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IFAC-PapersOnLine 49-21 (2016) 274–281

Track Modelling and Control of a Railway Vehicle

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Abstract:
In this paper, a railway vehicle AM96 train developed by Bombardier, Belgium is represented as a half-car model. First, random road defect inputs are modelled as track class 6 from Federal Railroad Administration (FRA) reports and vehicle vibration characteristics are studied for ten-degree-of freedom vehicle model. Next, lumped track model in terms of ballast and soil parameters is added to the structure and the vehicle response to random vibrations are studied. An active suspension problem is formulated and solved by using Linear-Quadratic Gaussian methodology. The improvement in root-mean-square (rms) vertical and pitch accellerations, the suspension travels and the tire deflections are obtained by active suspension design.

Keywords: High speed railway vehicle; AM96 Bombardier; Active suspension design; LQG control; lumped parameter model

1. INTRODUCTION

Railway vehicles nowadays are used to provide a solution to the traffic congestion and pollution caused by ground vehicles. Many vibration standarts consider thelow-frequency vibrations as one of the most important effects on human body and building infrastructures (ISO 2631-2 , 2003), (DIN 4150-2 , 1999). It is advantageous, therefore, to deliberately include the ground vibrations in the vehicle simulations in order to accurately predict the railway vehicle responses throughout the design process. Evaluation of noise and ride comfort characteristics related to railway vehicles is described in (Kouroussis et al., 2014). The ride quality of a vehicle is highly dependent on displacement, rate of change of acceleration and other environmental factors such as noise, dust, humidity and temperature. But as stated in (Nakagawa, 2011) the major effect is caused by rail transmitted ground excitations during driving since the track and soil impose a unilateral geometric boundary constraint on rolling tires to which the vehicle responds by generating loads, moments, motions and deformations. The terrain profile remains a consistent excitation to the railway vehicle, even when the vehicle design changes. In (Kouroussis et al., 2014) the effect of vibration propagation is studied for different type of trains and various modellinging approaches are presented for the evaluation of dynamical characteristics caused by wheel/rail irregularities. It is computationaly impractical to measure a trail and compile these data while studying the vehicle bahaviour over long stretches of railway terrain. Towards this end the statistical properties of the railway profiles are examined and power spectral density representations are used to characterize the terrain profiles. In (Pacchioni et al., 2010) the profile roughness is represented with a power spectral density function and the ride performance potential of an active suspension system design is studied by a sky-hook controller and a linearquadratic-gaussian (LQG) methodology.

The vibration levels experienced by passengers can be reduced by incorporating some modifications to the physical structure of the vehicle. Many studies suggest lighter carbody and bogie designs to achieve the goal. But this is not very obvious, as stated in (Ornväs, 2010) due to the complex couplings between the mechanical and electronic components in railway vehicles, lowering the carbody weight would ussually results in lower natural frequencies which in turn increases the risk of resonance vibrations.

In (Zhou et al., 2009), (Sun et al., 2014), (Kaiser, 2012),(Gangadharan et al., 2008) the effect of the flexibility properties of the carbody and their effect on the ride comfort characteristics are examined. Also in (Zhou et al., 2009) both rigid and flexible modes of the railway vehicle have been discussed. Here, finite element and boundary element methods are suggested for flexible body modelling. In most studies the track and vehicle dynamics have been handled separately. But, the bases of the vehicle-track coupling dynamics theory (Zhai et al , 2008) show that the railway vehicle and the track are two bounded subsytems that are inseperable from each other. The contact force between the wheelsets and the rail track can be either modelled as rigid or elastic. Elastic contact is generally modelled by using the Hertzian contact theory and the rail is usually represented as an infinite Euler or Timoshenko beam. In the computational simulations or in the laboratory testing of prototype parts and subsystems of the vehicle finite element methods are widely used and generally accepted. But because of the assembled larger system of equations that models the entire problem, they

10.1016/j.ifacol.2016.10.566

are not suitable for control design applications. On the other hand lumped parameter models described by ordinary differential equations have a small system size and yet are accurate enough to study the vibration characteristics of the vehicle (Li et al., 2015), (Ahlbeck et al., 1975).

In this study, the railway vehicle is modelled as a multi-body rigid system ten-degree-of-freedom (10 DOF) half-car model. In Section 2 vehicle vertical dynamics is considered for 10 DOF model and vehicle performance evaluations are discussed for passively and actively designed secondary suspension system. In Section 3, the lumped parameter track model is defined with elastic deflection of wheel/rail interface for 10 DOF vehicle model. LQG methodology is used for control design purposes. The control objective is to decrease the root-mean-square (rms) vertical and pitch accelerations, suspension travels and tire deflections. This is the well known ride comfort-road holding trade off experienced in the design of active suspension systems. The paper is concluded by Section 4.

2. 10 DOF RAILWAY VEHICLE VERTICAL THEORITICAL MODEL

A schematic representation of ten-degree-of freedom high speed railway vehicle is shown in Fig. 1. The model consists of a car body m_c, 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 wheelsets 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 variable pairs $(z_c,\ \theta_c)$ and $(z_{ti},\ \theta_{ti})$, $i = 1, 2$ show the vertical displacements at the center of gravity and the pitch angles for both the car body and bogie masses at the front and rear corners, respectively. z_{wi} refers to the $i'th$ wheel vertical displacement at the center of gravity.

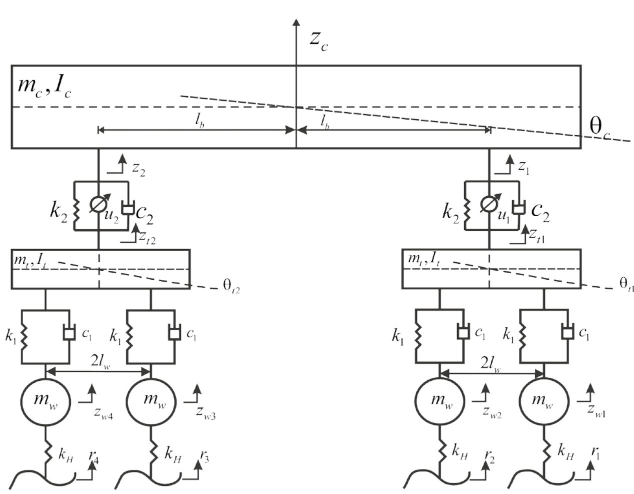

Fig. 1. 10 DOF Railway Vehicle Model

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 interaction of the vehicle. The parameter values chosen for this study are given in Table 1. They are typical for a Bombardier AM96 railway passenger vehicle in (Kouroussis et al., 2014). This vehicle is used in Belgium railways especially for the long distances.

The state vector $x = [x_1, ..., x_{20}]^T$ can be chosen as:

$$\begin{aligned} &x_1 = z_1 - z_{t1}, \quad x_2 = z_2 - z_{t2}, \quad x_3 = z_{t11} - z_{w1}, \\ &x_4 = z_{t12} - z_{w2}, \quad x_5 = z_{t21} - z_{w3}, \quad x_6 = z_{t22} - z_{w4}, \\ &x_7 = z_{w1}, \quad x_8 = z_{w2}, \quad x_9 = z_{w3}, \\ &x_{10} = z_{w4}, \quad x_{11} = \dot{z}_1, \quad x_{12} = \dot{z}_2, \\ &x_{13} = \dot{z}_{t11}, \quad x_{14} = \dot{z}_{t12}, \quad x_{15} = \dot{z}_{t21}, \quad x_{16} = \dot{z}_{t22}, \\ &x_{17} = \dot{z}_{w1}, \quad x_{18} = \dot{z}_{w2}, \quad x_{19} = \dot{z}_{w3}, \quad x_{20} = \dot{z}_{w4}. \end{aligned} \tag{1}$$

with input vector $r = [r_1,\ r_2,\ r_3,\ r_4]^T$. The equations of motion for car body and bogies are omitted for a sake of brevity and the reader is referred to (Leblebici et al, 2015). But, for wheels the equations of motion can be written as:

- for front wheels (first and second wheels);

$$\begin{aligned} m_w \ddot{z}_{w1} = &-k_H(z_{w1} - r_1) + k_1(z_{t11} - z_{w1}) \\ &+ c_1(\dot{z}_{t11} - \dot{z}_{w1}), \end{aligned} \tag{2}$$

$$\begin{aligned} m_w \ddot{z}_{w2} = &-k_H(z_{w2} - r_2) + k_1(z_{t12} - z_{w2}) \\ &+ c_1(\dot{z}_{t12} - \dot{z}_{w2}), \end{aligned} \tag{3}$$

- for rear wheels (third and fourth wheels);

$$\begin{aligned} m_w \ddot{z}_{w3} = &-k_H(z_{w3} - r_3) + k_1(z_{t21} - z_{w3}) \\ &+ c_1(\dot{z}_{t21} - \dot{z}_{w3}), \end{aligned} \tag{4}$$

$$\begin{aligned} m_w \ddot{z}_{w4} = &-k_H(z_{w4} - r_4) + k_1(z_{t22} - z_{w4}) \\ &+ c_1(\dot{z}_{t22} - \dot{z}_{w4}), \end{aligned} \tag{5}$$

Then the genereal state space representation takes the following form:

$$\begin{aligned} \dot{x} &= Ax + B_1 r + B_2 u \\ z &= C_1 x + D_{11} r + D_{12} u \\ y &= C_2 x + D_{21} r + D_{22} u \end{aligned} \tag{6}$$

with secondary suspension travel measurements y and the regulated outputs $z = [x_1,\ x_2,\ x_3,\ x_4,\ x_5,\ x_6,\ \dot{z}_c,\ \dot{\theta}_c]^T$. The state space matrices are,

$$A = \begin{bmatrix} 0_{10\times10} & \tilde{F} \\ \tilde{K} & \tilde{C} \end{bmatrix}, \quad B_1 = \begin{bmatrix} 0_{16\times4} \\ k_H P_1 \end{bmatrix}, \quad B_2 = \begin{bmatrix} 0_{10\times4} \\ L \\ 0_{4\times2} \end{bmatrix},$$

$$C_1 = \begin{bmatrix} I_6 & 0_{6\times14} \\ K_z & \tilde{C}_z \end{bmatrix}, \quad D_{11} = 0_{8\times4}, \quad D_{12} = \begin{bmatrix} 0_{6\times4} \\ -M_1^{-1} S_1^T \end{bmatrix},$$

$$C_2 = [\, I_2 \ 0_{2\times18} \,], \quad D_{21} = 0_{2\times4}, \quad D_{22} = 0_{2\times2}.$$

And for the state space matrices the variables are defined in (Leblebici et al, 2015) and the new variables are,

$$M_3 = \begin{bmatrix} m_w & 0 \\ 0 & m_w \end{bmatrix}, \quad P1 = \begin{bmatrix} M_3^{-1} & 0_{2\times2} \\ 0_{2\times2} & M_3^{-1} \end{bmatrix},$$

$$\tilde{P} = [\, 0_{4\times2} \ P_1 \,], \quad \tilde{F} = \begin{bmatrix} F & N \\ 0_{4\times6} & I_4 \end{bmatrix},$$

$$\tilde{C} = \begin{bmatrix} C & 0_{6\times4} \\ c_1\tilde{P} & 0_{4\times4} \end{bmatrix}, \quad \tilde{K} = \begin{bmatrix} K & 0_{6\times4} \\ k_1\tilde{P} & -k_H P_1 \end{bmatrix},$$

$$\tilde{C}_z = -c_2 M_1^{-1} S_1^T \left[\, 0_{2\times4} \ I_2 \ -T_1 S_2^{-1} \ -T_2 S_2^{-1} \ 0_{2\times4} \,\right]$$