

Dynamic Response of Commuter Rail Vehicle under Lateral Track Irregularity

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Abstract. Lateral track irregularities normally occur when both rail lines have some displacement laterally with respect to the original track due to prolonged exposure to sun's heat, or may also arise from specific features such as switch and crossing work of track. These track irregularities will cause unwanted body vibration of commuter rail vehicle. These vibrations have to be suppressed for the purpose of ride comfort. This paper presents two control strategies in semi-active suspension systems which are PID and disturbance rejection control to improve passenger ride comfort. A half car model of commuter rail vehicle with three-degree-of-freedom (3-DOF) was developed based on Second Newton's Law. Vibration analyses based on simulation results in time domains are compared with passive system using MATLAB-Simulink software. The results show that the semi-active controllers are able to suppress rail vehicle body responses effectively.

Introduction

Various unwanted vibrations appear in a running commuter rail vehicle on uneven track. These vibrations are transmitted through primary and secondary suspensions which connect the wheel-sets and the bogies, and the bogies with the vehicle body. Excessive vibration on the commuter rail body will indirectly affect passenger comfort and at the same time will result in increased track maintenance cost. To this day, most commuter rail vehicle still using passive suspension system as one of the system to reduce the unwanted vibration. With increased railway vehicle speed, the passive suspension system has to be modified in order to compensate for the deteriorated dynamic behaviour [1]. Controllable and adjustable suspensions have been widely studied by many researchers and academicians experimentally and theoretically for railway vehicles and automobiles. This is due to the limitation of a passive suspension system to dissipate energy occurred during train runs on the track. Therefore, passive, semi-active and active suspensions have their advantages and disadvantages. Passive system provides low performance and less costly when compared with semi-active and active. Both semi-active and active suspension provides good performance especially active system, but semi-active system is also able to perform as good as active system and if it well designed, it will be less costly than active system.

Many successful theoretical work of semi-active [1-3] and active [4-6] control for body vibration of railway vehicle have been carried out in China [2, 3] and Japan [5]. But in contrast, there are many challenges that must be taken into account especially in research and development of semi-active suspension system. The practical application has been limited because of the existence of system error, random error and also an external disturbance in all tests and control processes [2]. Particularly

in the area of heavy vehicle, little research has been done, and there is no commercially available controllable damper suitable for the railway vehicle. This research focus on the secondary lateral damper and the idea is to replace secondary lateral damper with semi-active damper where its damping coefficient can be varied electronically.

3-DOF Railway Vehicle Dynamic Model

Equation of motions

Fig. 1 shows the sectional view of a 3-DOF half commuter rail vehicle model. The model consists of two vertical dampers and springs, and a set of lateral spring and damper in lateral direction. Some of the assumptions considered in this model are as follows: the vehicle body, bogies and wheel-sets are considered as rigid and aerodynamic drag force is ignored. The suspension components between the vehicle body and bogies are modelled as passive secondary system with viscous dampers. The disturbance input is considered from the bogie displacement. Only secondary suspension is considered.

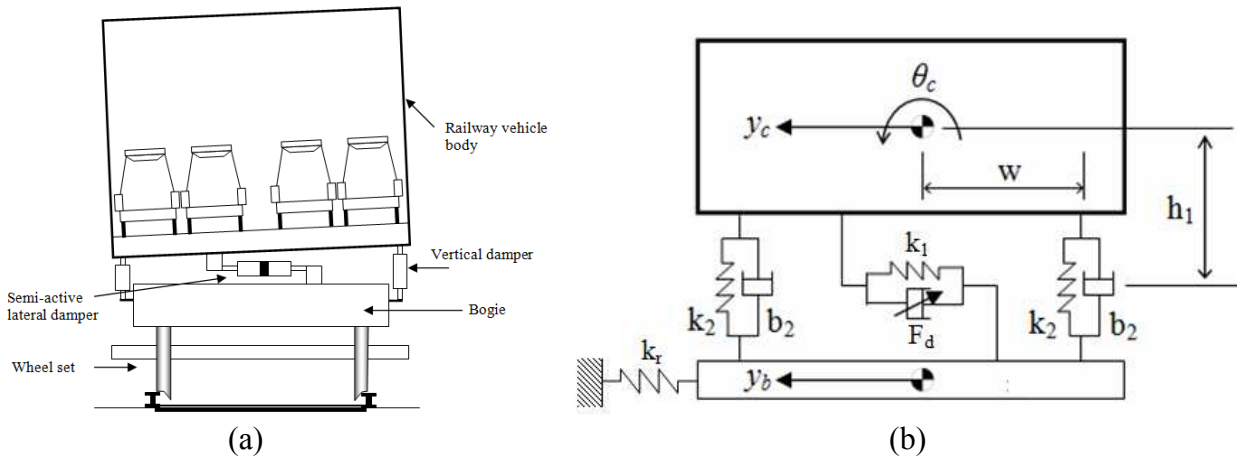


Fig.1 The 3-degrees-of-freedom commuter railway vehicle model.
(a) Sectional front view, (b) schematic diagram

The governing equations of the suspension model are as follows:

$$m_b \ddot{y}_b = k_1 (y_c - h_1 \theta_c - y_b) + k_r (y_r - y_b) - F_d \tag{1}$$

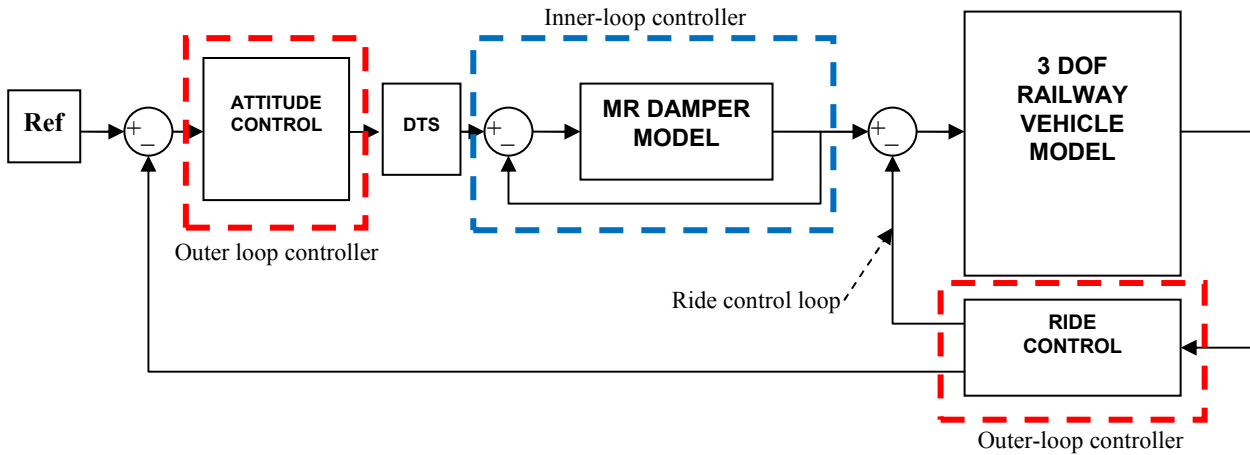
$$m_c \ddot{y}_c = -k_1 (y_c - h_1 \theta_c - y_b) + F_d \tag{2}$$

$$I_r \ddot{\theta}_c = -k_1 (y_c - h_1 \theta_c - y_b) h_1 - 2k_2 w^2 \theta - 2b_2 w^2 \dot{\theta} + F_d h_1 \tag{3}$$

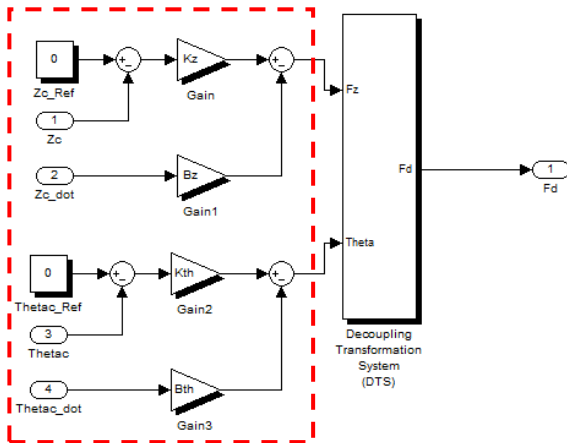
The model was simulated at amplitude of 0.05 m with 2 Hz frequency of bogie frequency response. The parameters used in this simulation based on the half-scale model are; mass of car body $m_c = 50$ kg, mass of bogie $m_b = 30$ kg, secondary lateral spring stiffness $k_1 = 17$ kN/m, secondary vertical spring stiffness $k_2 = 21$ kN/m, stiffness of bogie disturbance $k_r = 240$ kN/m, secondary lateral damping coefficient $b_1 = 15$ kNs/m, secondary vertical damping coefficient $b_2 = 15$ kNs/m, height between body centre of gravity and secondary lateral suspension $h_1 = 0.169$ m, and width of body centre gravity and secondary vertical suspension $w = 0.2$ m.

Outer-loop controller (System controller)

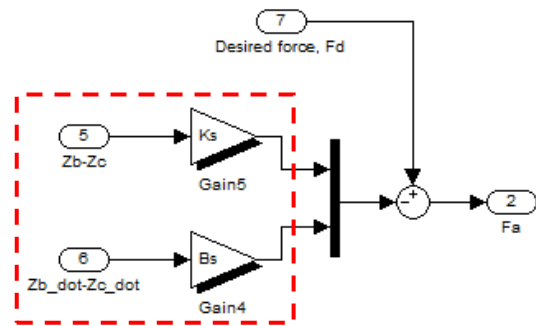
The proposed controller structures of the semi-active suspension system for railway vehicle are shown in Figs. 2 and 3. Fig. 2(a) illustrates the disturbance rejection control which consists of inner and outer loop controllers. Figs. 2(b) and 2(c) show the attitude and ride control of disturbance rejection control. An unwanted weight transfer is rejected by the inner loop controller while the outer loop controller is to stabilize roll response due to the effect of track disturbance. Fig. 3 depicts the control structure of a PID controller.



(a)



(b)



(c)

Fig.2 (a) Disturbance rejection control structure, (b) Attitude control, (c) Ride control

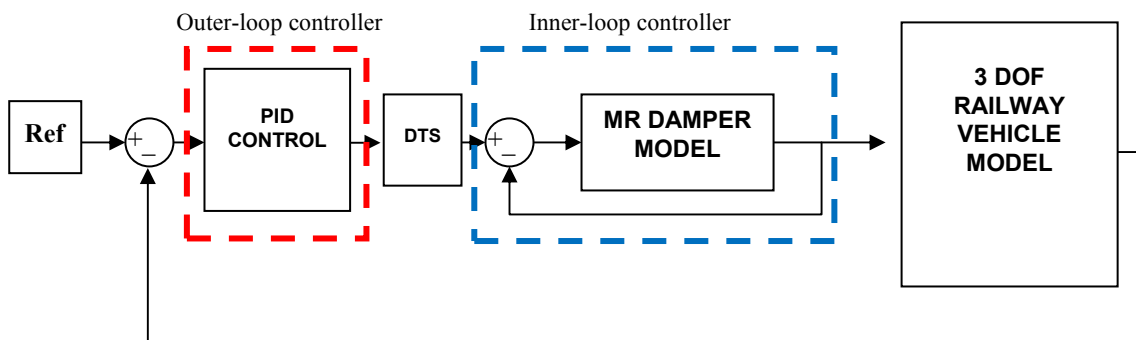


Fig. 3 PID controller

Inner-loop controller (Damper controller)

Inner loop controller also known as MR damper controller is used to mitigate and minimize the unwanted vehicle body motion of railway vehicle. In simulation analysis, MR damper is modelled using Bouc-Wen model due to its versatility and mathematical tractability. The algorithm of the proposed MR damper control can be stated as:

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

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

$$\text{Else } G(F_d - BF_{MR})\text{sgn}(F_{MR}) \tag{3}$$

The values of feedback gain G and B of MR damper controller are set 0.1 and 1 respectively. These values are tuned by sensitivity analysis method in order to find an optimum value of these two controller parameters [7].

Simulation results

The performances of the semi-active controllers of commuter rail vehicle are described in Fig. 4. The time responses of the controlled and uncontrolled system are observed in term of body displacement, body acceleration, body roll angle and body roll rate of commuter rail vehicle. It reveals that there are more important improvements with controlled system especially disturbance rejection control algorithm. PID controller also able to cancel out unwanted vehicle body response effectively but disturbance rejection controller is better than PID controller for all simulation time.

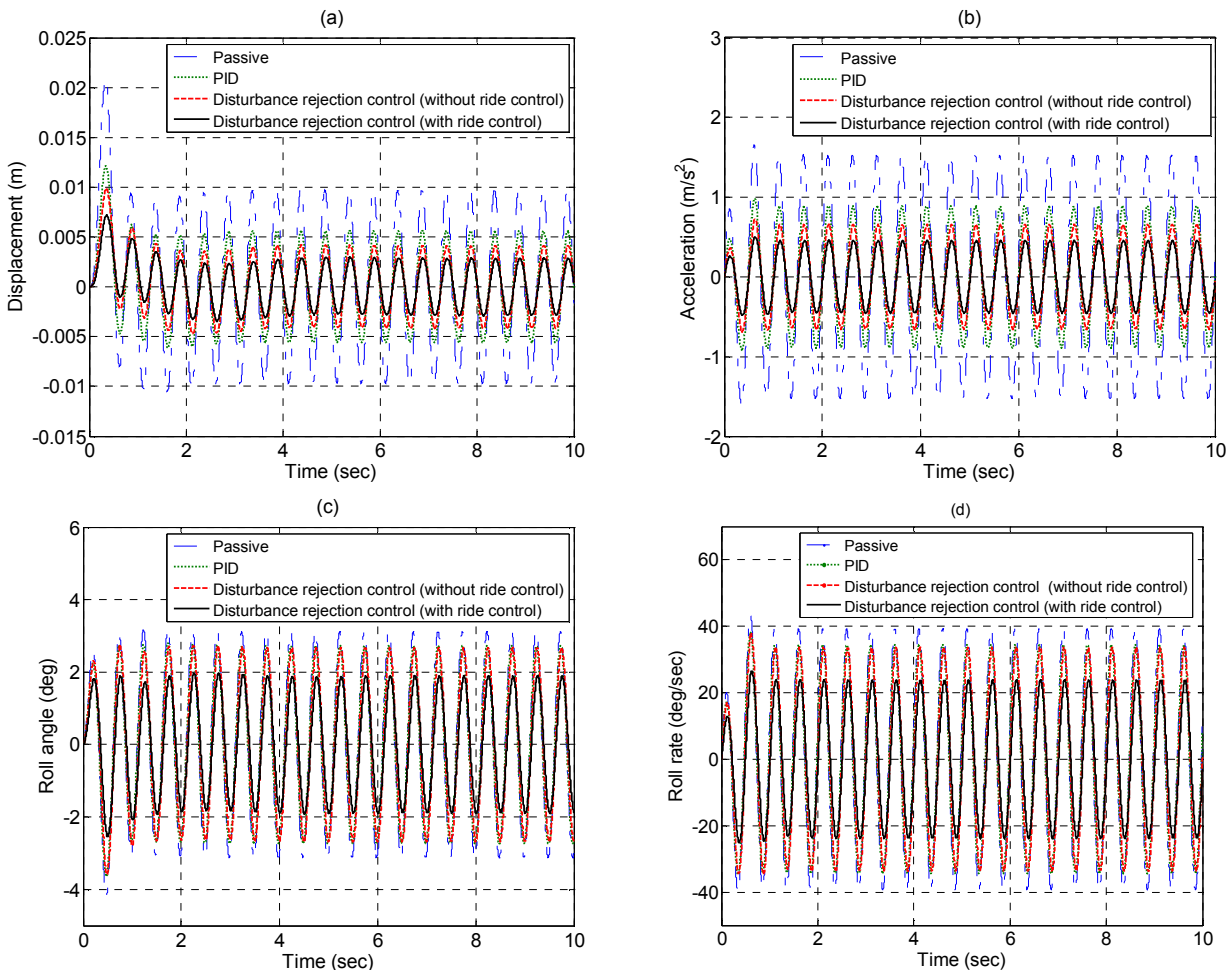


Fig.4 Performance of the body response due to lateral track irregularity. (a) body displacement. (b) body acceleration. (c) body roll angle. (d) body roll rate

Conclusion

This paper presents a comparison of PID and disturbance rejection control schemes of semi-active suspension system on a 3-DOF half commuter rail vehicle model. These two controllers have been simulated by comparing the results with a passive model in time domain. The commuter rail vehicle model is exposed to lateral track irregularity and caused unwanted bogie motion. As the conclusion, disturbance rejection control and PID controller are able to improve vehicle body responses in terms of body displacement, acceleration, roll angle and roll rate.

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