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A Modified Delayed Resonator for Active Suspension Systems of Railway Vehicles

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Abstract: Car body vibrations caused by the track irregularities needs to be suppressed in order to provide a good ride quality. Although there are numerous researches on active suspension systems, practical implementations of the active suspension systems for vertical car body vibrations are limited. In this study, a modified Delayed Resonator (DR) with delayed speed feedback is used as an active vibration absorber for suppressing vertical vibrations of the car body. In order to maintain suppression at a wide frequency range a new term is added to a classical DR which is affecting on the natural frequency of the vehicle.

***Keywords**:* Delayed Resonator, Time delay, Active vehicle suspension, Railway Vehicles.

1. INTRODUCTION

As travel speeds of trains are increasing, needs for advanced, cost efficient engineering solutions to provide safe and comfortable journeys are also increasing. In order to improve ride quality, car body vibrations which are caused by the vertical irregularities of the rail tracks should be suppressed through suspension systems.

Studies on suspension systems of railway vehicles evolved from passive to active systems over years. Researches on suppressing vertical car body vibrations mostly focus on active (fully or semi) secondary suspensions. Different control strategies like skyhook method (Li 1999), LQG (Zamzuri et al. 2007), LQR (Yusouf 2013), H_∞ (Hirata and Takahashi 1993) are proposed for secondary active suspension systems. Studies on fully active (Pratt 1996; Hirata et al. 1995; Sasaki et al. 1994) and semi active suspension systems (Pugi et al. 2009; Sugahara et al. 2009; Sugahara et al. 2011; Stribersky et al. 1998; Roth and Lizell 1996; Narinao 1997; Tanifuji et al. 2002) shows that active systems gives better results than passive systems. Despite theoretical results, practical implementations of active suspension systems for vertical car body vibrations are limited (Orvnas 2010) comparing to lateral suspension systems. Other than practical tests (Sugahara et al. 2009; Sugahara et al. 2011), active suspension systems for vertical car body vibrations have not been implemented on trains, mostly because of the high costs and their complex structures.

As a different approach, Delayed Resonators (DRs) can be used for suppressing car body vibrations. DRs are active vibration absorbers based on the strategy to oscillate a simple mass-spring-damper system at a desired frequency by using a delayed feedback (Olgac and Holm-Hansen 1994). In addition to being simple and tunable, implementation of the DRs are easy and independent from the bogies or the secondary suspension systems which makes them suitable for railway applications. Research regarding to the use of DRs on active suspension systems of railway vehicles (Pandarathil et al. 2006; Eriş et al. 2014) reports successful results though, further studies considering changing train speeds should be made since previous studies are made for fixed train speeds.

Aim of this study is to present a different control strategy, for suppressing vertical car body vibrations in a wide frequency range. In order to do this multiple DRs with a newly introduced term are used. Simulations of the vertical car body vibrations for different train speeds are conducted using randomly generated track profile. Stability analysis and comparisons between classical DRs and the proposed method are also given.

2. MATHEMATICAL MODELS

2.1 Track Irregularity Model

Irregularities of the vertical track profile can be considered as a Gaussian random process for straight rail lines without turnouts or road crossings (Lei and Noda 2012). Characteristic of the road profile is obtained through a power spectral density (PSD) function as,

$$S(\omega) = \frac{\mathrm{k}\,\mathrm{A_v}\,\omega_c^2}{(\omega^2 + \omega_c^2)} \tag{1}$$

where k is a constant and A_v, ω_c are coefficients related with the road quality.

The road profile is a stochastic process with an expected value zero and generated from (2).

$$\eta(t) = \sum_{k=1}^{N} \alpha_k \sin(\omega_k t + \phi_k) \tag{2}$$

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Here ϕ_k is a random number uniformly distributed between 0 and 2π. α_k is a Gaussian random variable with an expected value zero and variance σ_k. Variance is calculated using the PSD function given in (1), within a frequency band $\Delta\omega$.

$$\Delta\omega = (\omega_u - \omega_l)/N \tag{3}$$

$$\omega_k = \omega_l + \left(k - \frac{1}{2}\right)\Delta\omega, \;\; k = 1,2,..,N \tag{4}$$

$$\sigma_k^2 = 4S_x(\omega_k)\,\Delta\omega, \;\; k = 1,2,..,N \tag{5}$$

Vertical track profile is simulated using Matlab© for the best line grade according to American Railway Standards by choosing $k = 0.25$, $A_v = 0.039cm^2 \cdot rad/m$ and $\omega_c = 0.8245\text{rad/m}$ (Lei and Noda 2012). N is taken sufficiently large enough as 2500. Upper (ω_u) and lower (ω_l) limits of the frequency band are chosen as (0.004π)rad/m and (4π)rad/m respectively. Generated track profile is given in Fig. 1

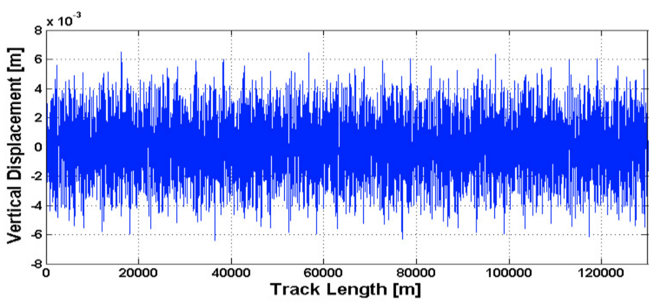

Fig. 1 Generated vertical track profile with irregularities.

2.2 Train Model with Delayed Resonators

In this study, a simple side-view model of the train with two DRs with identical parameters but different control signals, given in Fig. 2 is used. Aim of the DRs is to suppress vibrations of the car body (m_3) that are caused by the forces (f_1 *and* f_2) applied to the bogies(m_1, m_2).

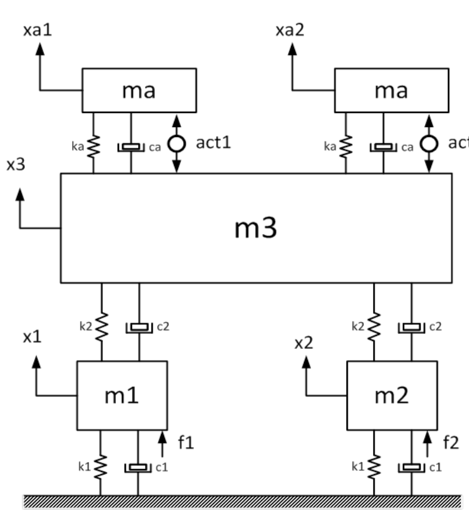

Fig. 2 Train model with delayed resonators.

Defining states as $x = [x_{a1}\; \dot{x}_{a1}\; x_{a2}\; \dot{x}_{a2}\; x_1\; \dot{x}_1\; x_2\; \dot{x}_2\; x_3\; \dot{x}_3\,]$, motion equations regarding to the system with passive absorbers given in Fig. 2 are constructed as in (6).

$$\begin{aligned}
\ddot{x}_{a1} &= (k_a x_3 + c_a\dot{x}_3 - k_a x_{a1} - c_a\dot{x}_{a1})/m_a \\
\ddot{x}_{a2} &= (k_a x_3 + c_a\dot{x}_3 - k_a x_{a2} - c_a\dot{x}_a)/m_a \\
\ddot{x}_1 &= (k_2 x_3 + c_2\dot{x}_3 - (k_1 + k_2)x_1 - (c_1 + c_2)\dot{x}_1)/m_1 \\
\ddot{x}_2 &= (k_2 x_3 + c_2\dot{x}_3 - (k_1 + k_2)x_2 - (c_1 + c_2)\dot{x}_2)/m_2 \\
\ddot{x}_3 &= (k_a x_{a1} + c_a\dot{x}_{a1} + k_a x_{a2} + c_a\dot{x}_{a2} + k_2 x_1 + c_2\dot{x}_1 + \\
&\quad k_2 x_2 + c_2\dot{x}_2 - 2(k_a + k_2)x_3 - 2(c_a + c_2)\dot{x}_3)/m_3
\end{aligned} \tag{6}$$

Physical parameters of the vehicle that are given in Table 1 are the approximate values of a Shinkansen SKS300 train (Pandarathil et al. 2006).

Table 1. Physical parameters of the vehicle

Mass of bogie and wheel (m_1, m_2)	5000kg
Mass of vehicle (m_3)	42500kg
Primary Stiffness (k_1)	1180000N/m
Secondary Stiffness (k_2)	530000N/m
Primary Damping (c_1)	39200N·s/m
Secondary Damping (c_2)	90200N·s/m

3. DELAYED RESONATORS

3.1 Designing the Delayed Resonators

Motion equations regarding to a typical DR with delayed speed feedback can be given as,

$$m_a\ddot{x}_a(t) + c_a\dot{x}_a(t) + k_a x_a(t) - g_c\dot{x}_a(t - \tau_c) = 0. \tag{7}$$

where, m_a, c_a, k_a, are mass, damping coefficient and stiffness of the DR respectively. Here g_c, actuator gain and τ_c is the time delay of the actuation of the DR.

Using the Laplace transformation, characteristic equation of the each DR is obtained as,

$$m_a s^2 + c_a s + k_a - g_c s e^{-s\cdot\tau_c} = 0. \tag{8}$$

From the characteristic equation, controller parameters g_c and τ_c that drives the DR marginally stable for a desired resonance frequency are derived as,

$$g_c = (1/w_c)\sqrt{(c_a w_c)^2 + (-m_a {w_c}^2 + k_a)^2}\,, \tag{9}$$

$$\tau_c = \frac{1}{w_c}\left\{\tan^{-1}\left[\frac{-m_a {w_c}^2 + k_a}{c_a w_c}\right] + 2l\pi\right\}, \quad l = 0,1,2\,.... \tag{10}$$

Choosing physical parameters (m_a, k_a, c_a) properly is important in DR design. Delayed Resonators are designed for a fixed frequency (w_c) and any change at this frequency may lead to a performance loss even if the control parameters (g_c, τ_c) are auto-tuned since the calculated gain might not be applicable due to stability issues. In this study, a new term is introduced in order to attain better suppression for the vibrations which have varying frequencies with respect to speed of the vehicle.

Characteristic equation including the newly introduced term can be given as,

$$m_a s^2 + c_a s + k_a - g_{ci} s e^{-s\cdot\tau_{ci}} - h_{ci} = 0 \tag{11}$$

The additional controller parameter h_{ci} , where *i* indicates the number of the DR, is calculated from the natural frequency of the DR and the desired frequency to be suppressed as,