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## Targeted Energy Transfer in Automotive Powertrains

Ahmed Haris, Eliot Motato, Stephanos Theodossiades, Alexander F. Vakakis, Lawrence A. Bergman, D. Michael McFarland and Ben Struve

*Abstract:* Torsional oscillations generated by the internal combustion engine induce various NVH phenomena in the drivetrain system, one being transmission rattle. Palliative devices such as the clutch predampers or dual mass flywheel have been used to mitigate these NVH phenomena. However, usually these devices are effective over a limited range of frequencies, and not so for broadband transient phenomenon, such as any impulsive actions. This paper considers the Targeted Energy Transfer (TET) method to mitigate torsional vibrations in automotive powertrains. TET is a concept which attempts to direct the mechanical (vibration) energy (in a nearly irreversible manner) from a source (primary system) to a strongly nonlinear attachment (Nonlinear Energy Sink – NES), where it is absorbed, redistributed and/or dissipated. In contrast to the classical powertrain palliative methods, NES should be capable of operating over a broader band of frequencies (with the additional aim of being lightweight and compact). Although the TET concept has been extensively studied for translational systems, there is a dearth of studies for rotational (torsional) ones. In the present work, preliminary parametric studies are performed on a reduced automotive powertrain model, incorporating a NES attachment. The NES parameters, including nonlinear stiffness, viscous linear damping and inertia are varied in order to determine NES effects on engine order (EO) vibration.

### 1. Introduction

Vehicle fuel consumption and emissions are the foremost powertrain development objectives. The EU automotive emissions regulation currently imposes a limit of 130 g/km for any manufacturer's fleet of vehicles. This limit is expected to be reduced to 95 g/km by 2020. To meet this stringent limit engine downsizing, light weight and compactness are seen to have the positive effect on fuel consumption and emissions. On the other hand, there is the trend to improve upon output power-to-weight ratio as desired by customer base, thus leading to increased powertrain NVH, mostly of torsional oscillations, owing to the inherent combustion signature and inertial imbalances of piston-connecting rod-crankshaft system [8]. Body boom, clutch judder, axle whine, driveline clonk and gearbox rattle are phenomena generated by the driveline oscillations across a broadband of NVH response, but all initiated by engine vibration or impulsive action.

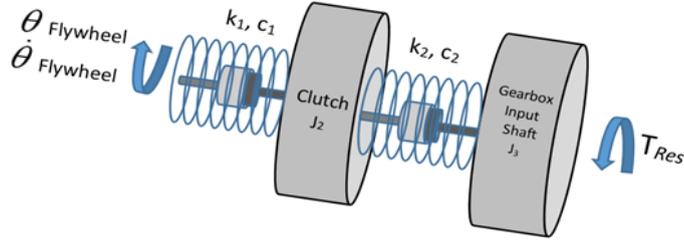
Traditionally, vehicle manufacturers have implemented clutch predampers to control powertrain vibrations, but unacceptable oscillation levels still transmit through to the driveline system. The Dual Mass Flywheel (DMF) [1] and the DMF with centrifugal pendulum vibration absorbers (CPVA) [15] are additional palliatives used to minimize vibrations at specific

frequency ranges. Nevertheless, in modern engines, the vibration energy is distributed over a broad range of frequencies, reducing the effectiveness of the DMF and CPVA as passive vibration absorbers. This paper investigates the use of Targeted Energy Transfer (TET) as a way to attenuate broadband vibrations in powertrains.

TET is a concept where the vibration energy is directed from a source to a receiver in a uni-directional, nearly-irreversible manner [11]. From an engineering perspective, Gendelman et al. [2] and Vakakis et al. [12] have published studies of TET in two and three-degree-of-freedom (DOF) systems. The literature includes a plethora of publications dedicated to the understanding of TET (“energy pumping”). Vakakis et al. have studied the Non-linear Normal Modes (NNMs) [13] to classify the interaction of the Nonlinear Energy Sink (NES – non-linear attachment) with the modes of the primary linear system. Jiang et al. [4] studied the effect of NES on the steady-state dynamics of a weakly coupled system. Panagopoulos et al. [7] studied the transient resonance interactions of a finite number of linear oscillators coupled to a NES. Tsakirtzis et al. [10] studied TET transient resonance interactions. Kerschen et al. [5] conducted parametric studies to understand the dynamics of energy pumping. McFarland et al. [6] performed investigations in structures with grounded and ungrounded non-linear attachments. All the above studies have focused on translational systems, subjected to (mainly) impulsive inputs.

The first study on implementing a NES in torsional systems considers the application of a discrete torsional NES to stabilise a drill-string system [14]. Gendelman et al. [3] studied, numerically and experimentally, the dynamics of an eccentric rotational NES mounted inside a linear oscillator. Sigalov et al. [9] expanded the study of Gendelman et al. by analysing the resonance captures and the NNMs of the system. In all these works, impulsive excitation has been applied in the primary system. Nevertheless, transient excitations are a common input in the automotive powertrains.

This work proposes a methodology that uses the TET method (NES absorbers) to reduce the broadband vibrations generated in vehicle powertrains, subjected to transient excitations. The powertrain considered is a Front Wheel Drive (FWD) transaxle system coupled to a three cylinder engine. The NES parameters are numerically tuned to minimise the gearbox input shaft acceleration produced by the different engine order (EO) harmonics. The NES proposed is passive in nature and lightweight, and can be used in compact spaces. It is also able to operate over a broad range of frequencies.



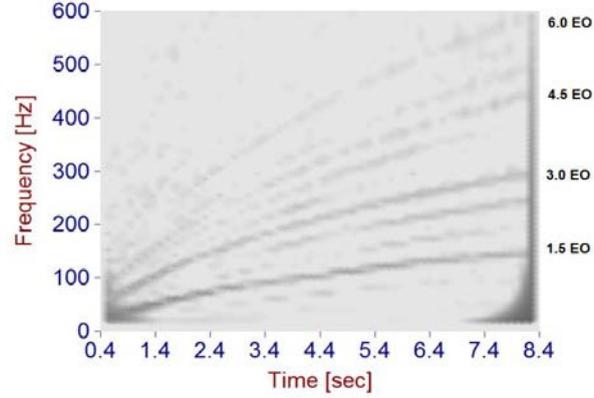
**Figure 1:** Schematic of the Reduced Powertrain Model

## 2. Powertrain Model Validation

A generic powertrain configuration comprises the engine, flywheel, clutch, gearbox, differential, transaxle half-shafts and tyres. Modelling the entire system can be computationally intensive, thus a simplified two DOF lumped parameter model (Figure 1) is used to introduce the proposed procedure. The reduced model includes the clutch assembly inertia ( $J_2$ ) and the gearbox input shaft inertia ( $J_3$ ). The effect of the other components located after the input shaft is represented by the resisting torque ( $T_{Res}$ ) transferred to the gearbox input shaft. The resisting torque is modelled as a function of aerodynamic drag and tyre-road rolling resistance. The effect of engine torque on the clutch disc is represented by a torsional signal applied through a spring with stiffness  $k_1$  and a damper with coefficient  $c_1$ . The applied torque is a function of the flywheel position and velocity.

Experimental data obtained from a vehicle equipped with a similar powertrain was used to validate the simplified numerical model. The 3-cylinder 1.0l engine produces 170 Nm peak torque. The vehicle uses a clutch torsional damper with a Single Mass Flywheel (SMF). The angular rotations of the shafts are measured using sensors located at the flywheel and gearbox input shaft for the vehicle operating at partial engine throttle (the engine was swept through the whole operating speed range with 25% throttle input).

The Continuous Wavelet Transform (CWT) was used for the analysis of the experimental, non-stationary angular velocities using AutoSignal commercial software. The CWT analysis highlights the 1.5, 3.0, 4.5 and 6.0 EO frequencies as expected. This particular engine has the 1.5 EO as dominant because combustion occurs three times over two crankshaft revolutions. The obtained frequency domain response is shown in Figure 2.



**Figure 2:** Experimental Input Shaft CWT in 1<sup>st</sup> gear with 25% open throttle

The equations of motion of the simplified model are:

$$\begin{bmatrix} J_2 & 0 \\ 0 & J_3 \end{bmatrix} \begin{bmatrix} \ddot{\theta}_2 \\ \ddot{\theta}_3 \end{bmatrix} + \begin{bmatrix} c_2 + c_1 & -c_2 \\ -c_2 & c_2 \end{bmatrix} \begin{bmatrix} \dot{\theta}_2 \\ \dot{\theta}_3 \end{bmatrix} + \begin{bmatrix} k_2 + k_1 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \begin{bmatrix} \theta_2 \\ \theta_3 \end{bmatrix} = \begin{bmatrix} k_1 \theta_1 + c_1 \dot{\theta}_1 \\ -T_{Res} \end{bmatrix} \quad (1)$$

where  $\dot{\theta}_1$ ,  $\dot{\theta}_2$ , and  $\dot{\theta}_3$  are the angular velocities of the flywheel, clutch disc and gearbox input shaft, respectively. The parameter values of torsional linear stiffness  $k_1$ ,  $k_2$  and inertias  $J_2$  and  $J_3$  were provided by industrial partners.

**Table 1:** Model parameters

Inertia [kgm <sup>2</sup> ]	Stiffness [Nm/rad]	Damping
$J_2 = 0.001$	$k_1 = 1000$	$\zeta_1 = 0.8$
$J_3 = 0.003$	$k_2 = 15000$	$\zeta_2 = 0.5$

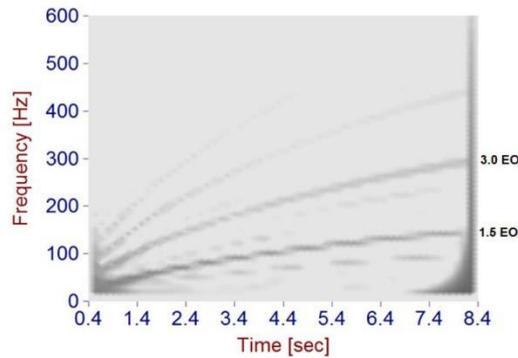
The coefficients  $c_1$  and  $c_2$  were obtained using Caughey's method;  $\varphi$  is the modal matrix obtained through solution of the generalised eigenvalue problem, thus:

$$[C] = \begin{bmatrix} J_2 & 0 \\ 0 & J_3 \end{bmatrix} \varphi \begin{bmatrix} \zeta_1 & 0 \\ 0 & \zeta_2 \end{bmatrix} \varphi^T \begin{bmatrix} J_2 & 0 \\ 0 & J_3 \end{bmatrix} \quad (2)$$

The damping ratios  $\zeta_1$  and  $\zeta_2$  corresponding to the 1<sup>st</sup> and 2<sup>nd</sup> natural frequencies of the system were fine tuned to obtain a near identical time and frequency domain response.

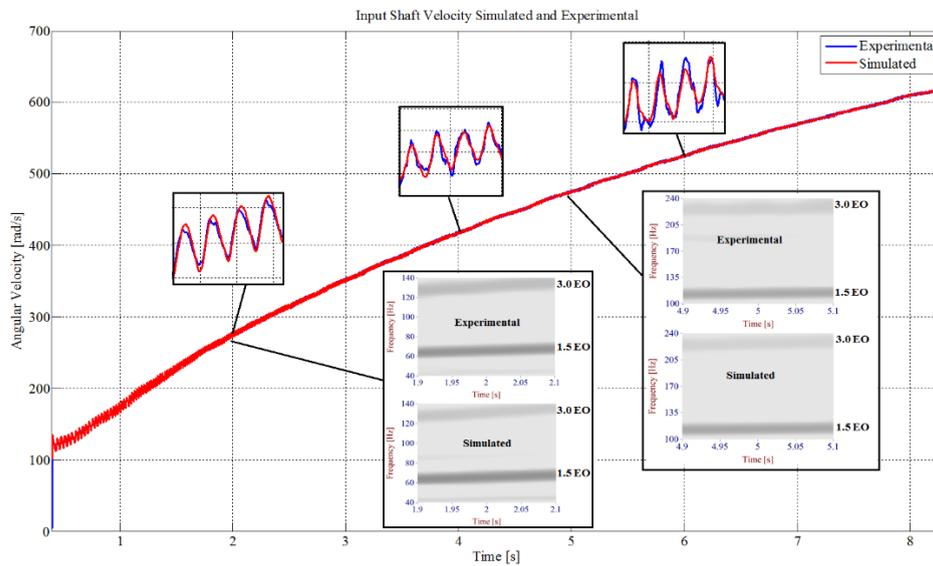
A Matlab/Simulink model, representing Eq. (1) is made, and the damping coefficients obtained through Eq. (2) are incorporated. Several combinations of damping ratios were evaluated with the objective of achieving a time and frequency response similar to the experimental data response, whilst

keeping  $\zeta < 1$ . The validation was conducted using the 1<sup>st</sup> gear engaged at 25% throttle manoeuvre. The damping ratios which resulted in a near identical system response were  $\zeta_1 = 0.8$  and  $\zeta_2 = 0.5$  with the corresponding natural frequencies being  $f_1 = 62$  Hz, and  $f_2 = 800$  Hz.

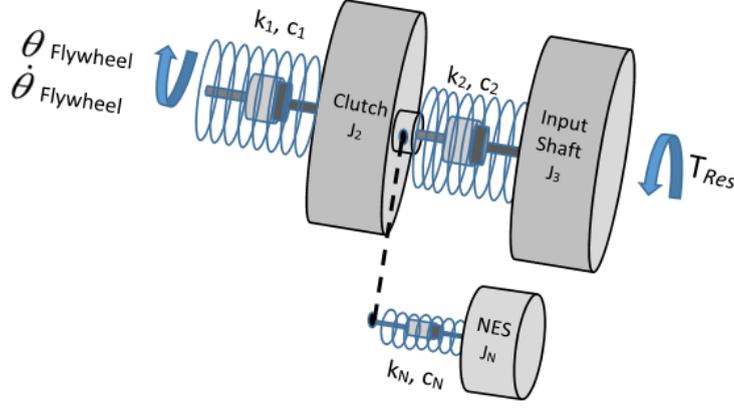


**Figure 3:** Simulated Input Shaft CWT response for 1<sup>st</sup> gear at 25% open throttle

CWT analysis was performed using the numerical data obtained. The results are shown in Figure 3. Indeed, the simulated CWT is similar to the corresponding experimental CWT displayed in Figure 2. A time domain plot of the gearbox input shaft angular velocity comparing the experimental and simulated responses is shown in Figure 4. The time domain CWT insets show very good correlation between the simulated and the experimental responses.



**Figure 4.** Simulated and experimental results comparison in frequency and time domains



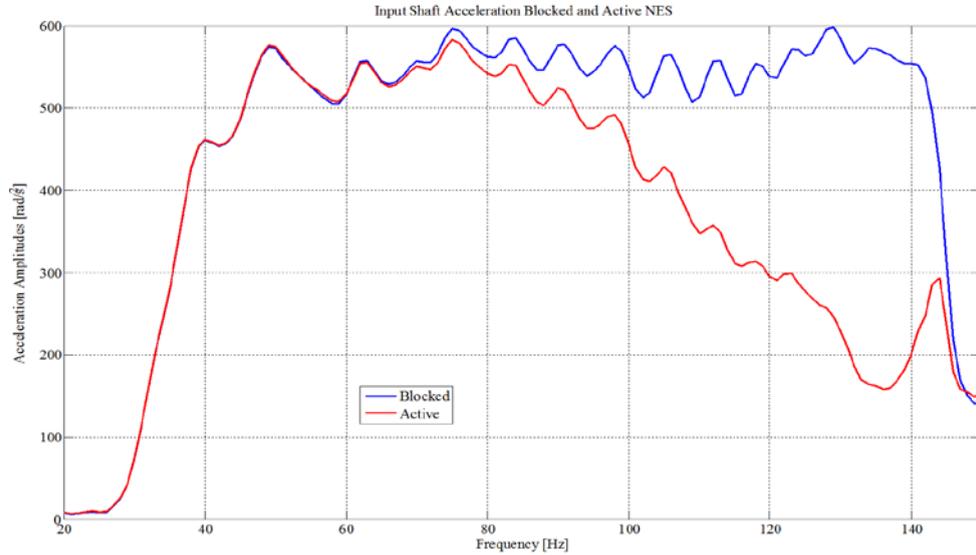
**Figure 5:** Powertrain model with NES attachment

### 3. Powertrain system with coupled NES attachment

The simplified powertrain model described in Eq. (1) was modified to incorporate a Non-linear Energy Sink (NES) with inertia  $J_N$  as shown in Figure 5. The NES is coupled in parallel with the clutch disc through an essentially nonlinear torsional cubic spring with stiffness constant  $k_N$  and a linear viscous damper with damping coefficient  $c_N$ . The NES is considered to be mounted on the spline coupling the clutch with the gearbox input shaft. The differential equations of motion describing the dynamics of the powertrain coupled with the NES are given by

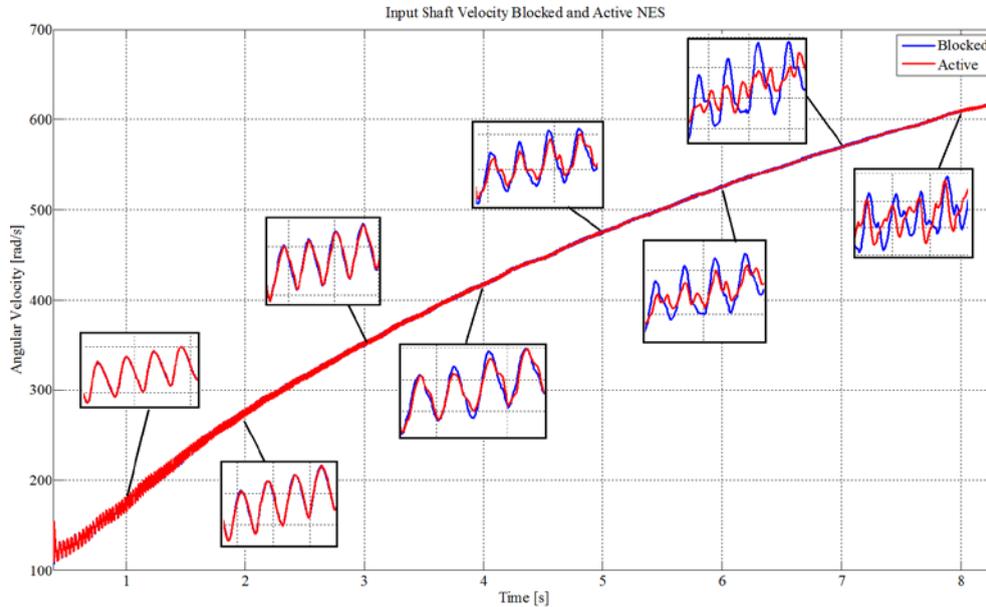
$$\begin{bmatrix} J_2 & 0 & 0 \\ 0 & J_3 & 0 \\ 0 & 0 & J_N \end{bmatrix} \begin{bmatrix} \ddot{\theta}_2 \\ \ddot{\theta}_3 \\ \ddot{\theta}_N \end{bmatrix} + \begin{bmatrix} c_1 + c_2 + c_N & -c_2 & -c_N \\ -c_2 & c_2 & 0 \\ -c_N & 0 & c_N \end{bmatrix} \begin{bmatrix} \dot{\theta}_2 \\ \dot{\theta}_3 \\ \dot{\theta}_N \end{bmatrix} + \begin{bmatrix} k_2 + k_1 & -k_2 & 0 \\ k_2 & k_2 & 0 \\ 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \theta_2 \\ \theta_3 \\ \theta_N \end{bmatrix} = \begin{bmatrix} k_1 \theta_1 + c_1 \dot{\theta}_1 - k_N (\theta_2 - \theta_N)^3 \\ -T_{Res} \\ k_N (\theta_2 - \theta_N)^3 \end{bmatrix} \quad (3)$$

Since the 1.5 EO harmonic carries most of the vibration energy of the powertrain, the performance of the NES is measured based on the reduction of the amplitude of this harmonic. This is analysed by comparing the gearbox input shaft acceleration of the 1.5 EO when (a) the powertrain is operating with the NES (*active*) versus (b) the powertrain operating with the NES inertia simply added to the clutch inertia (*locked*). The system response time histories are estimated from the dynamic model. The Matlab command *Pwelch* is then used to obtain the power spectral density (PSD) of the gearbox input shaft acceleration for *locked* and *active* systems using Welch's overlapped segment averaging estimator. To obtain the plot shown in Figure 6, the following method was used. The acceleration time histories for the gearbox input shaft were used as the input to the *Pwelch* command. The resultant PSD obtained contained all the engine harmonics with the units  $\text{rad}^2/\text{s}^4/\text{Hz}$ . Therefore, to obtain the 1.5 EO acceleration, the product of PSD x Frequency was square rooted i.e.  $1.5 \text{ EO} = \sqrt{\text{PSD} \times \text{Frequency}}$ , where frequency is the frequency of 1.5 EO harmonic.



**Figure 6:** 1.5 EO acceleration amplitudes for active and locked NES

A series of simulations were performed for ranges of  $k_N$ ,  $J_N$  and  $c_N$  using the numerical model. It is noted that significant acceleration reduction is achieved for  $J_N = 11\%$  of the gearbox input shaft inertia,  $k_N = 2 \times 10^6$  Nm/rad<sup>3</sup>, and  $c_N = 0.001$  Nms/rad. The corresponding 1.5 EO acceleration amplitudes in the frequency domain for both systems (*active* and *locked* NES) are shown in Figure 6. The frequency range where substantial reduction in the 1.5 EO harmonic occurs is 80 - 150 Hz. It is also possible to observe a reduction in the amplitude of oscillations of the angular velocity (Figure 7) during the time (and corresponding frequency range) in the manoeuvre, where the NES operates effectively, absorbing the oscillations of the primary system. The amplitude of oscillations in the velocity is clearly linked to the magnitude of the 1.5 EO harmonics.



**Figure 7.** Input Shaft Angular velocity for systems with active and locked NES

#### 4. Conclusion

This work studied the effect of implementing a NES vibration absorber to an automotive powertrain for passively absorbing torsional oscillations. From the results obtained it can be concluded that the NES induces an energy transfer mechanism, which redistributes the energy in the powertrain; as a result substantial reduction in vibrations is observed at the gearbox input shaft over a broad frequency range. Intensive simulations were conducted for different combinations of NES parameters to identify those that lead to reduction of the 1.5 EO harmonics of the gearbox input shaft. Furthermore, it appears that the range of NES operation is dependent on the input energy applied to the system, unlike the conventional palliatives which are tuned to specific engine frequencies.

#### 5. Acknowledgments

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