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Effect of Multi-Objective Control on Ride Quality in High Speed Railway Vehicle

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Abstract: The railway transportation has a significant impact on both the freight and passenger carriage, therefore it should provide a comfortable, safe, fast and inexpensive riding. In this paper, a six-degree-of-freedom half-car model for a typical high speed passenger vehicle is derived to study the ride motions of the vehicle under the random rail inputs. The random rail excitation is considered as an output of a second-order linear filter to white noise excitation and the temporal correlation between the front and the rear wheels is predicted by a second-order Pade filter. The unified "track-vehicle-human" model is formulated and an active suspension based on a Linear-Quadratic-Gaussian (LQG) control is designed to minimize the performance index which is a weighted sum of vehicle performance measures such as carbody vertical and pitch accelerations, front and rear suspension strokes, rail holding and control forces.

Keywords: Random vibration, LQG, high-speed railway vehicle, 6 DOF half-car model,

1. INTRODUCTION

Railway vehicles take an important role in high speed transportation because of their eco-friendly, safe and inexpensive features. They constitute of very huge and complex structures which accumulate the experienced vibration and noise levels together with the increasing speed demands. Thus, they should be comfortable enough to compete with other transportation systems Wei and Rush (1998). A typical high speed train has two bogies that are connected to the carbody via secondary suspensions and each bogie has two wheelsets which are connected via primary suspensions. The secondary suspension in the vertical direction cares with the ride comfort issue and primary suspension is related to stability. Suspension systems are responsible for critical speed determination with the help of lateral suspension, ride comfort and stability analysis.

Incorporating intelligent mechatronical systems and usage of actively controlled suspensions is recently used to overcome the vibration problems faced both in literature and industry. Vibrations are mainly caused by track or rail irregularities, from which they are transmitted via the bogies and the carbody to the passengers. Evaluation of noise and ride comfort characteristics in relation to railway vehicles is described in Kouroussis et al. (2014). The study of rail-vehicle dynamics and carbody structural flexibility is supported by the framework of flexible multibody-dynamics in Dietz (1999). Ride quality evaluation was studied with a weighted power spectral density of the carbody acceleration in the work of Tanifuji (1986). Low vibration level is one of the important contributers to a good ride comfort and fatigue experienced during travelling Pintado (1990), Zhou (2009).

Vertical structural vibrations become more prominient when studying the human whole-body dynamics ISO (1981), Wei and Rush (1998). It has been shown that discomfortable driving negatively impacts the human health. According to worldwide accepted standards ISO (1981), humans are mainly sensitive to vertical vibrations in the frequency range of 5 to 15 Hz and to lateral vibrations in the interval of 1 to 2 Hz, where the sensitivity reaches the maximum at approximately 8 Hz. The sensibility of the human reaction to changing frequencies in the 1 to 50 Hz bandwidth was studied in Nakagawa (2011). It was seen that the effects of vibrations depend on the magnitude and the frequency of the acceleration. Therefore, different configurations in suspension systems including active and semi-active designs are proposed in the literature aiming to reduce the passenger discomfort especially at high speeds. Optimal designs for rail vehicle suspensions is widely considered in Pintado (1990).

The applications of active systems to vertical and lateral vibration isolation and stability augmentation involving the secondary suspension have been outlined by Hedrick et al. (1981). For conventional rail vehicles, a variety of active systems were designed and tested in terms of ride quality concepts but the additional cost of the active equipment involved in the suspensions mostly deterres the commercial applications of these designs Zolotas (2009). In order to reduce the high costs, some studies suggest using lighter bogie configurations and some others suggest solving this problem with semi-actively controlled systems Yusof (2012), Ornväs (2010), Zolotas (2009), Zhou (2009). Most of the control system techniques aim to associate an active torque connection between the two wheels of a wheelset to form a magnetic coupling and various control laws are used with the object of providing

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a good torque connection between the wheels at low frequencies to maintain the curving ability since at high frequencies the wheels are more or less uncoupled so that instability does not occur. It can be seen that, control engineering techniques will exercise a strong influence on the dynamics of railway vehicles either by improving the dynamics of vehicles using the conventional wheelset, or by supporting the development of more innovative systems.

This paper is organized as follows: In Section 2, a six-degree-of-freedom (6 DOF) vertical half-car model of a typical high speed passenger vehicle is condidered. The random rail excitation is considered as the output of a second-order linear shaping filter to white noise excitation and the time correlation between the front and rear wheels is modelled with Pade approximations. Then, in Section 3 an optimization problem is formulated and solved by using Linear Quadratic Gaussian scheme. The control objective is to decrease the quadratic norms of the vertical and pitch accelerations of the carbody while keeping the primary and secondary suspension travels bounded. The results show that the active suspension performs significantly better than the passive suspension and the root-mean-square (RMS) performance of the active suspension with LQG control can be significant, in particular for high speed vehicles.

2. MODELLING

A schematic representation of s six-degree-of freedom high speed railway vehicle is shown in Fig. 1. The model consists of a car body m_b, two bogie masses m_t, and two wheel-axle sets at the front and rear corners of the vehicle. The car body and each bogie mass is assumed to be rigid and have freedoms of motion in vertical (bounce) and pitch directions. The wheel sets are connected to the bogie frames by primary suspension systems that are modeled with linear springs and viscous damping elements. The suspension system between the bogie frames and the car body, referred to as the secondary suspension consists of actuators $u_i, i = 1, 2$ in parallel with another set of linear passive suspension elements of springs and dampers. The pair (z, θ) shows the vertical displacement at the center of gravity and the pitch angle of the car body while the pair (z_{ti}, θ_{ti}), $i = 1, 2$ corresponds to the vertical displacement at the center of gravity and the pitch angle of the bogie masses at the front and rear corners, respectively.

The primary and secondary suspensions are going to be used to study the ride quality, the safety performance during curve negotiations and the dynamic wheel-rail track interactions of the vehicle. The parameter values, chosen for this study are obtained from Zhou (2009) and are presented in Table 1. They are typical for a high-speed passenger vehicle.

The characteristic equations of the secondary and primary secondary suspensions provide the *heave*, *pitch* and the *wheel-hop* rigid modes of the ride motion, respectively. Considering the half-car configuration in Fig. 1 these modes are calculated and presented in Table 2. They are quite useful at early stages of the suspension design. When the rail excitations' frequency content coincides with the natural frequencies of the vehicle, output responses with large amplitude are induced. High resonance frequencies may negatively impact the ride handling while the low frequencies are more influencial on the ride comfort of the vehicle. Also the speed has a detrimental effect on the railway vehicle. The higher the vehicle speed, the larger the rail excitation is, resulting in a worsened vibration response even on undamaged rails and smooth terrains.

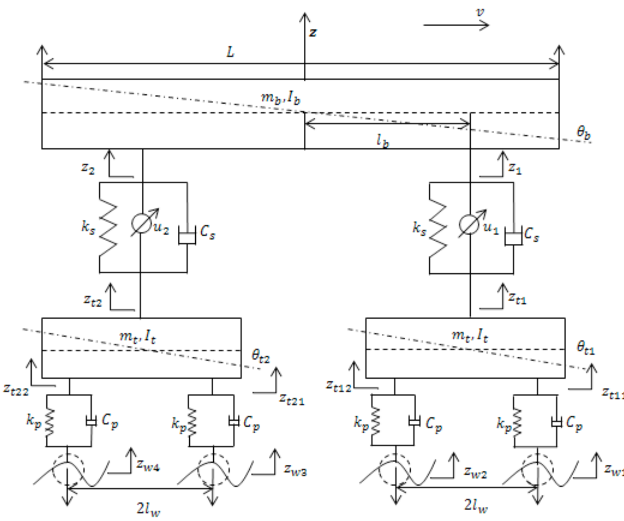

Fig. 1. 6 DOF Vertical half car model.

Table 1. The Vehicle System Parameters for the 6 DOF Model

Variable	Description	Value
m_b	Car-body mass	26, 000 kg
I_b	Car-body pitch moment of inertia	1274, 000 kgm^2
m_t	Bogie frame mass	2600 kg
I_t	Bogie frame pitch moment of inertia	1423.8 kgm^2
C_s	Front (rear) bogie damping coeffient	60, 000 Ns/m
C_p	Tire damping coefficient	30, 000 Ns/m
k_s	Front (rear) bogie stiffness	450 kN/m
k_p	Tire stiffness	2400 kN/m
l_b	Longitudinal distance from the front (rear) bogie to the car-body c.g	8.75 m
l_w	Half of the wheel base	1.25 m
L	Car-body length	24.5 m

Table 2. Rigid modes of the railway vehicle

Mode shape	Damped natural freq. (rad/s)	Damping ratio
Carbody bounce	5.40	0.3922
Carbody pitch	6.41	0.4903
Bogie bounce	41.39	0.2685
Bogie pitch	64.65	0.4536

Assuming that small motions take place, the equations of motion take the following form:

For carbody:

$$m_b\ddot{z} = -k_s x_1 - C_s\dot{x}_1 - k_s x_2 - C_s\dot{x}_2 - u_1 - u_2 \tag{1}$$

$$I_b\ddot{\theta}_b = -l_b k_s x_1 - l_b C_s\dot{x}_1 + l_b k_s x_2 + l_b C_s\dot{x}_2 - l_b u_1 + l_b u_2 \tag{2}$$

For bogies:

$$m_t\ddot{z}_{t1} = k_s x_1 + C_s\dot{x}_1 - k_p x_3 - C_p\dot{x}_3 - k_p x_4 - C_p\dot{x}_4 + u_1 \tag{3}$$

$$I_t\ddot{\theta}_{t1} = -l_w k_p x_3 - l_w C_p\dot{x}_3 + l_w k_p x_4 + l_w C_p\dot{x}_4 \tag{4}$$

$$m_t\ddot{z}_{t2} = k_s x_2 + C_s\dot{x}_2 - k_p x_5 - C_p\dot{x}_5 - k_p x_6 - C_p\dot{x}_6 + u_2 \tag{5}$$