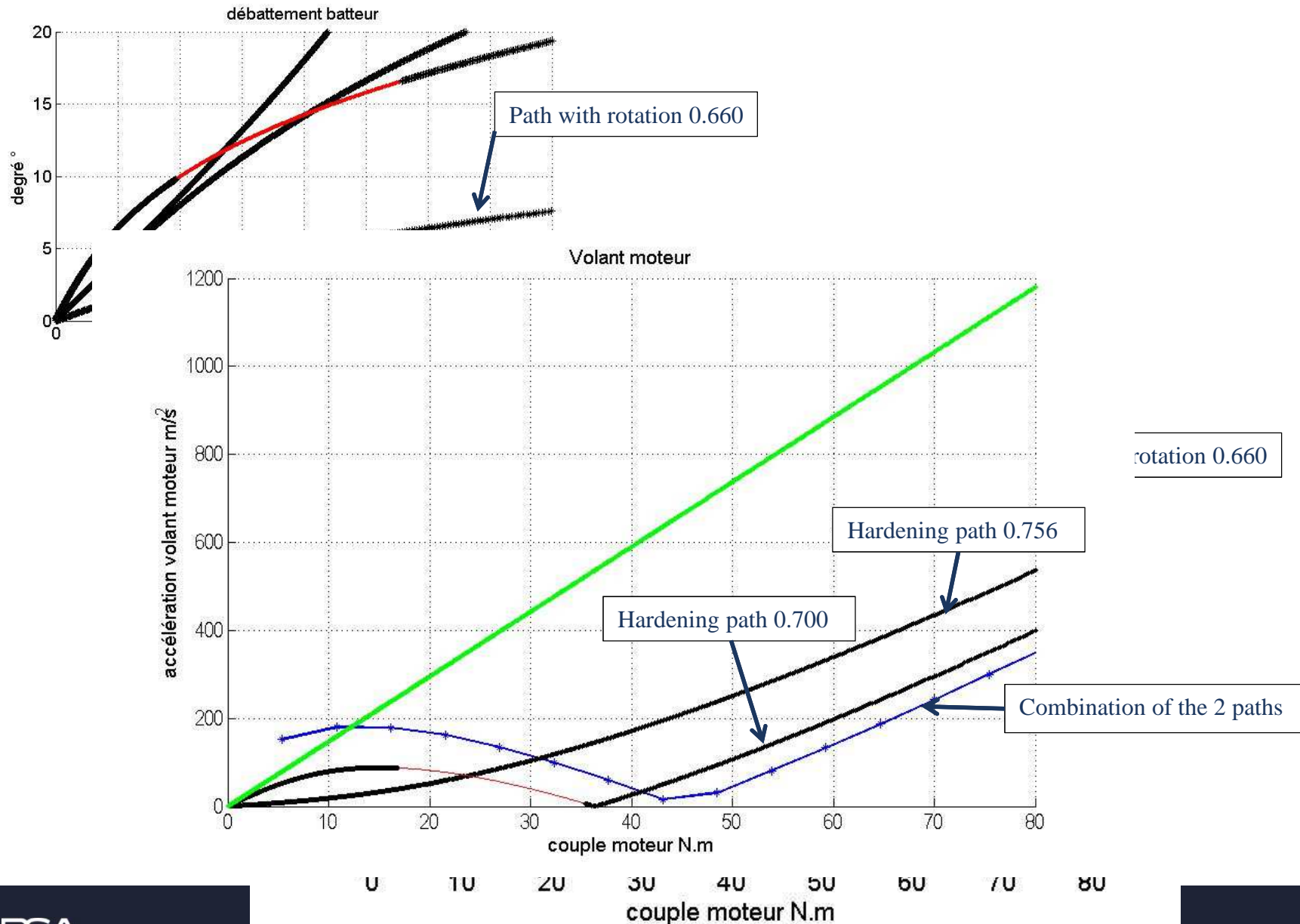
- Recall for circular paths



DESIGN OF THE HARMONIC ABSORBERS



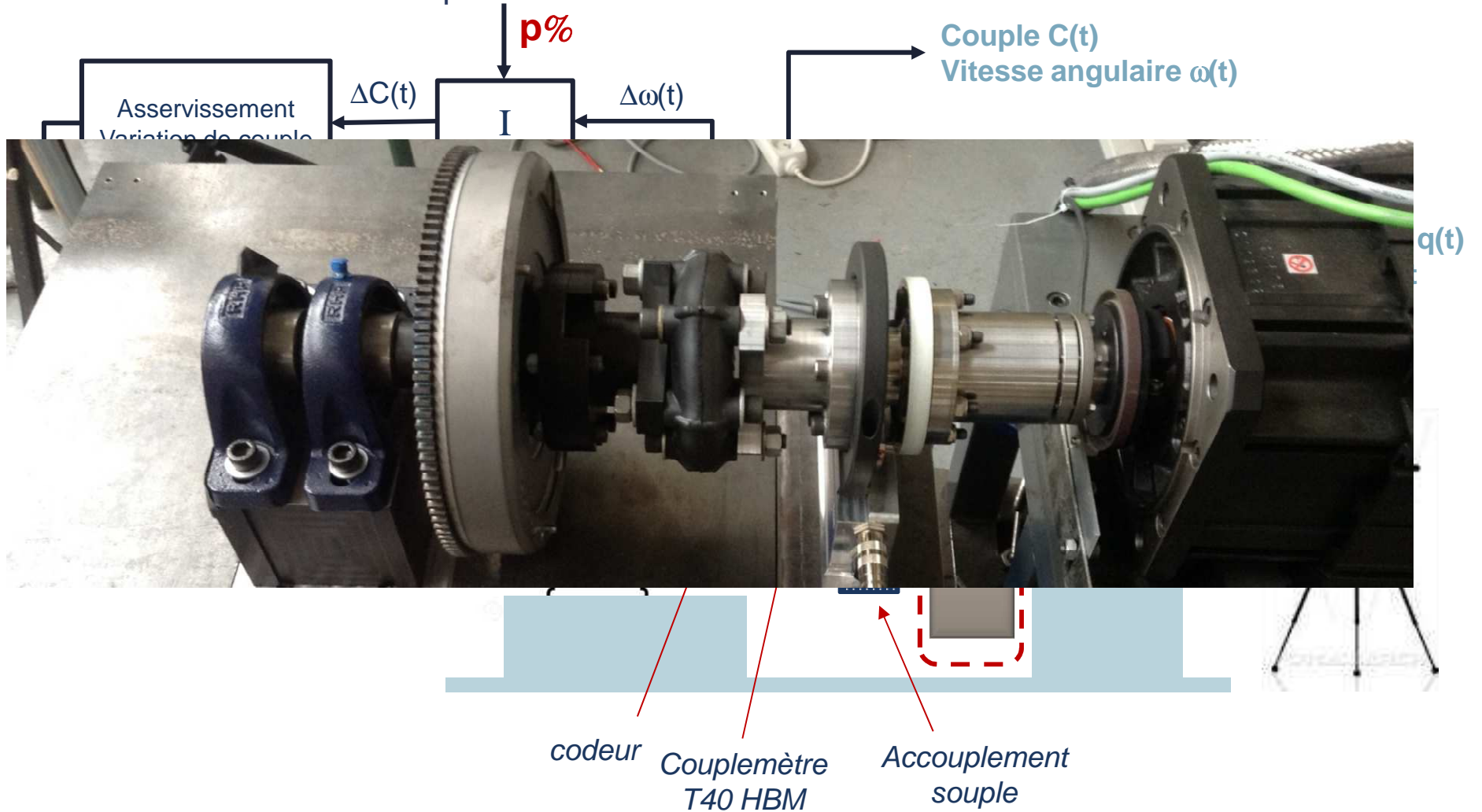
DESIGN OF THE HARMONIC ABSORBERS



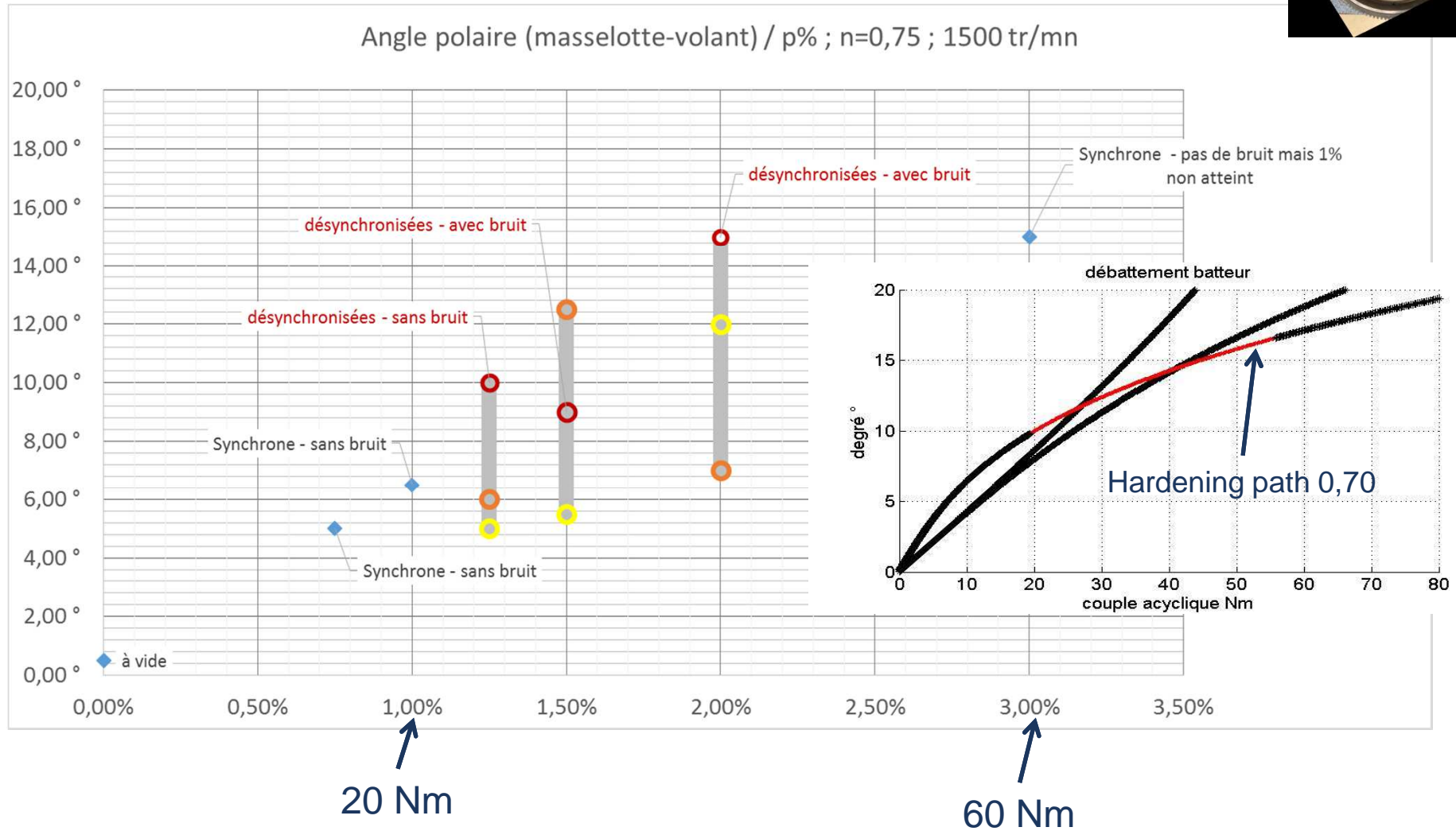
ORGANIC BENCH VALIDATION OF THE CONCEPT



- A test bench was developed in collaboration with ENSAM @ Aix en Provence:

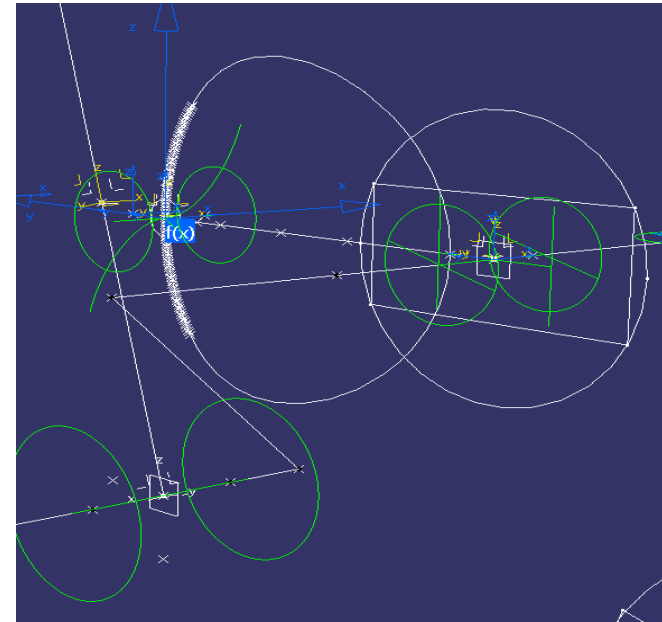
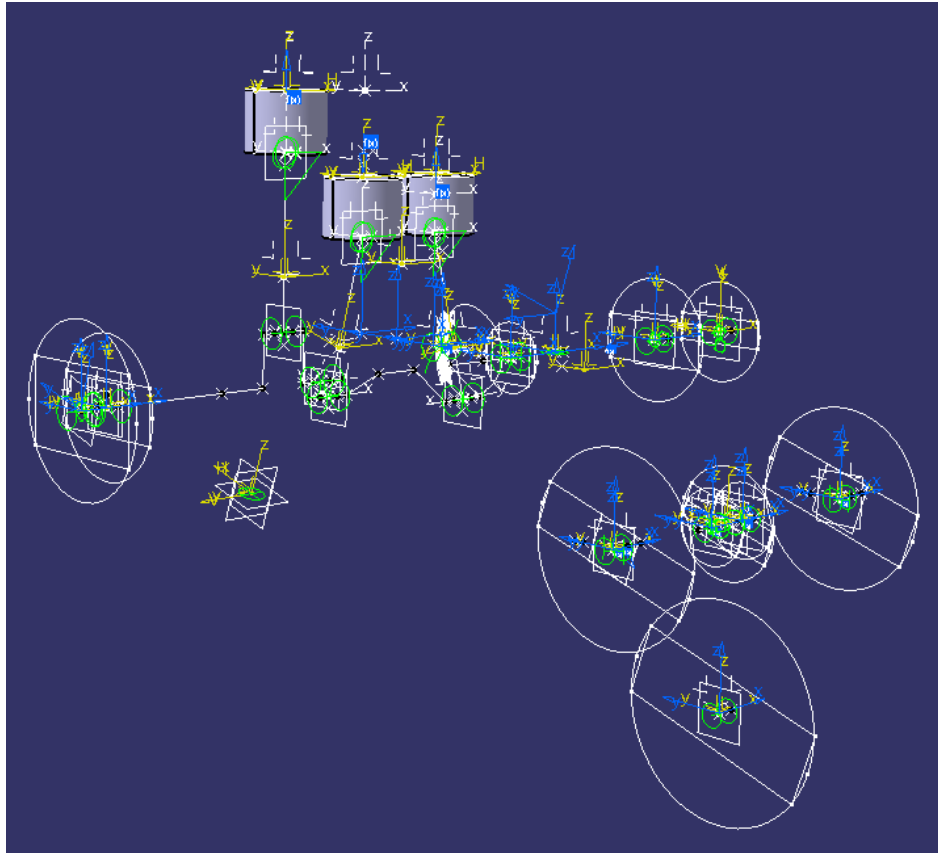


ORGANIC BENCH VALIDATION OF THE CONCEPT



SIMULATION OF THE POWERTRAIN WITH PENDULAR ABSORBERS

- Further simulations integrated in a vehicle powertrain synthesis model



ROTOR WITH PENDULARS TUNED TO H1,5 AND H0.075

- Here we need to consider two acyclic excitation of the rotor

$$T(\theta) = T_0 + T_n \sin(n\theta) + T_{2n} \sin(2n\theta + \phi)$$

- Dynamics equations are not averaged and we use MANLAB 2.0 to explore branches of steady solutions of the system.

$$z_i'' + n^2 z_i = \varepsilon \left(2\gamma_0 - 2n_i^2 - 2n_i^2 \sigma_i z_i - \tilde{\mu}_{ai} z_i' - \frac{1}{N} \sum_{j=1}^N n_j^2 z_j - \tilde{\Gamma}_n \sin(n\theta) - \tilde{\Gamma}_{2n} \sin(2n\theta + \phi) \right)$$

$$n = n_i = \tilde{n}_i + \varepsilon \sigma_i \quad i = 1 \dots N_1$$

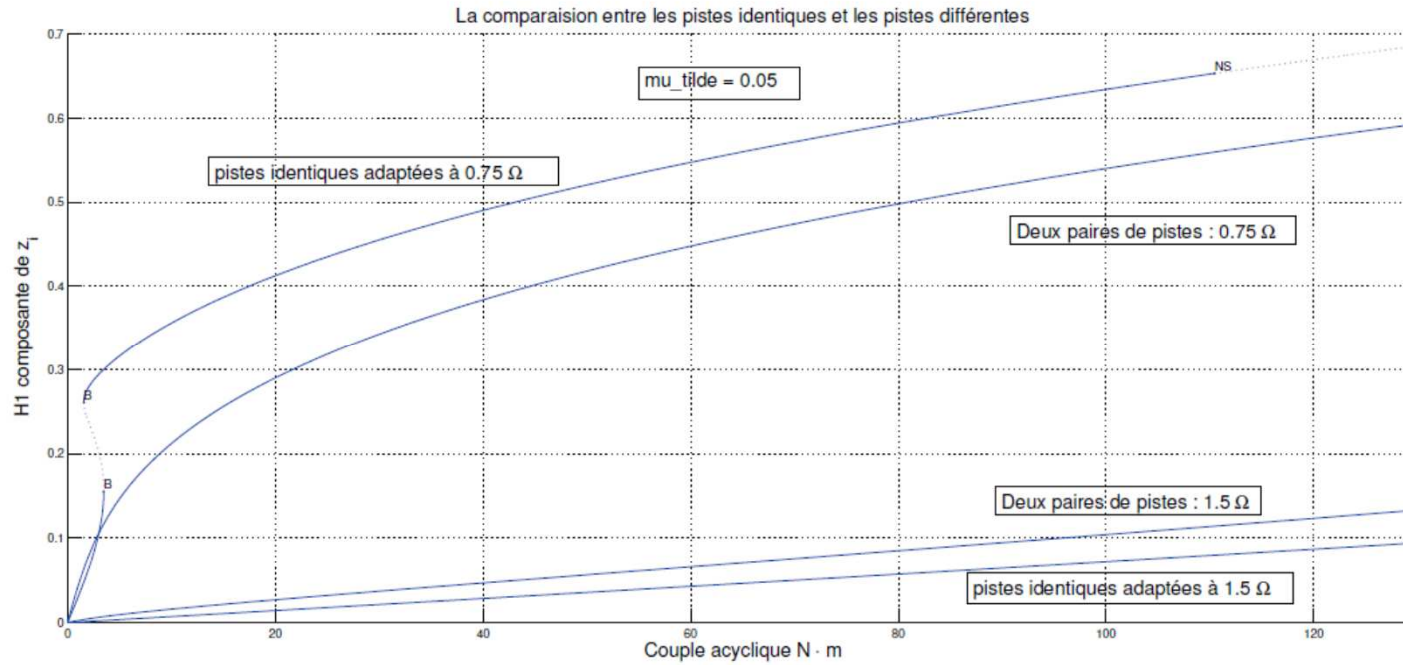
$$2n = n_i = \tilde{n}_i + \varepsilon \sigma_i \quad i = N_1 + 1 \dots N$$

$$\sum_{i=1}^N m_i = m, \quad \lambda m = \sum_{i=1}^{N_1} m_i, \quad (\lambda - 1)m = \sum_{i=N_1+1}^N m_i$$

- The aim is
 - To understand the coupling between pendulars and possible multiple resonances,
 - To give the efficient inertia partition λ for the two paths considered.

ROTOR WITH PENDULARS TUNED TO H1.5 AND H0.75

- First results



CONCLUSION AND PERSPECTIVES

- Hardening paths with rotations seems to be convenient for the automotive applications for low harmonics dissipation...
- The analytical and numerical simulations are validated by the first measures on a organic bench
- We need :
 - To explore asynchronous motion of the pendular absorbers : following the branches of solutions after bifurcations, determine internal resonances seen on the bench...
 - To take into account mechanical stop effect on the dynamics;
 - To further explore the adaption of 2 different tuning paths on the rotor
- We should use more complex absorbers creating a metamaterial composed of different paths adapted to n , $2n$, $3n$ harmonics...