Concept selection for clutch nonlinear absorber using PUGH matrix

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Abstract: Noise, vibration and harshness (NVH) refinement as well as fuel efficiency and reduced emission levels are the key objectives in modern powertrain engineering. There is an increasing plethora of NVH concerns associated with the underlying high output power-to light weight and compact concept in powertrain engineering. These phenomena contribute to a broad-band vibration response from low frequency rigid body oscillatory responses to high frequency impulsive actions. Various phenomena are briefly described and the importance of their attenuation through palliative measures emphasised. The role of non-linear oscillators as energy sinks over a broader range of responses is also described. A predictive model is presented. Predictive analysis shows effective action of non-linear energy sinks.

A feasible design of a NES absorber in an automotive powertrain is constrained by multiple operating requirements such as temperature, available space, reliability and other attributes, requiring an objective analytic-subjective experiential method to arrive at an optimum solution within a series of plausible alternatives. A methodology based on the PUGH matrix approach is presented.

Keywords— Nonlinear Energy Sink, Targeted Energy Transfer, Clutch Vibration, Automotive Drivetrain, PUGH matrix.

1-Introduction

Engine torsional vibration, often referred to as engine order vibration is one of the main sources of a plethora of noise, vibration and harshness (NVH) phenomena in vehicles, particularly the powertrain system [1]. Engine order vibration corresponds to the power torque fluctuations in the combustion process and inertial imbalances in the piston-connecting rod-crankshaft sub-system [2]. These sources of vibration are a function of crankshaft speed, thus referred to as engine order vibration. The lower spectral content of engine vibration is dominated by these sources of vibration and vary according to the engine cycle (2-stroke versus 4-stroke), the number of cylinders in an engine, sources of out-of-balance as well as system compliances, among a host of other measures.

Key drivers in powertrain development are improved fuel efficiency, reduced emissions and NVH refinement. These have been predominantly addressed by the concept of high output power-to weight ratio compact systems. Therefore, engine downsizing, incorporating boosting technologies are now commonplace as well as an increasing use of other technologies such as cylinder deactivation and stop-start in congested urban driving. However, increased power-to-weight ratio, and particularly use of light weight, flexible, but durable components whilst ensuring structural integrity has led to a host of NVH concerns [2, 3]. With the modern relatively more flexible piston-connecting rod-crankshaft sub-system compared with the more rigid traditional equivalents, the spectrum of vibration due to the aforementioned sources of engine order vibration contains a broader spectrum, including a plethora of torsional-deflection contributions at all multiples of half-engine order [2]. Kushwaha et al [4] showed these exacerbated conditions using flexible multi-body dynamics. It was shown that crankshaft flexibility exacerbates the inherent imbalance, leading to conical whirl motion of the flywheel, which during the clutch pedal movement causes a clutch noise and vibration concern, known as the clutch whoop [5,6]. A rubber mass damper, called Diehl fix was traditionally used to attenuate clutch whoop phenomenon, which took up a considerable space in the powertrain set up as well as adding to the cost of the clutch system. Alternative, solution to attenuate this problem was later reported, dealing with clutch cover stiffness with detailed multi-body analysis, giving the opportunity of eliminating the Diehl-fix, thus spacing on the package space, weight and manufacturing costs [7]. This example serves the purpose of demonstrating the need for detailed analysis, using new torsional and axial models to arrive at an optimal solution to meet the demands of wide-ranging criteria. This has been achieved through development of models simulate the complex interactions and represent the variations involved in creating a high quality NVH result.

Other key NVH concerns also employ costly palliations, which are often not the optimum. They include gear rattle, which occurs as the result of transmission of engine order vibration through the gearbox input shaft to the loose (unengaged gears) of the transmission system. This problem is exacerbated with increased engine, particularly with diesel engine vehicles (higher torques) and flexible light-weight crankshafts. At lower overall engine noise (at idle or low speed crawling), the noise and vibration is particularly poignant [8-11]. However, the problem is also manifested under various driving conditions as well, and termed as creep, drive or over-run rattle, depending on the combination of engine load, throttle application and speed [12-14]. Throughout the years various palliative measures have been used to attenuate the effect of transmission rattle, including clutch pre-dampers, various slip devices and dual mass flywheel [15, 16]. Essentially, these devices are tuned to the significant engine order contribution, depending on the engine type, to dissipate its excess energy content. For example, for 4-cylinder, 4-stroke engines the principal engine order response is at the second engine order and a few of its immediate higher order harmonics. Therefore, the arc spring stiffness is so chosen to tune and dampen the amplitude of vibration at these frequencies through oscillation of the secondary mass. Therefore, the effect of dual mass flywheel is focused to a narrow band of frequencies and is only partially effective over a broader range of responses, particularly the higher frequencies under transient conditions [17], typical of impulsive/impact conditions such as driveline clonk [18]. Hence, the DMF has become the palliative device of choice for attenuation of rattle at the expense of cost, significant added inertia [19] and package space. The arc spring characteristics is quasi-linear, thus a trend towards more effective broad-band attenuation, using a non-linear damper would appear to be a rational option.

Non-linear energy sinks (NES) operate on the principle of targeted energy transfer (TET). The vibration energy from the primary linear system is transferred in a nearly irreversible manner to the non-linear absorber, where it is absorbed or re-distributed to higher modes or dissipated locally through structural damping [20]. Numerous studies have been reported in literature on the TET working principle, and the design and parameter optimisation of the non-linear absorber attached to a linear translational system [21-25]. However, there have been limited studies reported on the performance of NES in a rotational primary linear system. Recently, Haris et al. [26] and Saava et al. [27] reported implementation of NES for attenuating driveline torsional vibrations under transient and idling conditions. It was demonstrated that a relatively light NES has the ability to attenuate vibrations over a broad range of frequencies with the potential of optimising the same to operate over a desired frequency range.

2-Drivetrain dynamic model, measuring the NES performance

As the main measure in the process of concept selection, the ability of the NES in achieving the required attenuation should be established through numerical analysis. A dynamics model, taking into account the driveline equipped with the NES is developed. The model is for a front wheel drive (FWD) powertrain equipped with a three cylinder turbocharged engine. This specific powertrain incorporates a solid mass flywheel (SMF) with a clutch damper for attenuating the undesired torsional vibrations. In order to reduce the model complexity, a reduced model is considered as shown in Fig 1. The model incorporates the clutch and the transmission input shaft with the input excitation measured a priori in the form of flywheel velocity $\dot{\theta}_1$ and displacement θ_1.

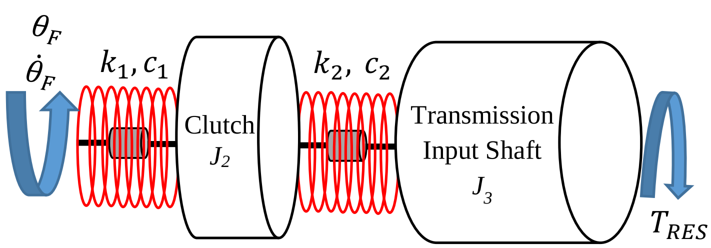

Figure 1: Schematic of the reduced powertrain model

The dynamical equations of motion are:

$$J_2\ddot{\theta}_2 = k_1(\theta_1 - \theta_2) + c_1(\dot{\theta}_1 - \dot{\theta}_2) - k_2(\theta_2 - \theta_3) - c_2(\dot{\theta}_2 - \dot{\theta}_3) \quad (1)$$

$$J_3\ddot{\theta}_3 = k_2(\theta_2 - \theta_3) + c_2(\dot{\theta}_2 - \dot{\theta}_3) \quad (2)$$

where, J_2 is the clutch inertia, J_3 is the transmission input shaft inertia, θ_2, $\dot{\theta}_2$, θ_3 and $\dot{\theta}_3$ are the displacements and velocity of the clutch and transmission input shaft respectively.

The damping coefficients are obtained using the Caughey's method through the relationship:

$$\boldsymbol{C} = \boldsymbol{J} \times \boldsymbol{V} \times \boldsymbol{Z} \times \boldsymbol{V}^{\boldsymbol{T}} \times \boldsymbol{J} \quad (3)$$

where, $\boldsymbol{J}$ is the inertia matrix, $\boldsymbol{V}$ is the mass normalised modal matrix and $\boldsymbol{Z}$ is the diagonal matrix comprising the damping ratios.

As the reduced model does not include other drivetrain components, their effect is included using the resistance torque T_{RES} , which is a function of aerodynamic drag torque T_A and tire rolling resistance torque T_R as:

$$T_A = \frac{1}{2}\rho S C_d V^2 r_w \quad (4)$$

$$T_R = (f_0 + f_2 V^2) F_z r_w \quad (5)$$

where, ρ is the air density, S is the vehicle frontal area, V is vehicle longitudinal velocity, C_d is the aerodynamic drag coefficient, r_w is the tyre rolling radius and F_Z is the vertical tyre load.

Using equations (1)-(5), a Matlab/Simulink model is developed to numerically solve the equations of motion. The predicted response of the transmission input shaft is validated both in time and frequency domains (Fig 2 and 3) against the experimentally measured responses from a vehicle equipped with a similar powertrain configuration. The manoeuvre used is in 2nd gear with 25% open throttle for the entire engine speed range.

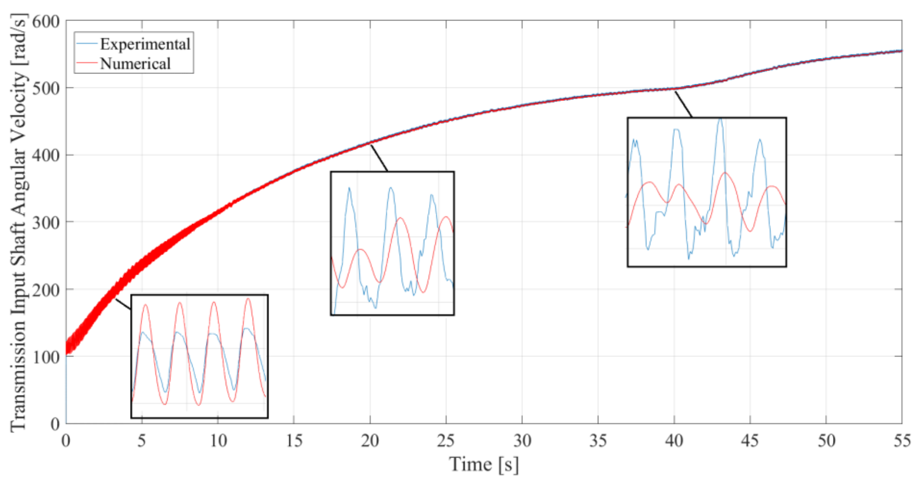

Figure 2: Comparison of the transmission input shaft velocity for numerical and experimental time histories

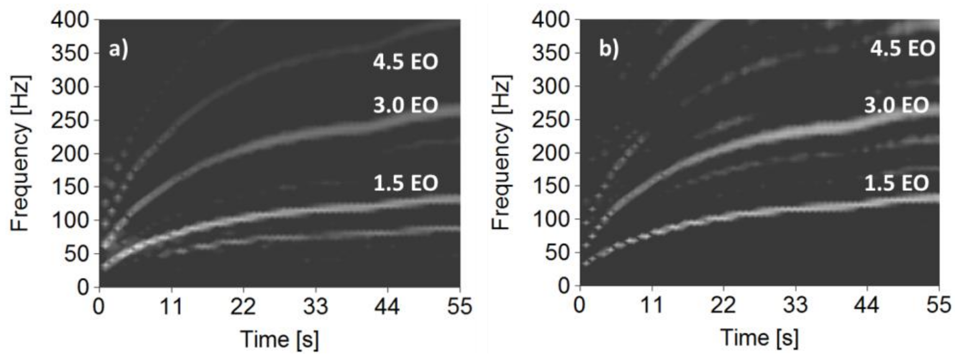

Figure 3: Continuous wavelet transform for: a) numerical and b) experimental transmission input shaft acceleration

The reduced powertrain dynamic model is modified to incorporate a single cubic non-linear absorber as shown in Fig 4.

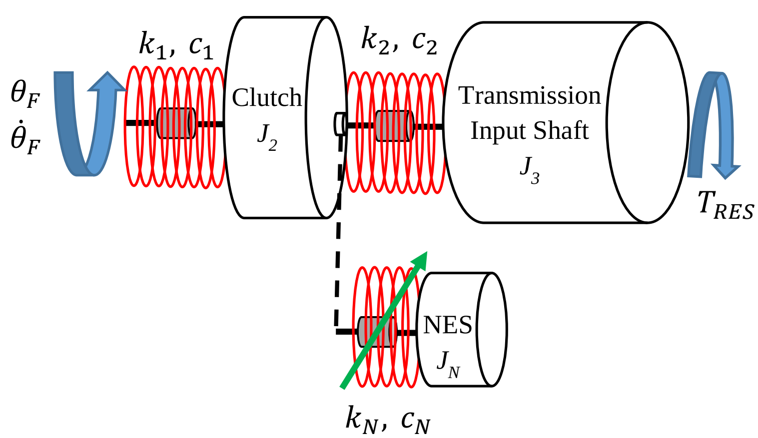

Figure 4: Schematic of the reduced powertrain model with a cubic nonlinear absorber

The equations of motion for the model in figure 4 are:

$$J_2\ddot{\theta}_2 = k_1(\theta_1 - \theta_2) + c_1(\dot{\theta}_1 - \dot{\theta}_2) - k_2(\theta_2 - \theta_3) - c_2(\dot{\theta}_2 - \dot{\theta}_3) - k_N(\theta_2 - \theta_N)^3 - c_N(\dot{\theta}_2 - \dot{\theta}_N) \quad (6)$$

$$J_3\ddot{\theta}_3 = k_2(\theta_2 - \theta_3) + c_2(\dot{\theta}_2 - \dot{\theta}_3) \quad (7)$$

$$J_N\ddot{\theta}_N = k_N(\theta_2 - \theta_N)^3 + c_N(\dot{\theta}_2 - \dot{\theta}_N) \quad (8)$$

where, J_N is NES inertia, θ_N and $\dot{\theta}_N$ are the displacement and velocity of the NES.

The performance of the non-linear energy sink (NES) is validated by comparing the 1.5 Engine Order (EO) acceleration amplitudes of the transmission input shaft for the system with *locked* and *active* NES. The former is essentially where the NES inertia is added to the clutch inertia, with latter being the system where the NES engages actively with the primary system. The reason for choosing 1.5 EO is because this is fundamental engine order of a three-cylinder engine. Parametric studies are performed to identify the parameters which yield vibration attenuation over a broad range of response frequencies. These are found to be: J_N = 8% of transmission input shaft inertia, k_N = 1.2 x 106 Nm/rad3 and c_N = 0.001 Nms/rad. The corresponding 1.5 EO transmission input shaft acceleration amplitudes are shown in Fig 5. Similarly, the effect can also be observed in the fluctuations of the transmission input shaft velocity as shown in Fig 6.

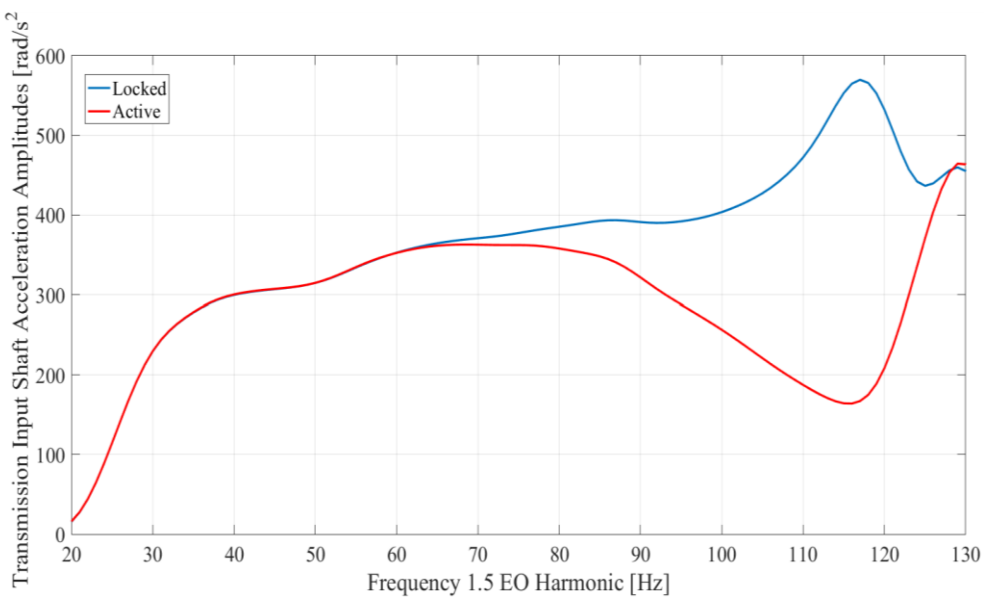

Figure 5: 1.5 EO acceleration amplitudes of transmission input shaft with locked and active NES (J_N = 8% transmission input shaft inertia, $k_N = 5 \times 10^5$ Nm/rad^3, c_N = 0.001 Nms/rad)

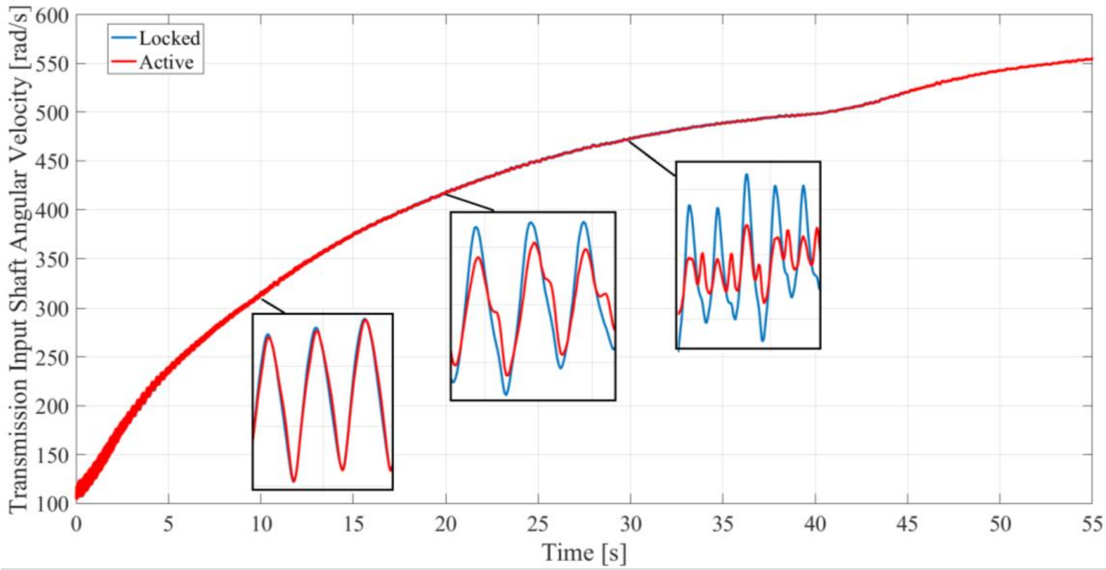

Figure 6: Comparison of transmission input shaft velocity for the systems with locked and active NES

It can be observed that indeed NES is effective in suppression of vibrations over a broad range of frequencies, whilst being compact and lightweight.

3-NES concept selection using PUGH matrix approach

In order to design an NES which can be physically mounted and would operate in a real powertrain system, a specific engineering design approach is required. This approach should take into account different attributes, including the performance of NES. The proposed methodology is based on the PUGH matrix method that is extensively used in industry in the early design selection stage. A PUGH matrix is a conceptual strategy, which compares and quantifies alternative design concepts with the objective of selecting an overall optimum configuration based on stated criteria [28]. Karnjanasomwong and Thawesaengskulthai [29] used the PUGH matrix to study the solder jet bonding process. Similarly, Ul Haque et al. [30] used the PUGH matrix to obtain a solution for the configuration of a hybrid buoyant aircraft. PUGH matrix has several advantages. Firstly, it allows designers to evaluate different alternatives at the conceptual stage for either further investigation or testing and development. It also provides the benefit of obtaining alternative designs in case the chosen design has some problems. Furthermore, it has both time and cost saving advantages for companies because if an unfeasible product is implemented it would have some negative repercussions. Moreover, it not only allows for a broader concept selection, but also enables critical examination by different functional teams of each individual criterion which should be satisfied. The chosen route would then have approval of a broad section of engineers even if their own idea was not selected.

The general process for constructing a PUGH matrix is:

1. Identify and clearly define the design criteria for concept selection.
2. Highlight alternative designs with potential to meet the NES requirement.
3. Rank each individual concept by giving assigning "+" or "-" potential outcome.
4. After scoring each concept, the total score can be determined by summing the number of +'s and –'s.

It is often encountered that the PUGH matrix is not capable of identifying a better design, but it certainly helps in narrowing down the options based on the concepts which best satisfy the stated criteria. If necessary weighting factors for different criteria can be introduced, if the first filter (iteration) does not lead to a clear decision.

For building the PUGH matrix the six alternative NES concepts were considered, see table 1. The seven highlighted selection criteria were evaluated for each of the six concepts and "+" or "-" scores allocated in the matrix cells in accordance. As it can be seen, for a specific criterion, some concepts performed better than the others. In general this difference is due to the type of material used to generate the desired non-linear characteristics. In the last rows of the PUGH matrix the total score through summation "+", "-" or "0" scores are stated. This highlights that the concept 6 is the most appropriate. If the alternative concepts yield similar overall ratings, then additional design or envisaged operational criteria can be included in the matrix.

Table 1: PUGH matrix for concept selection

Criteria - Essential	**Elastomer springs**	**Wire spring**	**Beam spring**	**Vibro-impact**	**Asymmetric vibro-impact**	**Conical spring**
Withstand high temperatures	-	+	+	+	+	+
Withstand harsh environments	-	+	+	+	+	+
Effect on other powertrain components	*Tbd*	*tbd*	*tbd*	*tbd*	*tbd*	*tbd*
Vibration reduction achieved	+	+	+	-	-	+
Dynamic and static balance	-	*tbd*	*tbd*	*tbd*	*tbd*	+
Compact design	*Tbd*	*tbd*	*tbd*	*tbd*	*tbd*	*tbd*
Low structural damping	0	0	0	0	0	0
Sum (tbd)	2	3	3	3	3	2
Sum (0 s)	1	1	1	1	1	1
Sum (+ s)	1	3	3	2	2	4
Sum (- s)	3	0	0	1	1	0
Total	-2	3	3	1	1	4

tbd=to be determined

4-Concluding Remarks

The paper presents the PUGH matrix as a feasible approach to determine the most appropriate and physically implementable design concepts for non-linear torsional absorbers for powertrain applications. The non-linear absorber is used with the objective of suppressing torsional vibrations over a broad range of frequencies. The PUGH matrix approach facilitates the design in the conceptual stage by including several criteria and design concepts. This was found to be a useful process as the NES has not been hitherto been used in vehicular powertrain applications. This also provided the opportunity to evaluate different designs from other applications and judge these against new ideas. The design concepts are evaluated for each criterion and finally ranked. For the proposed NES design, the PUGH matrix highlighted a concept with the most number of positives, but the others in line were not considered as totally inappropriate.

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