A simulation study on disturbance rejection control of railway vehicle secondary lateral suspension system

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ABSTRACT – The aim of this paper is to study the performance of secondary suspension system of railway vehicle with passive and semi-active systems. A3-DOF railway vehicle suspension model is governed which includes a half railway vehicle body and a set of bogie which are connected by secondary suspension system. A disturbance rejection control known as Stability Augmentation is presented to improve the performance of railway vehicle ride quality. The simulation results show that the disturbance rejection control is able to cancel out unwanted force which is produced from track irregularity.

1. INTRODUCTION

In recent years, advanced control technology has a major impact to the railway vehicle dynamic development since 1975. The vehicle dynamicists have been aware with the use of actuators, sensors and electronic controllers in vehicle suspension. The general benefits can be achieved is better ride quality and running stability of railway vehicle. In this study, a semi-active suspension with Stability Augmentation System (SAS) suspension system is designed for reducing unwanted railway vehicle body motion in lateral direction.

This research concentrates on the control strategies for the lateral movement on the secondary suspension which is concerning the improvement of ride quality to track irregularities. The contribution of this paper is to study the control strategy implemented via semi active dampers located on the secondary suspension that links the bogie to the structure of the car body of the railway vehicle. Two types of control are investigated in this by [1, 2] for vehicle and [3] for light armored vehicle.

2. METHODOLOGY

2.1 MR-Damper Model

A third order polynomial model is proposed to model the characteristics of the MR damper. The polynomial function of positive is y = ax3 - bx2 +cx +d and negative accelerations is y = ax3 + bx2 +cx - d. Five constant currents of 0, 0.5, 1.0, 1.5, and 2.0 Ampere were applied to the damper coils. The force produced by the MR damper due to 2.0 Hz sinusoidal excitation with amplitude 5 cm is presented in Table 2.1.

Table 2.1 Current values and force produced by MR damper

Current, *i* (Amp)	Force (N)
0.0	1106
0.5	1906
1.0	2257
1.5	2618
2.0	2946

The algorithm of the proposed MR damper control can be stated as:

If $G(F_d - BF_{MR})\mathrm{sgn}(F_{MR}) > V_{max}$ then $v = V_{max}$ (1)

Else If $G(F_d - BF_{MR})\mathrm{sgn}(F_{MR}) < V_{min}$ then $v = V_{min}$ (2)

Else $v = G(F_d - BF_{MR})\mathrm{sgn}(F_{MR})$ (3)

Figures 2.2(a) and 2.2(b) show the performance of force tracking control system of the MR damper model that is evaluated using several typical class of continuous and discontinuous input functions. From these figures, it can be seen that the MR damper controller performed well and able to follow the desired force.

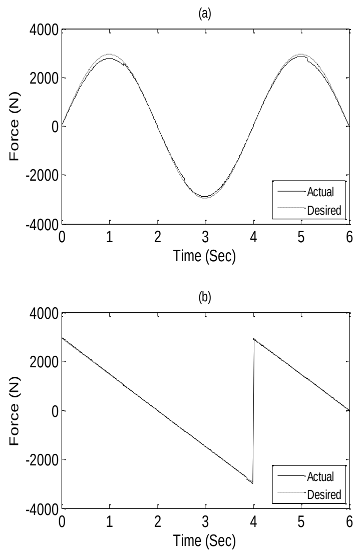

Figure 2.2 Force tracking control of desired force (a) sinusoidal function (b) saw-tooth function

2.2 Railway Vehicle Model

The governing equations of the suspension model are as in [4];

$$m_b \ddot{y}_b = k_1(y_c - h_1\theta_c - y_b) + k_r(y_r - y_b) - F_d \quad (4)$$

$$m_c \ddot{y}_c = -k_1(y_c - h_1\theta_c - y_b) + F_d \quad (5)$$

$$I_r \ddot{\theta}_c = -k_1(y_c - h_1\theta_c - y_b)h_1 - 2k_2 w^2 \theta - 2b_2 w^2 \dot{\theta} + F_d h_1 \quad (6)$$

2.3 Control Structure

The control structure of semi active suspension system implemented in this study for railway vehicle is shown in Figure 2.3 which consists of two loops namely inner loop and outer loop controllers, and decoupling transformation system (DTS).

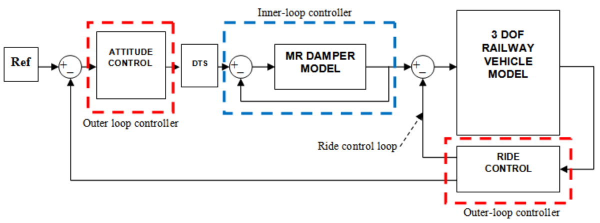

Figure 2.3 Semi-active control structure

3. RESULTS AND DISCUSSION

Four performance criteria are considered to evaluate the semi-active controller namely body displacement, body acceleration, body roll angle and body roll rate. Figure 3.1 shows significant improvement on the railway vehicle body responses in terms of railway vehicle body displacement and acceleration with disturbance rejection control over passive system.

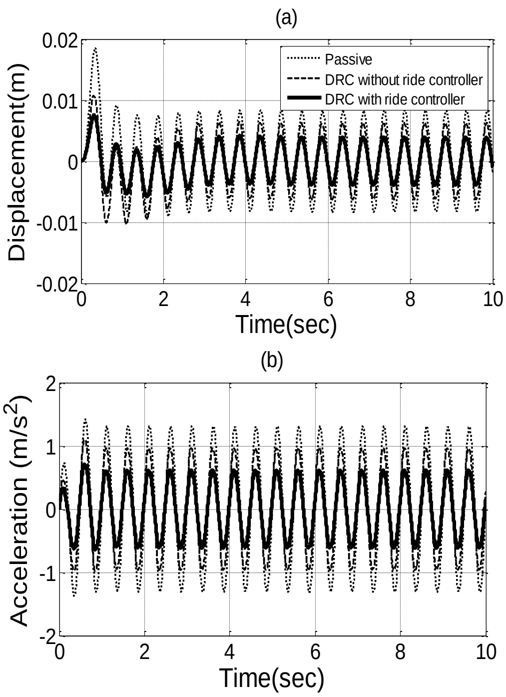

Figure 3.1 Railway vehicle body responses (a) body lateral displacement; (b) body lateral acceleration

Table 3.1 PTP Values of Simulation Results

Performance criteria	Peak-to-peak value		
	Passive	**DRC without ride control**	**DRC with ride control**
Lateral displacement (m)	0.02729	0.02001	0.01332
Lateral acceleration (m/s²)	2.769	2.0877	1.3255

The PTP values of body displacement are 27.29 mm, 20.01 mm and 13.32 mm for passive and DRC without and with ride controller. In terms of body acceleration response, the PTP values for passive system is 2.769 m/s^2, whereas for DTC without ride controller is 2.0877 m/s^2, and for DTC with ride controller is 1.3255 m/s^2. From the results, the disturbance rejection controller of the semi-active suspension system is effective in isolating vehicle body from unwanted motions. All results shown in Table 3.1.

4. SUMMARY

This paper presents a simulation study of railway vehicle body responses using a three degrees-of-freedom half vehicle model of railway vehicle. The disturbance rejection control with additional ride control loop is proposed in outer loop controller to increase or at least maintain the ride comfort level of railway vehicle body. From the simulation, the disturbance rejection control algorithm proved to be effective in achieving better performance of railway vehicle dynamic. The result also proved that the polynomial model has high accuracy even a simple algorithm is used.

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