

Extended Torsional Delayed Resonators as Applied to Reduce Driveline Vibrations

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Abstract

This study presents a new perspective to reduce automotive driveline vibrations. Active vibration dampers, namely torsional Delayed Resonators, are used instead of the classical approach to control the engine torque. Delayed Resonators are known to have a limited operable frequency range due to stability issues. Therefore, a new feedback strategy is introduced to the torsional Delayed Resonators in order to extend the operable frequency range of the method. The performance and the efficiency of the proposed approach is evaluated through stability analysis and system simulations.

1. Introduction

Vehicle drivelines are subject to inevitable torsional vibrations under acceleration and disturbances due to the existence of elastic components like shafts. These oscillations may cause unsatisfactory driving experience thus, a lot of engineering effort is spent by the manufacturers for NVH (Noise, vibration, harshness) improvements. In addition to improved driveline acoustics, drivability and ride quality, reducing the driveline oscillations will also have economic benefits as a result of the abridged calibration effort.

In literature different methods are proposed to suppress driveline oscillations [1-3]. Aim of these methods is to control the engine torque in order to reduce driveline oscillations. Note that, use of predefined look-up tables etc. are very common within the automotive industry and model based approaches are limited due to complexity. Another approach to reduce driveline oscillations is to use torsional dampers [4]. In this sense, torsional delayed resonators (DR) can be considered to be used as an active vibration damper.

DRs are first introduced in 1990's by Olgac and Holm-Hansen [5]. The idea behind the DR is to obtain a pure resonator from a spring-mass-damper system using an intentionally delayed feedback in order to suppress the undesired vibrations. This delayed feedback may be position [6], velocity [7] or acceleration [8] depending on the sensor type used in the application. DRs are also adapted to suppress torsional vibrations [9].

DRs are known to have stability problems against changing vibration frequencies. Although there are auto-tuning algorithms [10-11] proposed in the literature, the operable frequency range of a DR is limited due to stability problems. Thus, the application area of the DRs are limited to specific applications [12]. Recent studies on DRs propose different approaches to extend the operable frequency range of the method. In [13] use of distributed delay is proposed instead of lumped delay. In another study [14], delayed and non-delayed acceleration feedbacks are assessed

while in [15] a different approach to use position and velocity feedbacks together is proposed.

Aim of this study is to present a different approach, namely DRs, to reduce driveline fluctuations that cause NVH problems in vehicles. Considering the known drawbacks of the DR method a new feedback strategy that extends the stable operation range of the torsional DRs is introduced as the main contribution of this study. Evaluation of the proposed DR approach is made by stability analysis and system simulations using a driveline model.

2. Torsional Delayed Resonators

2.1. Classical Delayed Resonator Approach

Similar to the classical linear DRs, torsional DRs can be used to suppress the vibrations in rotational systems like electric drives, vehicle drivelines etc. A torsional DR is depicted in Fig. 1 where J_a is the DR inertia, k_a is the rotational stiffness and b_a is the rotational damping coefficient of the DR, respectively.

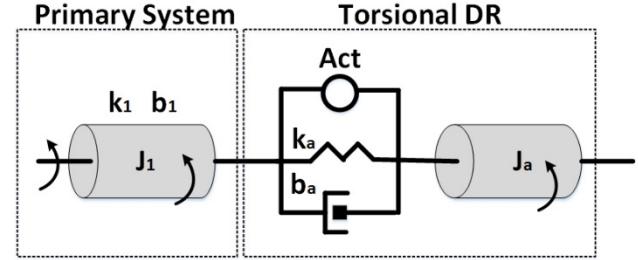


Fig. 1. Torsional DR

DR uses a partial state feedback with time delay to obtain a pure resonator which is known to be a perfect absorber against tonal vibrations. The control effort applied via the actuator is given as

$$u(t) = g\dot{\theta}_a(t - \tau) \quad (1)$$

for the delayed velocity case where g and τ is the feedback gain and delay, respectively. Consequently, the equation of motion regarding the torsional DR is given as,

$$J_a\ddot{\theta}_a(t) + b_a\dot{\theta}_a(t) + k_a\theta_a(t) = g\dot{\theta}_a(t - \tau) \quad (2)$$

The Laplace domain representation of the characteristic equation of the torsional DR is given as

$$CE(s) = J_a s^2 + b_a s + k_a - gse^{-s\tau} \quad (3)$$

The dominant pair of the infinitely many characteristic roots of the DR must be positioned at $\mp j\omega$ on the imaginary axis to suppress the undesired vibrations with the frequency ω . The feedback gain and delay that assign the dominant roots of the DR to the desired location on the imaginary axis are derived from equations, $Re[CE(j\omega)] = 0$ and $Im[CE(j\omega)] = 0$ as,

$$g = \frac{1}{\omega} \sqrt{(b_a\omega)^2 + (k_a - J_a\omega^2)^2} \quad (4)$$

$$\tau = \frac{1}{\omega} \left\{ \text{atan}2 \left[\frac{k_a - J_a\omega^2}{b_a\omega} \right] + 2(l-1) \right\} \text{ for } l = 1, 2, \dots \quad (5)$$

Apparently, for each feedback gain value there are multiple corresponding τ values depending on l which is known as the delay branch number.

2.2. Proposed Delayed Resonator Approach

Considering the stability constraints, a combined feedback strategy that consists of a rotational position and delayed rotational velocity feedback as

$$u(t) = g_v \dot{\theta}_a(t - \tau) + g_p \theta_a(t) \quad (6)$$

is proposed. Thus, the motion equation regarding the proposed DR approach becomes

$$J_a \ddot{\theta}_a(t) + b_a \dot{\theta}_a(t) + k_a \theta_a(t) = g_v \dot{\theta}_a(t - \tau) + g_p \theta_a(t) \quad (7)$$

The Laplace domain representation of the characteristic equation of the proposed DR is given as

$$CE(s) = J_a s^2 + b_a s + k_a - g_v s e^{-s\tau} - g_p \quad (8)$$

Analogously, the feedback gain and delay that assign the dominant roots of the DR to the desired location on the imaginary axis are derived as,

$$g_v = \frac{1}{\omega} \sqrt{(b_a\omega)^2 + (k_a - g_p - J_a\omega^2)^2} \quad (9)$$

$$\tau = \frac{1}{\omega} \left\{ \text{atan}2 \left[\frac{k_a - g_p - J_a\omega^2}{b_a\omega} \right] + 2(l-1) \right\} \text{ for } l = 1, 2, \dots \quad (10)$$

where g_p is the rotational position feedback gain. Aim of this new gain is to adjust the natural frequency of the absorber in order to obtain a wider operation range. Therefore, in this study g_p is proposed as,

$$g_p = k_a - \alpha J_a \omega^2 \quad (11)$$

where α is the tuning parameter that is used to adjust the natural frequency of the torsional DR.

Apparently choosing $\alpha = 1$ sets the natural frequency of the DR to the frequency of the vibrations thus velocity feedback gain and delay becomes $g = b_a$ and $\tau = 0$ for the first delay branch. Note that, the zero delay case may lead to a narrower operating range comparing to the delayed cases [15] therefore higher delay branches may be considered.

3. Modelling the DR and the Vibrations

3.1. DR and the Coupled System

While designing the DR, stability of the coupled system (CS) that consist of the primary system and DR together should be taken into consideration. Equations of motion regarding the CS can be given as,

$$J_a \ddot{\theta}_a(t) + b_a \dot{\theta}_a(t) + k_a \theta_a(t) - g_v \dot{\theta}_a(t - \tau) - g_p \theta_a(t) = b_a \dot{\theta}_1(t) + k_a \theta_1(t) \quad (12)$$

$$J_1 \ddot{\theta}_1(t) + (b_a + b_1) \dot{\theta}_1(t) + (k_a + k_1) \theta_1(t) = b_a \dot{\theta}_a(t) + k_a \theta_a(t) - g_v \dot{\theta}_a(t - \tau) - g_p \theta_a(t) \quad (13)$$

Defining the state vector as $x = [\theta_a \quad \dot{\theta}_a \quad \theta_1 \quad \dot{\theta}_1]^T$ the state space representation of the coupled system is given as

$$\dot{x} = Ax + A_t e^{s\tau} + Bf \quad (14)$$

where,

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -k_a & -b_a & k_a & b_a \\ \frac{J_a}{J_a} & \frac{J_a}{J_a} & \frac{J_a}{J_a} & \frac{J_a}{J_a} \\ 0 & 0 & 0 & 1 \\ \frac{k_a}{J_1} & \frac{b_a}{J_1} & -\frac{k_a + k_1}{J_1} & -\frac{b_a + b_1}{J_1} \end{bmatrix}, \quad (15)$$

$$A_t = \begin{bmatrix} 0 & 0 & 0 & 0 \\ \frac{g_p}{J_a} & \frac{g_v}{J_a} & 0 & 0 \\ 0 & 0 & 0 & 0 \\ -\frac{g_p}{J_1} & -\frac{g_v}{J_1} & 0 & 0 \end{bmatrix}, B = \begin{bmatrix} 0 \\ 0 \\ 0 \\ \frac{1}{J_1} \end{bmatrix}. \quad (16)$$

3.2. Driveline Model

Driveline vibrations may occur due to many different cause. One of the major sources of vibration is the clutch judder which can be explained as the transient behavior of the driveline due to sudden change of engine torque (i.e. gear shift). Frequency of the vibrations caused by clutch judder lies between 2-10 Hz [16]. Another source of vibration is the torque fluctuations which affects the steady state as well. Frequency range of the torque fluctuations are usually between 20 – 80 Hz [17]. Note that this study deals with the steady state vibrations, improvements in the transient behavior can be made by applying engine torque control parallel with the proposed DR approach.

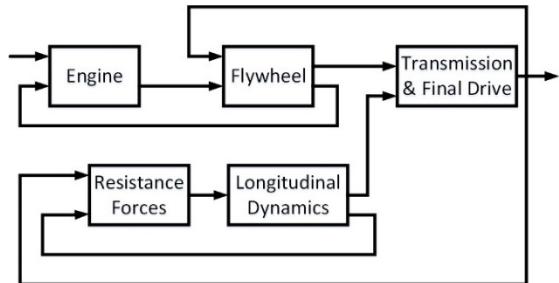


Fig. 2. Block diagram of the driveline model

The block diagram of the driveline model is given in Fig. 2. The driveline model is constructed following the procedure given in [18]. Considering the simplicity of the paper, detailed equations of the driveline and engine model is not provided in this study but can be found in [18].

4. Evaluation of the Proposed Method

System simulations and stability analysis are used to evaluate the performance of the proposed DR approach. Physical parameters of the driveline [18] and the DR are given in Table 1. Remarks on the choice of DR parameters can be found in [6] and [14].

Table 1. DR and driveline parameters

Parameter	Value
DR torsional stiffness k_a	320 Nm/rad
Driveline torsional stiffness k_1	6420 Nm/rad
DR torsional damping b_a	0.01 Nms/rad
Driveline torsional damping b_1	90 Nms/rad
DR moment of inertia j_a	0.009 Nms ²
Driveline moment of inertia j_1	0.045 Nms ²

4.1. Stability

A wide stable operation range is crucial considering the varying vibration frequencies due to changing speed and conditions. Characteristic polynomial of the coupled system is obtained as

$$\begin{aligned} s^4 + s^3 \left(\frac{b_a + b_1}{J_1} + \frac{b_a - g_v e^{-\tau s}}{J_a} \right) + \dots \\ s^2 \left(\frac{k_a - g_p}{J_a} + \frac{k_a + k_1}{J_1} + \frac{b_1(b_a - g_v e^{-\tau s})}{J_a J_1} \right) + \\ s \left(\frac{b_1 k_a + b_a k_1 - b_1 g_p - k_1 g_v e^{-\tau s}}{J_a J_1} \right) + \frac{b_1(k_a - g_p)}{J_a J_1} \end{aligned} \quad (17)$$

moving from (12) and (13).

Because of the transcendental terms and many parameters spectral analysis for the stability of the CS is almost impossible. Therefore numerical methods [19] are used to assess the stability of the system. The stable frequency range of the classical torsional DR and the proposed approach is obtained by finding the rightmost root of the CS by applying the procedure presented in [19] and depicted in Fig. 3. Note that the frequency is normalized by the natural frequency of the DR as,

$$\bar{\omega} = \omega / \sqrt{k_a/m_a} \quad (18)$$

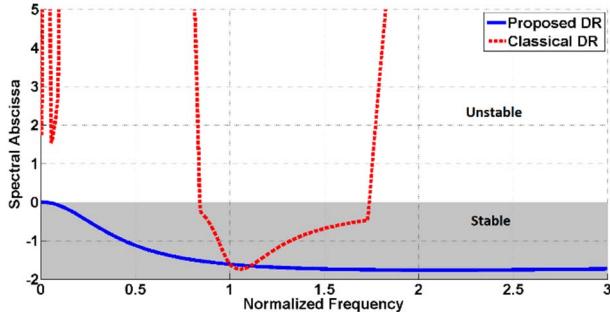


Fig. 3. Spectral abscissa of the CS

Parameters of the classical DR are calculated using (4) and (5) while the parameters for the proposed method are calculated using (9), (10) and (11). The second delay branch ($l = 2$) is used for both methods as it provides the widest stability range for the classical method [15].

The proposed method provides a wider stability range comparing to the classical DR's ($0.8\bar{\omega} - 1.74\bar{\omega}$) as seen in Fig. 3. Moreover, the spectral abscissa of the proposed approach is smaller than the classical approach for most of the operation range. Therefore the vibrations will be suppressed faster.

4.2. System Simulations

Effects of the torsional DR on the undesired vibrations are examined using system simulations. First, different vibration frequencies are applied to the systems presented in Table 1 to see the efficiency of the proposed method. Vibration frequencies and DR parameters regarding these frequencies are given in Table 2.

Table 2. DR and driveline parameters

Frequency ω (Hz)	$\bar{\omega}$	Classical DR		Proposed DR		
		g	τ	g_p	g_v	τ
15	0.5	Unstable		240.056	0.01	0.0667
30	1.0	0.0101	0.034	0.2248	0.01	0.0333
80	2.67	Unstable		-1954	0.01	0.0125

Three vibration frequencies are chosen to evaluate the performance of the torsional DRs within the frequency range of the engine torque fluctuations. Simulation results are depicted in Fig. 4-6 for $\omega = 15\text{Hz}$, $\omega = 30\text{Hz}$ and $\omega = 80\text{Hz}$ respectively.

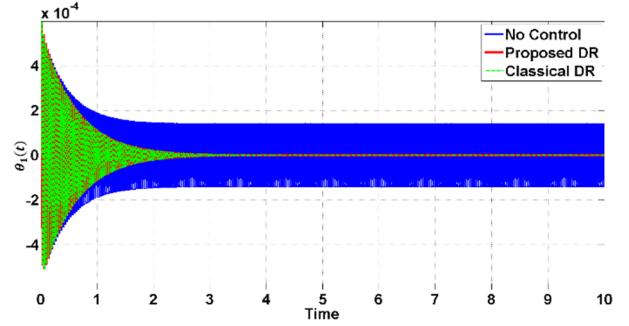


Fig. 4. Angular position of the primary shaft for $\omega = 30\text{ Hz}$

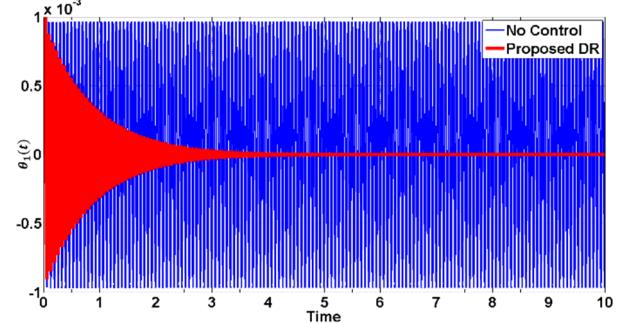


Fig. 5. Angular position of the primary shaft for $\omega = 15\text{ Hz}$

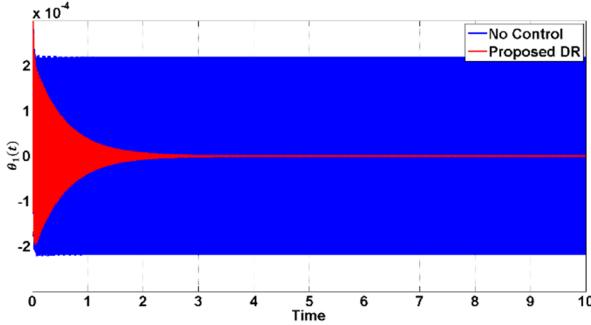


Fig. 6. Angular position of the primary shaft for $\omega = 80\text{Hz}$

As seen from Fig. 4, classical DR approach and the proposed DR approach present similar vibration suppression results when the vibration frequency is around the natural frequency of the DR ($\omega = 30\text{ Hz}$). The proposed approach maintains the same vibration suppression performance under changing vibration frequencies as depicted in Fig. 5 and Fig. 6, where the classical DR approach is not operable due to stability issues.

Secondly, the proposed DR approach is applied to the driveline model to evaluate its performance under working conditions. Output shaft speed obtained by the system simulations are shown in Fig. 7 and Fig. 8.

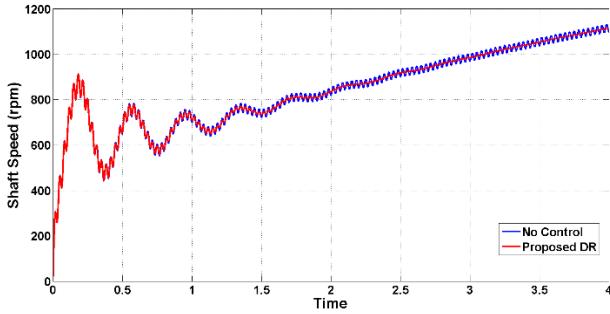


Fig. 7. Output shaft speed in rpm ($\omega = 40\text{Hz}$)

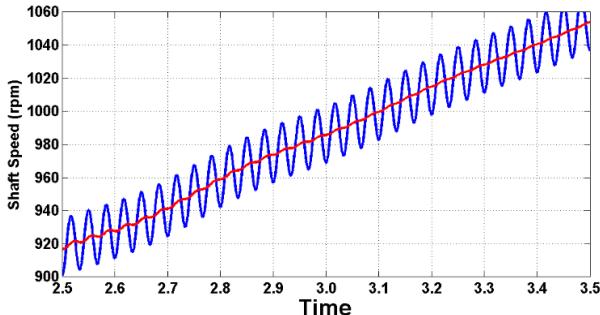


Fig. 8 Zoom in view output shaft speed in rpm ($\omega = 40\text{Hz}$)

As seen from the output shaft speeds presented in Fig. 7 and Fig. 8, speed fluctuations that cause driveline vibration and noise are significantly reduced.

6. Conclusions

A new approach, namely torsional DRs, are used as active vibration dampers to reduce automotive driveline vibrations. A

different feedback strategy that consists of delayed and non-delay terms are used together rather than the classical DR approach in order to obtain a wider operable frequency range. Aim of the non-delayed feedback is to artificially adjust the natural frequency of the torsional DR to the frequency of the vibrations while the delayed velocity feedback is used to tune the torsional DR to suppress the undesired vibrations.

Efficiency of the proposed method on extending the operable frequency range of the classical DRs is shown via stability analysis and simulation results. Considering the wide operating range and its effective vibration suppression capability, torsional DRs with the proposed feedback strategy can be considered as a promising alternative to reduce NVH problems in automotive drivelines.

7. References

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