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Vibrations control of light rail transportation vehicle via PID type fuzzy controller using parameters adaptive method

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Abstract

In this study, a conventional PID type fuzzy controller and parameter adaptive fuzzy controller are designed to control vibrations actively of a light rail transport vehicle which modeled as 6 degree-of-freedom system and compared performances of these two controllers. Rail vehicle model consists of a passenger seat and its suspension system, vehicle body, bogie, primary and secondary suspensions and wheels. The similarity between mathematical model and real system is shown by comparing uncontrolled simulation results and vibration measurements. For carrying a comfortable travel, rail vehicle body and passenger seat vibrations are minimized by adding a controller between rail vehicle body and bogie in the model of this system. To control vibrations actively, a PID type fuzzy controller which is obtained by combining fuzzy PI and fuzzy PD controllers is preferred because of its robust character and superior performance. The PID type fuzzy controller using parameter adaptive method is designed by tuning the parameters online. Also, obtaining higher performance from this controller is studied.

Key Words: *Railway system, vibration control, PID type fuzzy controller, parameters adaptive method*

1. Introduction

Fixed-guide way systems like railways produce significant vibratory motions that are known to affect passenger comfort. Especially with the development of the technology, expected comfort level by passengers is increased. In this regard, actively control of the vibrations which affects passenger comfort adversely in rail vehicles, has been become an important engineering problem. In this study, rail vehicle vibrations are controlled actively by using an actuator on secondary suspension, which is between rail vehicle body and bogie. For this purpose, rail vehicle is modeled as 6 degrees of freedom by taking into consideration passenger seat with its suspension system, vehicle body, bogie, primary-secondary suspensions and wheel sets. Because of its simplicity and ease of application, a quarter rail vehicle model is preferred in many studies and this study. A real rail irregularity effect

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as a disruptive input that cause to vibrations, is performed to the light rail vehicle system. In the last decade, many researchers applied some linear and non-linear control methods to rail vehicle models. Foo and Goodall applied active control to secondary suspensions which placed between rail vehicle body and bogie of a railway vehicle to minimize vertical accelerations of the vehicle body [1]. Guclu investigated the dynamic behavior of a non-linear 8 degrees of freedom vehicle model having active suspensions and a PID controlled passenger seat [2]. For another article, he studied this model for a fuzzy logic controlled passenger seat [3]. Yagiz and Gursel represented a sliding mode control method (SMC) to improve the ride comfort of a railway vehicle with flexible body [4]. Metin and Guclu designed a FLC to control rapid transit system vibrations actively under an ondulation effect [5]. Vibrations in rail vehicles which modeled 6 degrees of freedom are suppressed by designing fuzzy PID controller [6], [7]. In an other study, 11 degrees of freedom rail vehicle vibrations are controlled under a sinusoidal rail irregularity effect by designing two different control strategies, FLC and classical PID controller then the results are compared. Then, again on a 6 degree of freedom quarter-rail vehicle model for control of vibrations, PID, fuzzy PID and a PID type fuzzy controller with self-tuning scaling factors are used and the performances are compared [8]. In this study, a random rail irregularity measured from the real system is considered as disruptive effects. Vibrations on a travelling light rail vehicle are controlled actively using two different control algorithms: First, a PID type fuzzy controller design that contains proportional, integral and derivative control effects, is tried to obtain the necessary performance. Then, in order to increase controller performance, a parameters adaptive PID type fuzzy controller that sets the scaling factors online is designed and a better control was carried out. The performances of both controller designs were examined in the time domain by comparing with the obtained results at the end of the application.

2. Dynamic model of six degrees of freedom light rail transport system

Figure 1 demonstrates a conventional quarter rail vehicle model. The inputs of these systems such as track inputs, load changes etc., have been applied to the mechanical system and outputs are the acceleration and displacement levels on the vehicle body and passenger seat, etc. Passive system response depends on the mass, spring parameters, damping rates and suspension geometry. Active suspension structure is a dynamic system which has more complexity than the passive suspension. The performance of an active suspension consists of sensors, actuators and controllers algorithms and hardware. Based on this quarter rail vehicle model, M_p is the passenger seat mass, M_c rail vehicle body mass, M_b is the bogie mass, J_b is the bogie inertia, M_{w1} and M_{w2} are first and second wheel masses which placed under the bogie. k_p, c_p are the passenger seat suspension stiffness, damping coefficient and k_2, c_2 are the secondary suspension stiffness, damping coefficient respectively; k_{11}, k_{12} and c_{11}, c_{12} are the primary suspension stiffness, damping coefficients; u is the forces generated by the actuator. k_{h1} and k_{h2} are the Hertzian spring stiffness these represent the wheel rail contact in linear form. The values of these parameters are given in the Appendix. This model has six degrees of freedom; these are z_p, z_c, z_b, θ_b, z_{w1} and z_{w2}. z_p is the passenger seat vertical movement, z_c is vertical movement of the rail vehicle body, z_b is vertical movement of the bogie, θ_b is the bogie pitch, z_{w1}, z_{w2} and z_a are the vertical movement of the front and rear wheels and first axle box, respectively. The equations of motion of the railway vehicle are obtained by the use of the Lagrange Equation. The equation of motion of the system is:

$$[M]\ddot{z} + [C]\dot{z} + [K]z = Fz + Fu \tag{1}$$

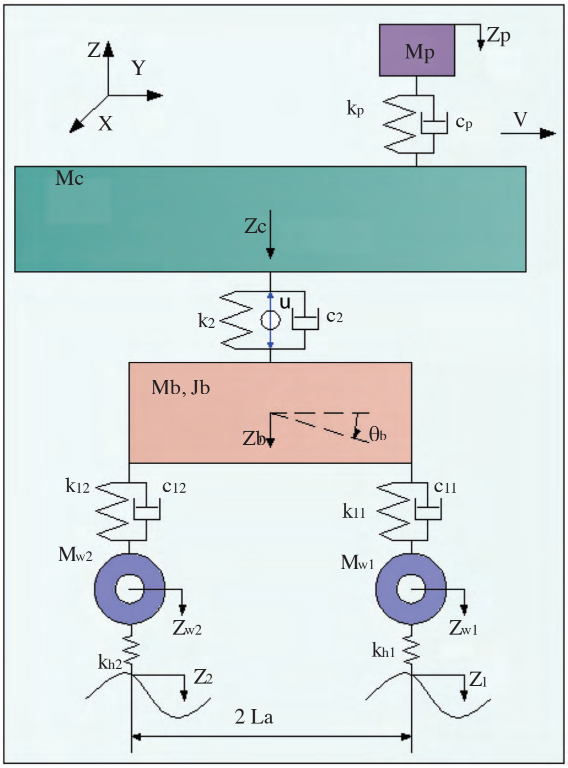

Figure 1. A quarter rail vehicle model.

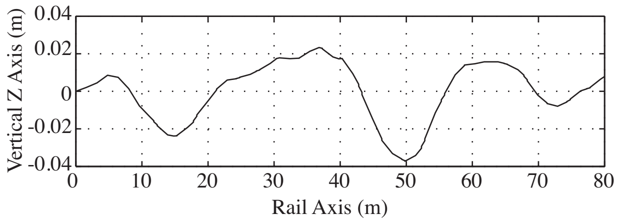

Figure 2. Real rail irregularity.

where, $z=[\,z_p\ \ z_c\ \ z_b\ \ \theta_b\ \ z_{w1}\ \ z_{w2}\,]^T$, $F_z=[\,0\ \ 0\ \ 0\ \ 0\ \ k_{h1}.z_1\ \ k_{h2}.z_2\,]^T$ and $F_u=[\,0\ \ u\ \ -u\ \ 0\ \ 0\ \ 0\,]^T$. F_z is the force resulting track irregularity motion, F_u is the control force applied by an actuator; $[M]$, $[C]$ and $[K]$ are mass, damping and stiffness matrices and were given in [6]. L_A and V demonstrate axle spacing and rail vehicle speed respectively. $z_1(t)$ and $z_2(t)$ represent the inputs to the wheel masses by irregularities on the track. A measured real rail irregularity as a disruptive effect has been used in simulations (Figure 2). Vibration measurements are performed on the same railway part. During vehicle-track interaction, the forces are transmitted by means of the wheel-rail contact area. The relationship between force and contact surface and the linearization of Hertzian spring stiffness k_h are clearly determined by Esveld [9].

3. Vibration measurements of the used rail vehicle model

"PULSE Type 3560C-E01" computer-based system, belongs to the company Bruel & Kjaer is used for measurements and analysis. Pulse-6 Channel DynX Input Module (ICP & Charge) (3035) and supported measurement device have the opportunity to be able to measure up to 160 dB for each channel. Bruel & Kjaer branded, 3-axis, high-precision accelerometer, the model 4506B-003 is used. Devices used for measurement are appropriate according to DIN 45669-1 and ISO 2954 standard. In order to perform the necessary vibration measurements on the ABB (Asea Brown Boveri Ltd.) light rail transport vehicle (Figure 3) which is modeled as quarter rail vehicle, the rail vehicle park of Istanbul Transportation CO. General Directory in Esenler was chosen as the measurement area. Vibration measurements between 0-400 Hz were carried out simultaneously from rail vehicle axle box, bogie and vehicle ground at 30 km/h (Figure 3). Appropriateness to reality of the model is shown by

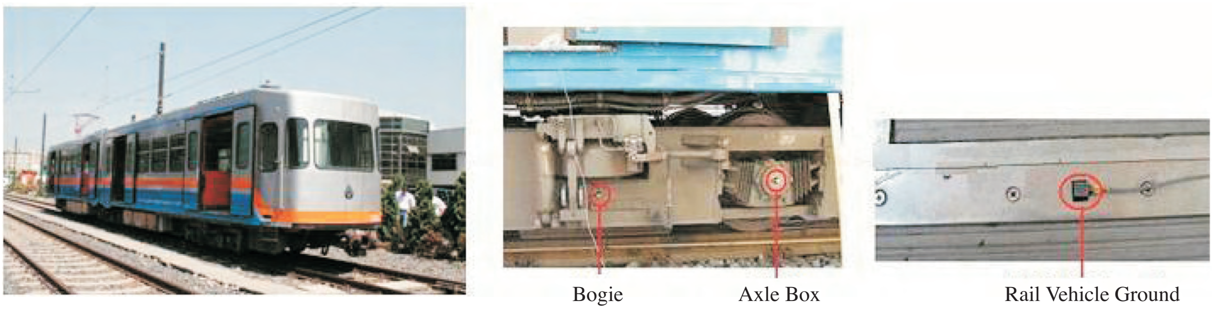

Figure 3. ABB Light rail transportation vehicle and accelerometer placements.

plotting the time domain displacement and acceleration results in the simulation using the established model and the vibration measurements, carried out from rail vehicle axle box, bogie and vehicle ground when 30 km/h operation speed, consecutively in Figure 4.

As shown in Figure 4, when the simulations are carried out under the effect of real rail irregularity in 6 degrees of freedom rail vehicle model, it is observed that the results of the axle moving with the wheel as the closest rail vehicle element with the railway are almost exactly overlapped with the real measurement results. The results obtained from bogie are very similar. Although real system has some nonlinearity, achieving similar results on the ground of the rail vehicle body from simulations and measurements shows the suitability of used model. So, important information about the real system can be obtained from this model.

4. Design of the controllers

4.1. Design of PID type classical fuzzy controller (CFC)

In the literature, the large number of controller structure is recommended for fuzzy PID (including PI and PD structures). A conventional fuzzy PID controller (CFC) requires 3 inputs, hence the rule base is 3-dimensional, therefore it is more difficult to design. However, the PID type fuzzy controller has only 2 inputs and the rule base is 2-dimensional and its performance is better than fuzzy PI and PD controllers [10]. Woo et. al. [10] and Karasakal et. al. [11] explained how to design a PID type classical Fuzzy Controller, clearly. In this study, Matlab Simulink with Fuzzy Toolbox is used. Input variables of FLC are given in Table 1. Where, the linguistic variables P, N, Z, B, M, S represent *Positive*, *Negative*, *Zero*, *Big*, *Medium* and *Small*, respectively. The membership functions for both scaled inputs (e, de) and output (u) of the controller have been defined on the common interval [-1, 1]. For the input scaling factors error (e), and derivative of error (de), triangular membership functions are defined. These functions cover each other by 50%. Improving the performance of the controller is aimed by selecting commonly used triangular membership functions. By using the input and output membership functions, 15 rules are written in rules base. For input and output defined [-1 1] range, membership functions have been scaled by K_e, K_d, β and α scaling factors. The values of scaling factors are presented in the Appendix. Through Table 1, the following rules base has been used. All the rules are written similarly using Mamdani Method to apply to fuzzification as below. In this study, the Centroid Method is used in defuzzification.

IF error (e) is NB and derivative of error (de/dt) is N THEN control force (u) is NB.

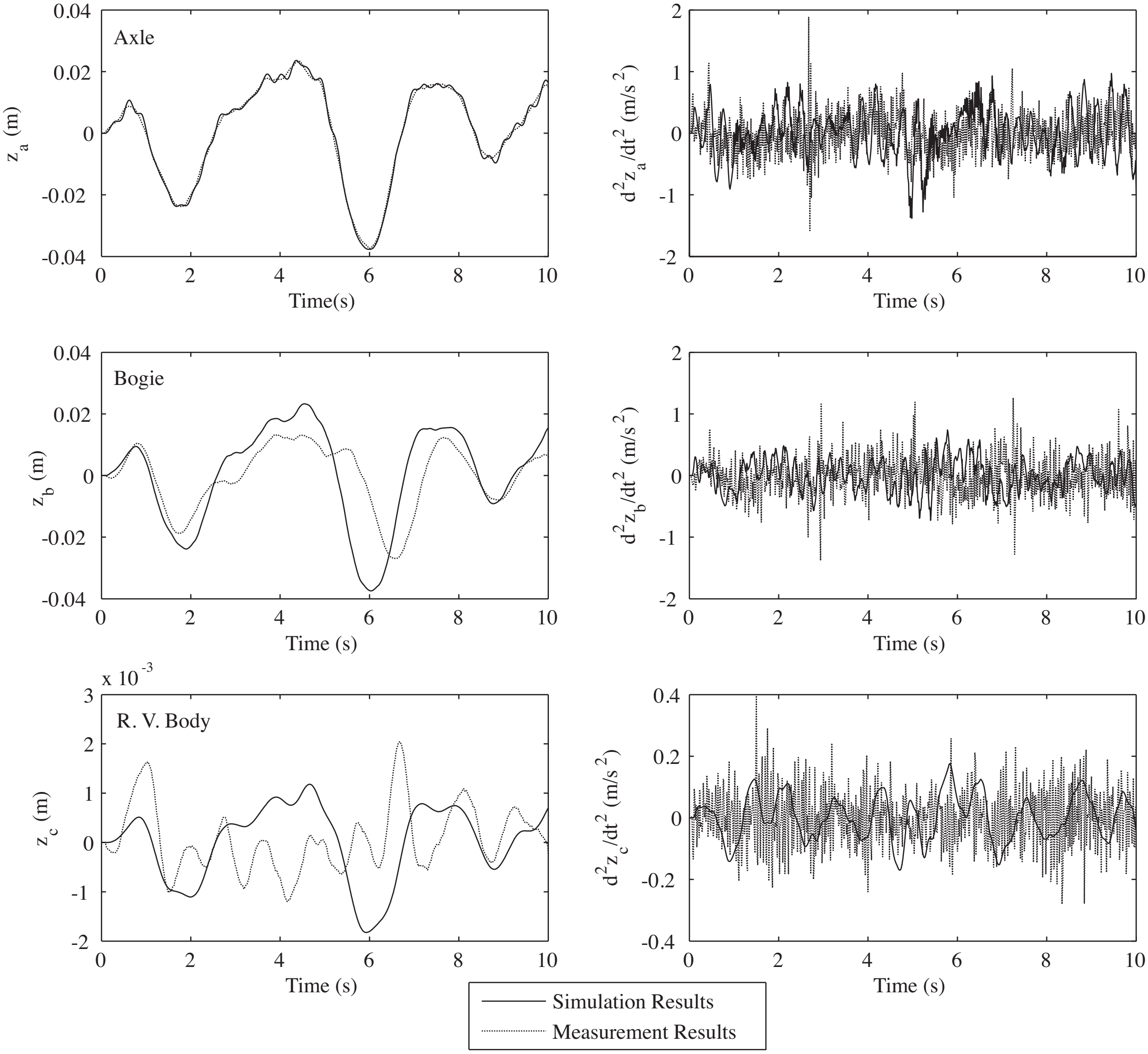

Figure 4. Comparison of measurement and simulation results.

Table 1. Rule base for the FLC.

	Derivative of error (de/dt)			
	Case	N	Z	P
Error (e)	NB	$U-NB$	$U-NM$	$U-NS$
	NS	$U-NM$	$U-NS$	$U-Z$
	Z	$U-NS$	$U-Z$	$U-PS$
	PS	$U-Z$	$U-PS$	$U-PM$
	PB	$U-PS$	$U-PM$	$U-PB$

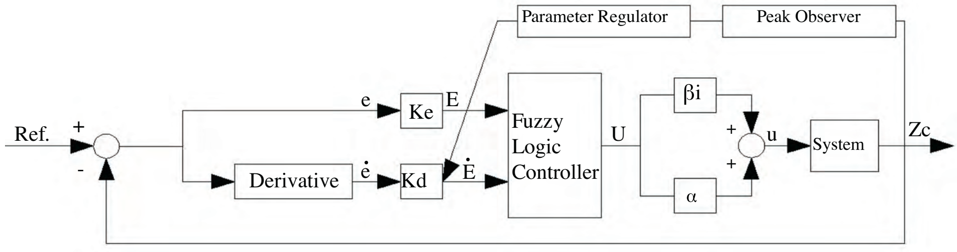

Figure 5. Block diagram of PAFC.

4.2. Design of PID type fuzzy controller using parameters adaptive method (PAFC)

Only by setting the scaling factors, the requisite of real time control can be achieved. Therefore, in order to obtain fuzzy logic control application, self-adjustment of the scaling factors is an important requirement [10]. It is known, the integration component of PID type fuzzy controller has an important role on the fuzzy control system performance. When the integration component is too weak, the response is slow; also when it is too strong, the system becomes unstable. Therefore, further improvements may be required on proposed controller. The equivalent integration component of the fuzzy controller, varying with time is a solution for performance and stability problem. Receiving a higher value in the first phase of system response can be provided, then reducing gradually with time to increase the system damping and it makes the system more stable. In this way, a rapid rise time and a short settling time of system response is seen. It can be seen in proportional, integral and derivative gains of PID type fuzzy controller, when β parameter is decreased, integration component of control is decreased and so the damping of the system is increased and the system becomes more stable. Proportional component contains the multiplication β and K_d. When β is decreased, proportional control component is decreased too. Thus, the reaction of the control system against the error will be slowed down. When β is reduced, K_d is increased at the same rate; equivalent proportional control force does not change and system protects the rapid response against errors. In addition, it can be seen, when K_d is increased, the equivalent derivative control component will be increased. However, derivative control law will increase the resistance against to overshoot and oscillation of the system; it does not harm the system [12]. According to this idea, the parameter adaptive PID type fuzzy controller (PAFC) will be designed. Proposed controller consists of PID type fuzzy controller, peak observer and a parameter regulator. In Figure 5, a PAFC block diagram is represented.

CFC is defined in the previous section. Peak observer monitors the output of the system and generates a signal at each peak time and measures the absolute peak value. Parameter regulator simultaneously sets the controller parameters β and K_d at every peak time for that peak value. Adjustment algorithm of scaling factors and integral gain are as follows:

$$K_d = K_{ds}/\delta_k, \; \beta = \delta_k \beta_s \tag{2}$$

where, K_{ds} and β_s are the initial value of K_d and β, respectively. δ_k is absolute peak values at peak times $t_k \, (k = 1, 2, 3, ...)$.

5. Simulation results

The comfort level of passengers is should be examined in the evaluation actual driving performance. When looking for the safety of driving of this assessment is based on displacement responses and depending on the acceleration response of comfort level will be seen. Controllers are designed taking in the consideration the shape of both responses and the responses remain within the permissible limits to be provided. All parameters belong to the both controllers were determined using the Genetic Algorithm Toolbox, according to the following Integral of Square Error(ISE) formula. The values of these parameters are given in the Appendix.

$$S_{ISE} = \int_{t=0}^{10} e^2(t)dt \tag{3}$$

The performance value of ISE is $5,97.10^{-9}$. All controllers of the system are designed by providing the limitation of control forces of equal value. With this application form, the system responses are compared under the same control force. In Figures 7 and 8, displacement and acceleration responses of axle box, bogie, vehicle body and passenger seat by applying CFC and PAFC are shown in the time domain, respectively. They show the improvement in simulation results by applying PAFC to system. The amplitude of vibrations is minimized with the both controller. When trying to suppress the light rail vehicle vibrations by using CFC and PAFC as seen in Figures 6 and 7, the maximum value of vibrations in the cases with controller and without controller are seen in Table 2.

Table 2. Maximum vibration values

Displacement (m)			
Measurement Location	Without Controller	With CFC	With PAFC
Vehicle Ground	0.001181	0.0003067	$1.544e-012$
Seat	0.001064	0.000336	$1.699e-12$
Acceleration (m/s^2)			
Measurement Location	Without Controller	With CFC	With PAFC
Vehicle Ground	0.1758	0.007966	0.006432
Seat	0.2098	0.004241	$1.558e-10$

As understood here, when the parameters of PID type fuzzy controller are set online according to disruptive effects, the controller performance can be greatly improved.

Between these controllers, CFC shows relatively low performance especially in terms of displacement. To make an assessment in terms of passenger comfort for the passenger seat acceleration in a matter, the superiority of PAFC will be seen here as well. In the rail transport body, the controller performance results are similar to the results in the passenger seat. Both control force changes in control methods are shown in Figure 8.

6. Conclusions

In this study, 6 degrees of freedom LRT vehicle model has been taken. At 30 km/h vehicle speed case, axle box, bogie and vehicle ground vibration measurements results and simulation results from the established model were compared. The suitability of established model has demonstrated by achieving similar results. In the next section, the controller performances were compared by performing PID type fuzzy controller and parameter adaptive PID type fuzzy controller applications at 60 km/h travelling speed. In this research,

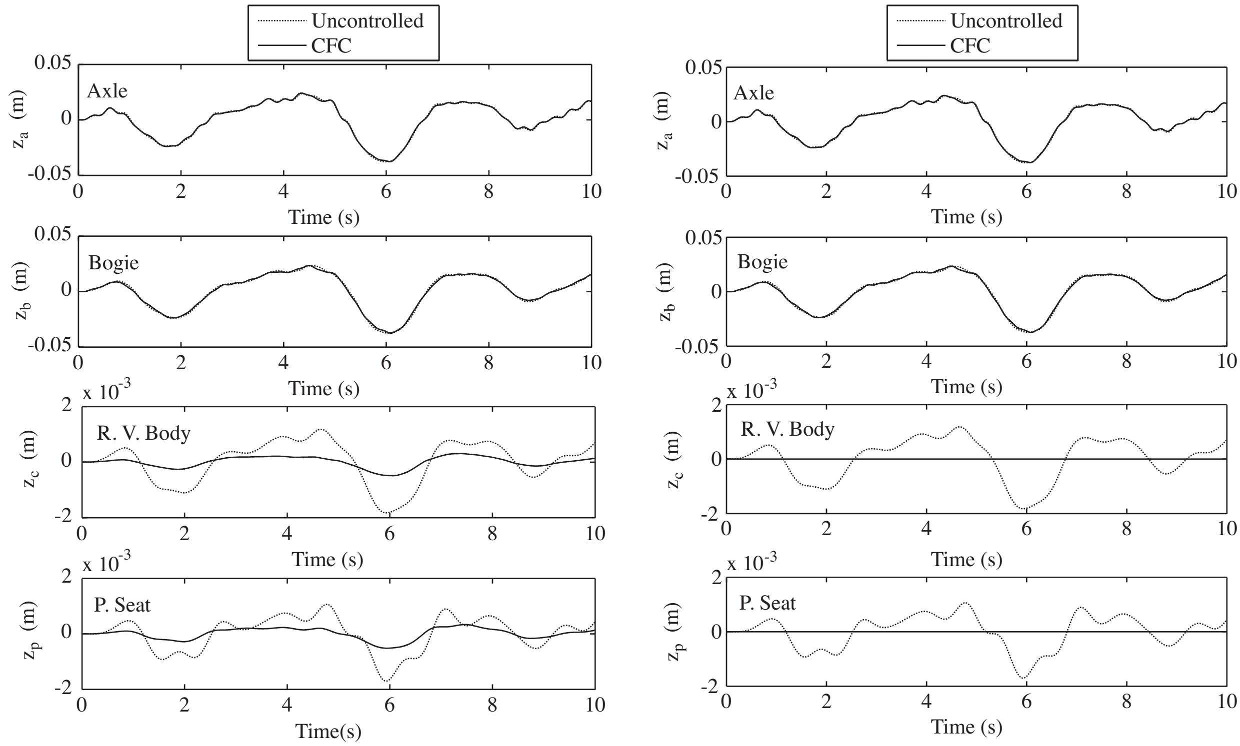

Figure 6. The displacement responses of LRT vehicle axle box, bogie, body and passenger seat.

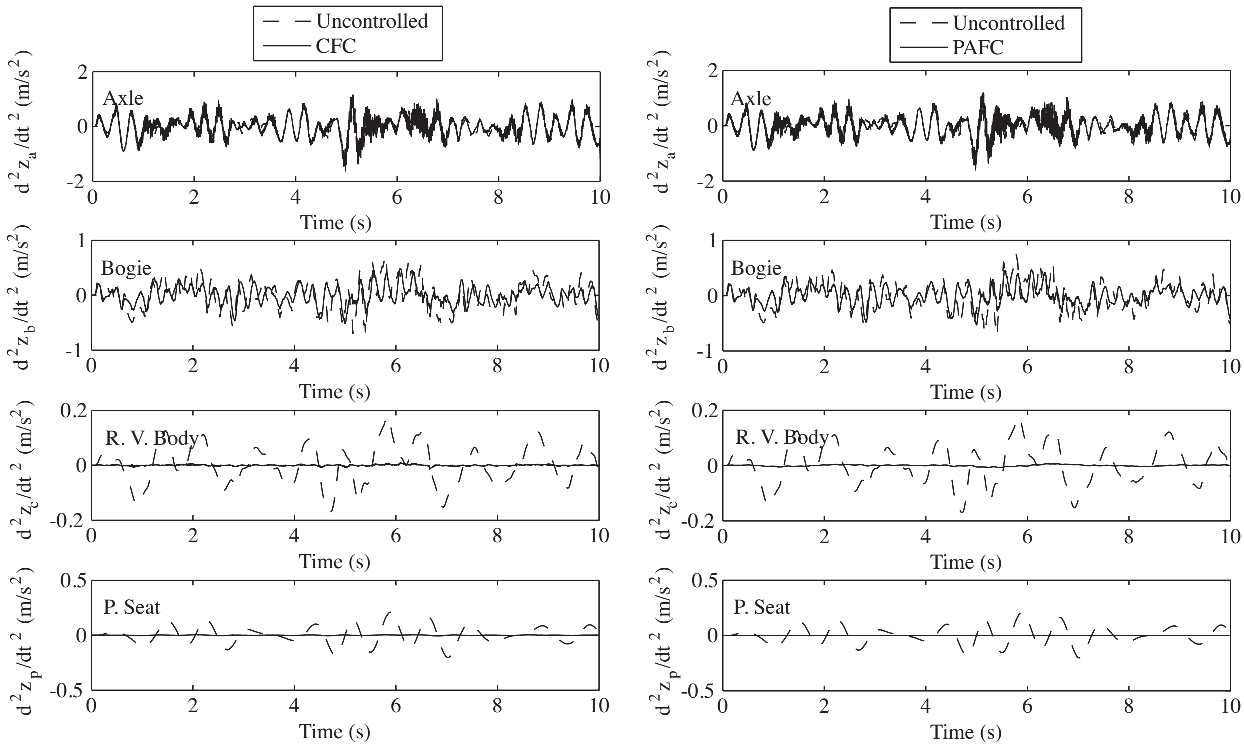

Figure 7. The acceleration responses of LRT vehicle axle box, bogie, body and passenger seat.

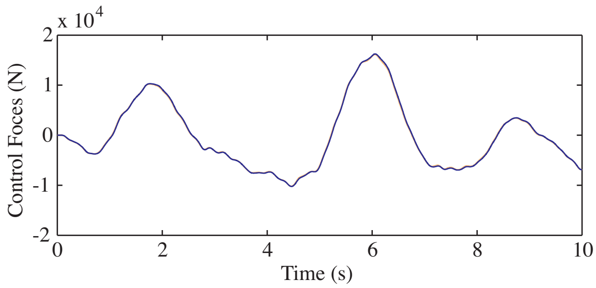

Figure 8. Control force changes for both controllers.

especially passenger seat and vehicle body vertical vibrations were examined. Displacement and acceleration responses are shown for with and without controller situations in time domain. It is seen that, the obtained displacement and acceleration results were minimized when compared with the uncontrolled and PID type fuzzy controller cases. However, this controllers performance was improved by using self adapting structure versus changing disruptive input and tuning the parameters online. At the end of the comparison, it was seen that, the required performance was achieved successfully by using both controller when passenger comfort was analyzed. But, parameter adaptive PID type fuzzy controller was found to provide superior performance.

7. Acknowledgements

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A. Appendix

Six degrees of freedom LRT vehicle model parameters

Controller Parameters		Mass Values	Stiffness Values	Damping Values	Others
CFC	PAFC				
k_e=40.02	k_e=40.02	M_p=80 kg	k_p=6000 N/m	c_p=200 Ns/m	L_A=1.28 m
k_d=12.20968	k_d=12.20968	M_c=5333 kg	k_2=430000 N/m	c_2=20000 Ns/m	V=60 km/h
β=8.90322e+5	β=8.90322e+5	M_b=1307.5 kg	k_{11}=1220000 N/m	c_{11}=40000 Ns/m	
α=9.0161e+5	α=9.0161e+5	M_{w1}=906.5 kg	k_{12}=1220000 N/m	c_{12}=40000 Ns/m	
		M_{w2}=906.5 kg	c_h=1e+11 $N/m^{3/2}$		
		J_b=738 kgm^2			