Modelling and simulation of a PID controlled active suspension system of a rail car

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Abstract

Adaptive control are often employed to improve the performance of rail car in terms of stability, safety and ride comfort. In this study, a PID controlled active suspension system was designed, modelled and simulated. The parameters investigated include; the vertical and pitch acceleration as well as the rate of rejection of input and output disturbances. The developed model for the linear rail car suspension system of two degrees of freedom was formulated using the equation of motion while the simulation was performed in the Matlab Simulink 2017 environment. Also, the PID controller was used to fine tune the systems performance according to the Ziegler-Nichols tuning rules. The results obtained show that the designed PID controlled active suspension system is highly efficient in minimizing the effect of load and rail disturbances as the magnitude of the parameters investigated were kept within the permissible limits. This enhances ride comfort, safety and good stability.

Key words: adaptive control, controller, PID, rail car, suspension system

1. Introduction

During the operation of a rail car, there are disturbances that offsets the balance of a rail car which arises as a result of irregular rail profile otherwise known as rail disturbances or as a result of load impressed upon the rail car during movement. Hence, the dynamic behaviour of a rail car is a major function of both the track irregularities or track inputs as well as load inputs which are modelled as deterministic and random variables (Sharma and Kumar, 2013). The bogie of a rail car consists of the primary and secondary suspensions designed to enhance ride comfort, safe curve negotiation and good dynamic behaviour on tangent rail track (Sharma 2013; Sharma *et al.,* 2015). The sets of wheel axle are connected to the bogie frame via the elastic and energy dissipative suspension elements. According to Hasbullah and Faris (2017), the function of the suspension system is to support the weight of the rail car, keep good contact between the tire and the rail and reduce excessive pitching and rolling motion of the rail car body. While the primary suspension permits the movement of the sets of wheel axle in relation to the bogie frame in order to reduce the vibrations transmitted to the car body, the secondary suspension located between the car body and the bogie frame supports the weight of the car. Over the years, adaptive control have been useful in effective control and optimizing the performance of a rail car by minimizing unpleasant motions, wheel deflection and suspension travel (Bideleh and Berbyuk, 2016; Wang *et al.,* 2017). The passive suspension system comprising of springs and dampers has fixed parameters and is relatively inflexible to accommodate the use of adaptive controls hence the performance of such systems are often difficult to control or improve (Nguyen *et al.,* 2000; Zhang *et al.,* 2013). However, the active suspension systems are developed with high degree of flexibility that permits system's improvement via the use of adaptive control (Al-Zughaibi and Davis, 2015) and response dependent dampers (Pekgökgöz *et al.*, 2010; Rosli *et al.* 2014). It essentially consists of an actuator which adjusts the body of the rail car in other to cancel out the dynamic vibrations resulting from disturbances. A good suspension system isolates the railcar body from disturbances and it is usually soft against railway disturbance and hard against load disturbance (Ahmed *et al.,* 2016; Nguyen and Nguyen 2017). A passive suspension system limited to the use of springs and dampers might be inflexible to successive adjustments in order to reach a compromise between the soft and hard settings hence the need for active suspension systems (Abbas *et al.,* 2013; Hashemipour, 2014). The vibration of a moving rail car if not kept within the permissible limits can results in passengers' discomfort, noise, energy losses, decreased durability of the car itself, increased time interval between maintenance and increased cost of operation and maintenance. There various types of adaptive control measures used to enhance ride comfort. For instance, the linear quadratic based semi active control (Unger *et al.,* 2013; Li *et al.,* 2014), the linear quadratic Gaussian control (Sharma, 2014), the fuzzy logic control (Salem and Aly, 2009; Choi and Liu, 2009), neuro-fuzzy logic control (Eltantawie, 2012; Kumar *et al.,* 2018), the proportional, integral and derivative control (Xiao, 2014), the fuzzy-PID control (Emam, 2015), neural network control system (Eski and Yildirim, 2009) etc. The formulation of models that characterizes the dynamic behaviour of rail vehicle to disturbances leads to the dynamic simulation and use of adaptive control to keep critical parameters within permissible limits. The aim of this work is to model and simulate a PID controlled active suspension system of a rail in order to control the response of the rail car suspension system to disturbances. Although many researchers have reported the design and simulation of a PID controlled active suspension systems for both automobile and rail vehicles but the dynamic simulation and control of the effects of rail car vertical acceleration, pitch acceleration as well as the rate of input and output rejection have not been sufficiently highlighted. The work focuses on the design and dynamic simulation of a PID controlled active suspension system of a rail car in order to check unwanted vibrations and improve ride comfort.

2. Materials and method

The modelling was done by representing the rail car and its supports system by a free body diagram of as shown in Figure 1. This was followed by the generation of equations of motion from the Newton's second law. The developed model is represented in the Matlab-simulink 2017 environment while the generated equations and model parameters are employed for the dynamic simulation. This was followed by the introduction of the PID control for enhancing the optimum performance of the system.

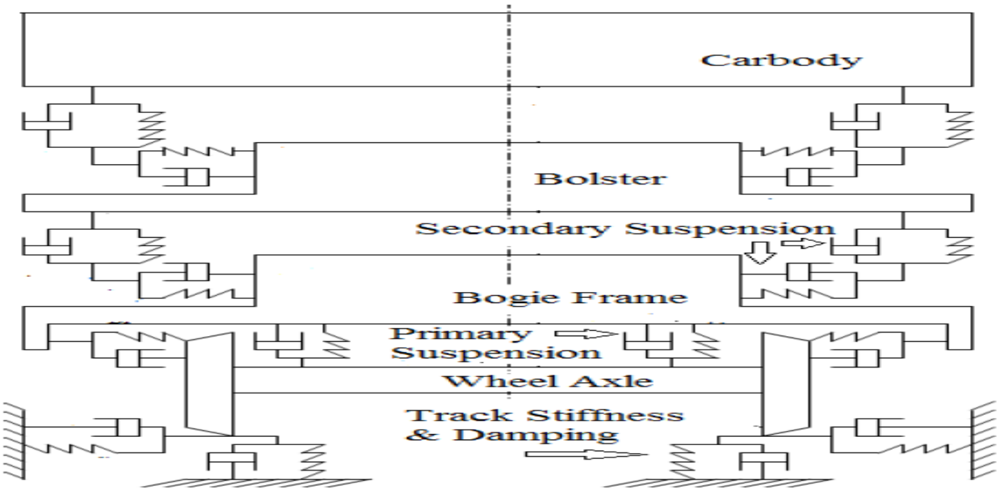

Figure 1: Rail car and its support system (Sharma *et al.*, 2015)

From the Newton's second law of motion, the dynamic equation was obtained as follows

$$F_{m_s} = m_s\ddot{x}_1 \quad (1)$$

$$F_{m_u} = m_u\ddot{x}_2 \quad (2)$$

$$F_{k_s} = k_s^l(x_2 - x_1) + k_s^{nl}(x_2 - x_1)^3 \quad (3)$$

$$F_{k_t} = k_t(x_2 - w) \quad (4)$$

$$F_{b_s} = b_s^l(\dot{x}_2 - \dot{x}_1) \quad (5)$$

Let $b_t = 0\ then, F_{b_t} = 0$

$$+\uparrow \sum F = m\ddot{x}$$

Considering m_s

This implies that,

$$F_{m_s} = F_{K_S} - F + F_{b_s} \quad (6)$$

$$m_s\ddot{x}_1 = k_s^l(x_2 - x_1) + k_s^{nl}(x_2 - x_1)^3 - F + b_s^l(\dot{x}_2 - \dot{x}_1) \quad (7)$$

$$\ddot{x}_1 = \frac{1}{m_s}[k_s^l(x_2 - x_1) + k_s^{nl}(x_2 - x_1)^3 - F + b_s^l(\dot{x}_2 - \dot{x}_1)] \quad (8)$$

Considering m_u

$$F_{m_u} = -F_{k_s} + F - F_{b_s} + F_{k_t} \quad (9)$$

$$m_u\ddot{x}_2 = -k_s^l(x_2 - x_1) - k_s^{nl}(x_2 - x_1)^3 + F - b_s^l(\dot{x}_2 - \dot{x}_1) + k_t(x_2 - w) \quad (10)$$

$$\ddot{x}_2 = \frac{1}{m_u}[-k_s^l(x_2 - x_1) - k_s^{nl}(x_2 - x_1)^3 + F - b_s^l(\dot{x}_2 - \dot{x}_1) + k_t(x_2 - w)] \quad (11)$$

For the state space representation,

Let

$$\dot{x}_1 = x_3 \quad (12)$$

$$\dot{x}_2 = x_4 \quad (13)$$

$$\dot{x}_3 = \ddot{x}_1 = \frac{1}{m_s}[k_s^l(x_2 - x_1) + k_s^{nl}(x_2 - x_1)^3 - F + b_s^l(\dot{x}_2 - \dot{x}_1)] \quad (14)$$

$$\dot{x}_4 = \ddot{x}_2 = \frac{1}{m_u}[-k_s^l(x_2 - x_1) - k_s^{nl}(x_2 - x_1)^3 + F - b_s^l(\dot{x}_2 - \dot{x}_1) + k_t(x_2 - w)] \quad (15)$$

Using the state space representation

$$\ddot{x} = f(x) + gu \quad (16)$$

Where; m_s is the sprung mass (kg); m_u is the unspring mass (kg), k_s^l is the linear component of the spring constant (N/m), k_s^{nl} is the nonlinear component of the spring constant (N/m), k_t is the wheel spring constant (N/m), b_s^l is the linear component of the damping coefficient (Ns/m), b_s^{nl} is the nonlinear component of the damping coefficient, (Ns/m), $y = x_2 - x_1$ is the suspension deflection (m), x_1 is the body deflection (m), x_2 is the wheel deflection (m), w is the road disturbance (m), u is the system input, $F = Ax_p$ is the actuator force (N), F_{k_s} is the spring force acting on the body (N), F_{b_s} is the damping force acting on the body (N) and F_{k_t} is the spring force acting on the wheel (N).

2.2 Design parameters

The model parameters are given as follows:

S/N	**Parameter**	**Notation**	**Value**	**Unit**
1.	Average mass of the rail car body	M_r	50,000	Kg
2.	Mass of suspension system (front)	M_p	38,000	Kg
3.	Mass of suspension system (rear)	M_s	32,000	Kg
4.	Moments of inertia	M_a	78,500	kgm^2
4.	Rail car pitch inertia	M_b	65,700	kgm^2
5.	Average mass of first wheelset and axle	m_1	1500	Kg
6.	Average mass of second wheel	m_2	1,500	Kg

6.	Distance between the centre of gravity and the front position of the rail car	d_i	6.5	m
7.	Distance between the centre of gravity and the middle position of the rail car	d_2	6.5	M
8.	Distance between the centre of gravity and the rear position of the rail car	d_3	6.5	M
9.	Spring constant of the suspension system (front)	k_1	5.0×10^5	N/m
11.	Spring constant of the suspension system (rear)	k_2	5.0×10^5	N/m
12.	Spring constant of the wheel	k_3	2.0×10^5	N/m
12.	Damping constant of the suspension system (front)	b_1	2.5×10^4	Ns/m
14.	Damping constant of the suspension system (rear)	b_2	2.5×10^4	Ns/m
15.	Damping constant of the wheel	b_3	4.0×10^4	Ns/m

The simulated model is shown in Figure 2.

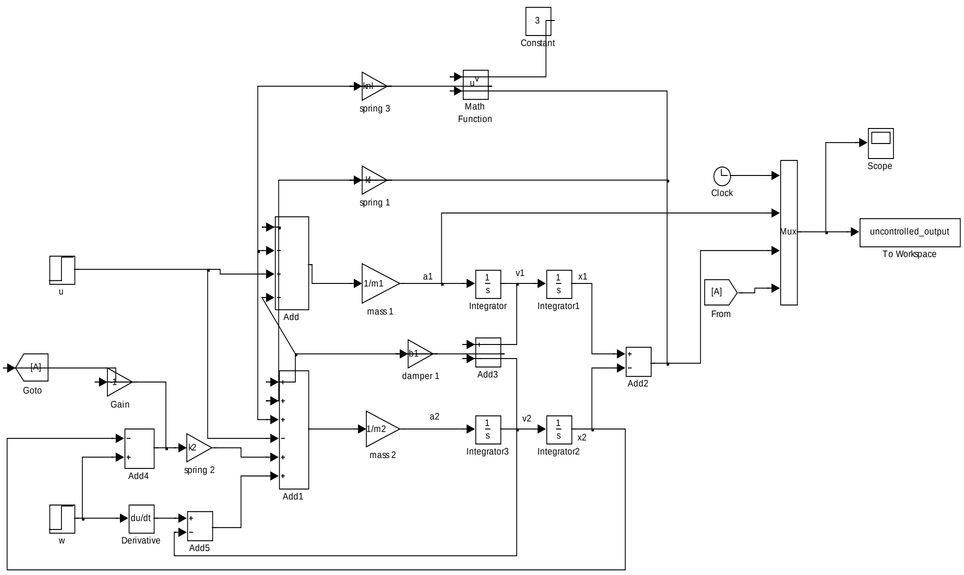

Figure 2: The simulated model

The PID control was employed for fine tuning during simulation using the Nichols Ziegler tuning rules and it is widely used because of its simplicity and control efficiency. The output of the PID controller calculated from the feedback error in the time domain is expressed as Equation 17.

$$u(t) = K_p e(t) + K_i \int e(t)dt + K_d \frac{de}{dt} \quad (17)$$

Where

e is the tracking error which is the difference between the desired output (x) and measured output (y).

The PID controller calculates the derivative and the integral of the input error signal (e) with respect to time. This control signal (u) expressed by Equation 17 is fed to the system and a new output signal (y) is obtained. The new output signal (y) is fed back and compared to the desired output (x) in order to determine the new error signal (e). The controller executes control action based on this new error signal and the system adjusts until the error signal becomes insignificant.

Taking the Laplace transform of Equation 17, the transfer function of the PID controller is expressed as Equation 18.

$$K_p + \frac{K_i}{s} + K_d s = \frac{K_d s^2 + K_p s + K_i}{s} \quad (18)$$

Where; K_p is the proportional gain, K_i is the integral gain and K_d is the derivative gain.

The PID controller is thereafter defined in the MATLAB-Simulink environment using the transfer function model obtained. The goal of the PID control is to increase the rise time, decrease the settling time with minimal percent overshoot and zero steady state error.

3. Results and Discussion

The use of the PID control is an iterative process involving random changes in the values of the proportional, integral and derivative terms until the system is fine-tuned having zero steady state error. Upon iteration, the proportional term K_p was observed to increase the rise time with increase in the percent overshoot and decrease in the steady state error. No significant changes was observed in the settling time. The addition of the integral term K_i eliminated the steady state error but with decrease in the rise time and the oscillatory response of the system (percent overshoot). However, the derivative gain (K_d) improves the stability of the system with significant damping which reduced the percent overshoot and decrease in the settling time. The PID gains was also observed to balance the system's performance in terms of the rise and settling time, bandwidth, stability margin and percent overshoot. Figure 3 shows the step plot for the reference tracking of the tuned response. An increase in the closed loop bandwidth (35 rad/sec) was observed to increase the rise time and an increase in the phase margin (90^o) was observed to significantly reduce the percent overshoot and settling time thereby improving the system's stability (Figure 3).

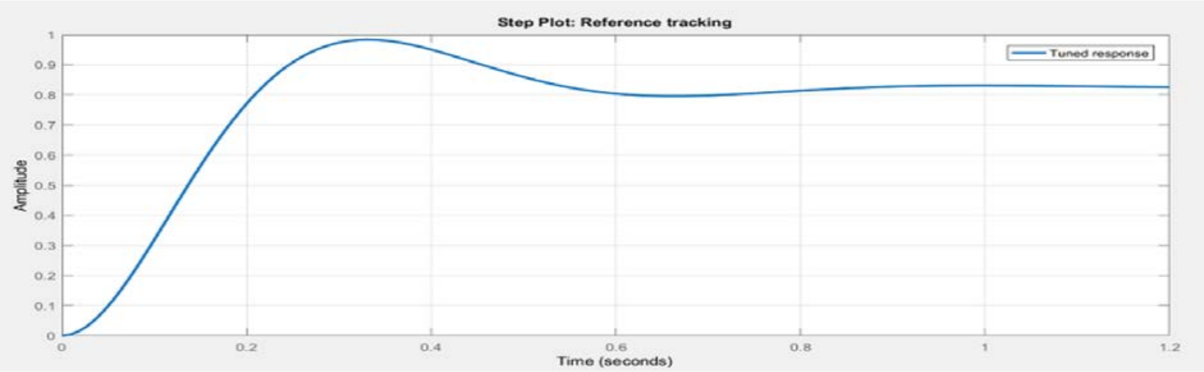

Figure 3: Step plot of system's reference tracking

Figure 4 shows the effect and control of vertical acceleration of the rail car body during movement. The overshoot of vertical acceleration of the rail car body due to disturbance was observed as 0.35 m/s^2. This is significant enough to cause ride discomfort. However the upon fine tuning, the vertical acceleration was eliminated thereby improving the ride comfort and stability.

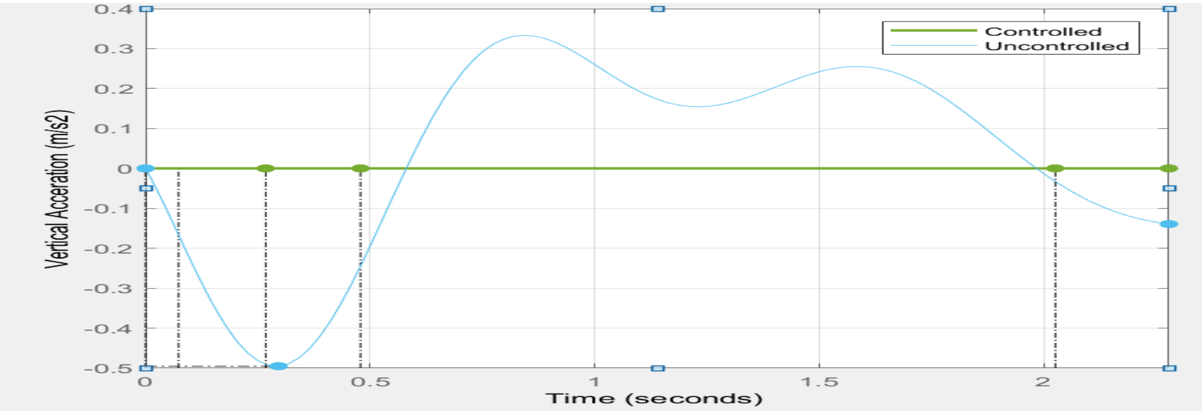

Figure 3: Control of vertical acceleration

Figure 4 shows the angular acceleration of the rail car body in the lateral direction which pushes the rail car sideways while moving along a curved rail track. The graph of the system in an uncontrolled state indicates the system is under damped hence there is significant oscillatory response for a time of 14 seconds before the system can regain its balance and stability. This will result in ride discomfort. Upon fine tuning, the control action of the PID reduces the settling time to less than 2 seconds with the percent overshoot highly negligible.

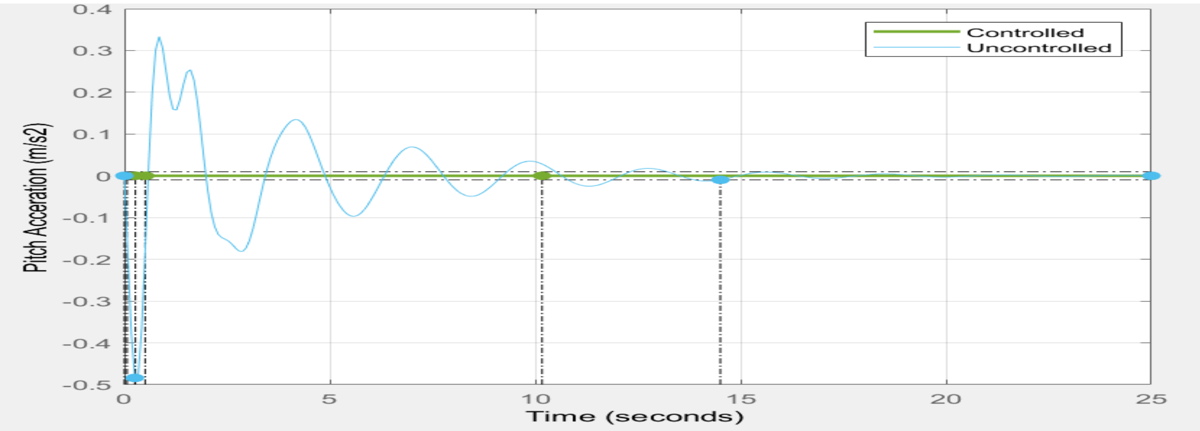

Figure 4: Control of pitch acceleration

Figure 5 studies the rate at which the control system prevents the influence of external disturbances from altering the system's performance. The amplitude of oscillation in response to the external disturbances was highly insignificant with a percent overshoot of about $1.2 \times 10^{-8}\ mm$ due to the efficient control action of the PID. However the settling time which represents the time it takes the system to regain balance after the disturbance is about 12 seconds and can be improved for better system's performance by fine tuning the derivative gain.

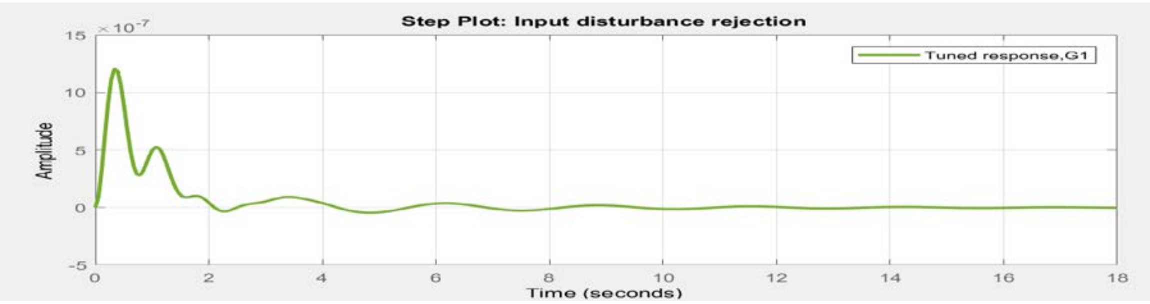

Figure 5: Plot of input disturbance rejection

Figure 6 studies the rate at which the control system prevents the influence of internal disturbances from altering the system's performance. The amplitude of oscillation in response to the internal disturbances decreases from 1 mm to 0. This points to the fact that the control action of the PID control is highly efficiently preventing internal disturbances from altering the desired output. However, the settling time which is 14 seconds can also be improved for better system's performance by fine tuning the derivative gain.

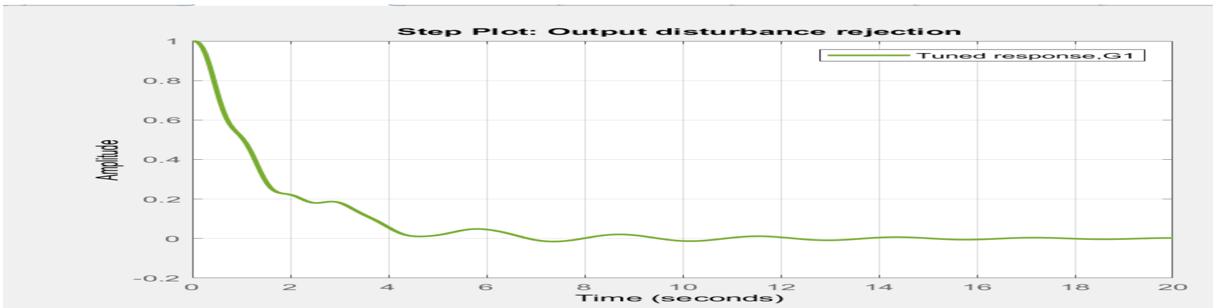

Figure 6: Plot of output disturbance rejection

4. Conclusion

A PID controlled active suspension system was modelled and simulated in a MATLAB-Simulink 2017 environment. The goal of the PID control is to increase the rise time, decrease the settling time with minimal percent overshoot and zero steady state error. Upon fine tuning, the proportional term was observed to improve the rise time, while the integral term sufficiently eliminated the overshoot and the derivative term was also observe to reduce the percent overshoot via adequate damping. The control system also proved efficient in rejecting the input and output disturbances which could alter the system's performance. Hence, the efficient control of the system via the PID significantly reduced the effect of vibrations due to disturbances thereby increasing ride comfort. The use of Fuzzy-PID control system is recommended for optimum performance of the system.

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