Jurnal Teknologi

Full paper

Modeling and Control of a Nonlinear Active Suspension Using Multi-Body Dynamics System Software

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Article history

Received :17 September 2013
Received in revised form :
26 December 2013
Accepted :30 January 2014

Graphical abstract

Virtual animation by CarSim for passive suspension system.

Virtual animation by CarSim for active suspension system with CCNF.

Abstract

This paper describes the mathematical modeling and control of a nonlinear active suspension system for ride comfort and road handling performance by using multi-body dynamics software so-called CarSim. For ride quality and road handling tests the integration between MATLAB/Simulink and multi-body dynamics system software is proposed. The control algorithm called the Conventional Composite Nonlinear Feedback (CCNF) control was introduced to achieve the best transient response that can reduce to overshoot on the sprung mass and angular of control arm of MacPherson active suspension system. The numerical experimental results show the control performance of CCNF comparing with Linear Quadratic Regulator (LQR) and passive system.

Keywords: Nonlinear active suspension; ride comfort; road handling; conventional composite nonlinear feedback control; linear quadratic regulator; control algorithm

Abstrak

Kertas ini menerangkan model matematik dan kawalan sistem penggantungan tak linear aktif untuk keselesaan perjalanan dan jalan prestasi pengendalian dengan menggunakan perisian dinamik multi-badan yang dipanggil CarSim. Untuk kualiti pemanduan dan pengendalian jalan menguji integrasi antara MATLAB/Simulink dan pelbagai badan perisian sistem dinamik adalah dicadangkan. Algoritma kawalan dipanggil Komposit Konvensional Linear Maklumbalas (CCNF) kawalan telah diperkenalkan untuk mencapai keputusan yang terbaik sementara yang boleh mengurangkan terlajak jisim badan kereta dan sudut bagi lengan kawalan. Keputusan eksperimen berangka menunjukkan prestasi kawalan CCNF membandingkan dengan Kawalan Linear Kuadratik (LQR) dan sistem pasif.

Kata kunci: Penggantungan aktif tak linear; keselesaan pemanduan; pengendalian jalan raya; komposit kawalan maklum balas tak linear konvensional; kawalan linear kuadratik, kawalan algoritma

1.0 INTRODUCTION

Performances in feedback control systems are often been specified in terms of the step response. For a fast response, a small damping ratio, however, it will increase the overshoot. Lin *et al.* proposed the influential idea of the Conventional Composite Nonlinear Feedback (CCNF) control [1]. In CCNF primarily damping ratio is kept very low to ensure fast response and as output approaches set point system is made very much damped to avoid overshoot by the nonlinear part. The CCNF was proposed by G.Y. Cheng *et al.* in previous research work on the application of DC motor transient performance in tracking the general for high-speed XY table positioning mechanism [2]. B.M. Chen *et al.* research work explores the design of Conventional Composite Nonlinear Feedback (CCNF) control law for a Hard Disk Drive (HDD) servo mechanism system for tracking the reference signal [3]. X. Yu *et al.* using Optimal CNF to control the gantry crane system in a target position as fast as possible without any overshoot while the load is limited in a specified swaying angle [4]. The active suspension systems were established over the last three decades, and a lot of papers have been published. Most of the papers focus on the computation of the expected force resulting from the road disturbances in the various vehicle conditions as mention by D. Hrovat [5]. The current research work by M.F. Ismail *et al.* in the vehicle active suspension system was used the CCNF as a controller. The CCNF is successfully applied to improve the suspension deflection, velocity of car body and velocity of car wheel form very high overshoot and fast rising time. A control performance comparison between a nonlinear and linear active suspension system is done [6, 7]. However, the simulation work is limited in an MATLAB / Simulink. There are various control techniques have been used in previous research work in an active suspension system. Y.M. Sam *et al.* proposed the sliding mode control technique with proportional integral sliding surface. The control algorithm used to reduce steady-state error occurs in an active suspension system [8]. H. Du *et al.* propose H-Infiniti control strategy as a robust control to the active suspension system [9]. A. Abu Khudhair *et al.* and M.B.A Abdelhady proposed fuzzy logic control to improve ride comfort and road handling performance [10,20]. X. Zhao *et al.* discussed generally body acceleration based on ISO 2631 standard. The results show that the speed for the vehicle was a larger influence of the vibration intensity [12]. X. D. Xue *et al.* did a study on the typical vehicle suspension system. The vehicle suspension can be classified as an active suspension system, semi-active suspension system, and passive system [13]. C. Sandu *et al.* developed a multi-body dynamics model of a quarter-car test-rig equipped with a McPherson strut suspension and applied a system identification technique on it. The MacPherson strut nonlinearity came from the magnitude of strut angle change it experiences through its travel. The greater the angle change, the greater the nonlinearity [14]. T. Shim *et al.* discussed on the vehicle roll dynamics that strongly influenced by suspension properties such as roll centre height, roll steer, and roll camber. The effects of suspension properties on vehicle roll response have been investigated using multi-body vehicle dynamics software. The experimental results can be useful in initial suspension design for rollover prone vehicles, in particular, sports utility vehicles and trucks [15]. B.C. Chen *et al.* proposed sliding mode control (SMC) for semi-active suspensions to achieve ride comfort and handling performance simultaneously. The author(s) claimed that SMC could reduce the sprung mass acceleration without increasing tyre deflection [16]. W. Jun proposed H-Infiniti controller for a full vehicle suspension system to improve vehicle ride comfort and steady-state handling performance. Based on the results, the author(s) claimed that the proposed controller improved handling without sacrificing vehicle cornering stability [17]. M. Kaleemullah *et al.* study on control performance of LQR, Fuzzy and H-Infiniti for active suspension system quarter car model. The authors claim the propose controller is better than the passive system as results shown from the simulation [18]. K. Chen *et al.* discuss on the fundamental for experimental measurement of base dynamic parameters and the application of base dynamic parameter estimation techniques in suspension design [19]. Y. Watanabe *et al* study on mechanical and control design of a variable geometry active suspension system. The authors also proposed PD controller and neural network as control performance comparison. Neuro control is better than PD controller based on the obtained results [21]. D.A Crolla *et al.* presented a state observer design for an adaptive vehicle suspension. Two main issues have been investigated issues, (1) the selection of measurement signals in relation to estimation and sensors need, and (2) the effects' variations in both road profiles and vehicle parameters on estimation accuracy [22]. D. A. Crolla *et al.* proposed Linear Quadratic Gaussian (LQG) for an active suspension. The LQG controller, incorporating a weighting controller, state observer and parameter estimator, was successful in adapting to road input conditions and also to vehicle parameters [23]. D. C. Chen *et al.* recognized the need for improved understanding of links between subjective and objective measures of vehicle handling an extensive program of instrumented testing and driver evaluation was conducted with a view to correlating results from the different sets of data [24]. L. Jung-Shan *et al.* proposed nonlinear Backstepping controller for active suspensions system. The results shown that the proposed controller is successfully applied to the active suspension systems for ride comfort and road handling [25].

The mathematical modeling of active suspension system will be explained in Section 2. The CCNF controller design is discussed in Section 3 and Section 4 will be explained the simulation results and discussion and LQR controller for control performance comparison.

2.0 SYSTEM MODELING

This section will explain details on a nonlinear active suspension based on the MacPherson strut. A quarter car mathematical modeling based on K.S. Hong *et al.* [11]. The quarter car model is consisting of the two degrees of freedom. The quarter car model is constructed based on the Newton law as follows:

$$(m_s + m_u)\ddot{z}_s + m_u l_C \cos(\theta - \theta_0)\,\ddot{\theta} - m_u l_C \sin(\theta - \theta_0)\,\dot{\theta}^2 + k_t(z_s + l_C(\sin(\theta - \theta_0)) - \sin(\theta_0) - Z_r) = -f_d \tag{1}$$

$$m_u l_C^2\ddot{\theta} + m_u l_C \cos(\theta - \theta_0)\,\ddot{Z}_s + \frac{c_p b_l^2 \sin(\alpha' - \theta_0)\dot{\theta}}{4(a_l - b_l\cos(\alpha' - \theta))} + k_t l_C \cos(\theta - \theta_0)(Z_s + l_C(\sin(\theta - \theta_0) - sin(-\theta_0)) - Z_r) - \frac{1}{2}k_s \sin(\alpha' - \theta)\left[b_l + \frac{d_l}{\left(c_l - d_l\cos(\alpha' - \theta)^{1/2}\right)}\right] = -l_B f_a \tag{2}$$

where $a_l = l_A^2 + l_B^2$ $\quad b_l = 2l_A l_B$ $\quad c_l = a_l^2 - a_l b_l \cos(\alpha + \theta_0)$ $\quad d_l = a_l b_l - b_l^2 \cos(\alpha + \theta_0)$

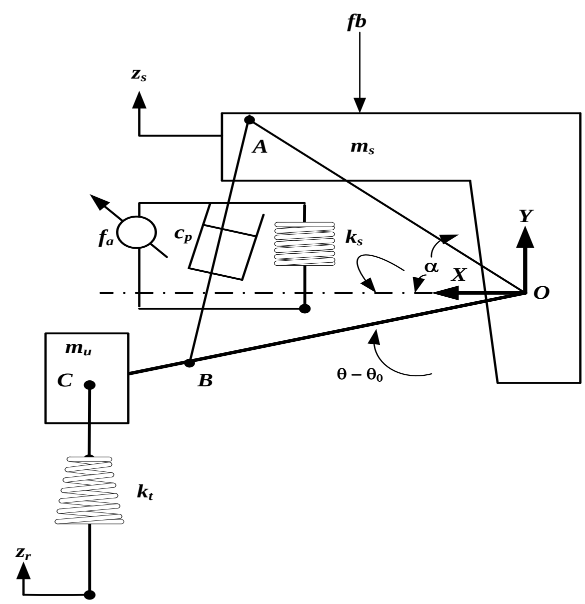

Figure 1 A free body diagram of a quarter car MacPherson strut model

Figure 1 shows a free body diagram of a quarter car MacPherson model. The model is similar to the actual car an active suspension system. Figure 1 consists of the sprung mass, m_s, signifies the car chassis, whiles the unsprung mass, m_u, and signifies the wheel assembly. The stiffness of the car body spring, k_s, and damping of the damper, c_p, signify a passive spring and shock absorber that are placed between the car body and the wheel, while stiffness of the car tire is k_t. f_b is the force applied in sprung mass. f_a is an active force applied in active suspension system. z_r is refer to the road irregularities or road profiles. l_A a distance from point 0 to point A. l_B a distance from point 0 to point B. l_C a distance from point 0 to point C. The state variables can be defined as follows:

$$[x_1\ x_2\ x_3\ x_4]^T = \left[Z_s\ \dot{Z}_s\ \theta\ \dot{\theta}\right]^T \tag{3}$$

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = f_1(x_1, x_2, x_3, x_4, f_a, z_r, f_b) \\ \dot{x}_3 = x_4 \\ \dot{x}_4 = f_2(x_1, x_2, x_3, x_4, f_a, z_r, f_b) \end{cases} \tag{4}$$

where

$$f_1 = \frac{1}{D_1}\left\{m_u l_C^2 \sin(x_3 - \theta_0) x_4^2 - \frac{1}{2} k_s \sin(\alpha' - x_3) \cos(x_3 - \theta_0) g(x_3) + c_p h(x_3)\dot{\theta} - k_t l_C \sin^2(x_3 - \theta_0) z(\cdot) + l_B f_a \cos(x_3 - \theta_0) - l_C f_b\right\}$$

$$f_2 = \frac{1}{D_2}\left\{m_u^2 l_C^2 \sin(x_3 - \theta_0)\cos(x_3 - \theta_0) x_4^2 + (m_s + m_u) c_p h(x_3) x_4 - \frac{1}{2}(m_s + m_u) k_s \sin(\alpha' - x_3) g(x_3) m_s k_t l_C \cos(x_3 - \theta_0) z(\cdot) + (m_s + m_u) l_B f_a - m_u l_C \cos(x_3 - \theta_0) f_b\right\}$$

$$D_1 = m_s l_C + m_u l_C \sin^2(x_3 - \theta_0)$$

$$D_2 = m_s m_u {l_C}^2 + {m_u}^2 {l_C}^2 \sin^2(x_3 - \theta_0)$$

$$g(x_3) = b_l + \frac{d_l}{(c_l - d_l \cos(\alpha' - x_3))^{1/2}} \quad h(x_3) = \frac{b_l^2 \sin^2(\alpha' - x_3)}{4(a_l - b_l \cos(\alpha' - x_3))}$$

$$z(\cdot) = z(x_1, x_2, z_r) = x_1 + l_C(\sin(x_3 - \theta_0) - \sin(-\theta_0)) - z_r$$

Based on [6, 7, and 11], there are some constraints need to be considered in the mathematical modelling of a MacPherson model as follows:

1. Only vertical displacement for the sprung mass Z_s.
2. θ is the angular displacement of the control arm. The linked of θ to the car body consider as a unsprung mass.
3. The value of vertical displacement and control arm will be measured from the static equilibrium point.
4. The sprung and unsprung masses are assumed to be two different elements.
5. The mass and control arm stiffness are ignored.
6. The forces are in the linear region for spring deflection, tire deflection and damping forces.

3.0 CONVENTIONAL COMPOSITE NONLINEAR FEEDBACK (CCNF) CONTROLLER DESIGN

The CCNF consists of linear control law and nonlinear control law. The state equation in (4) can be divided into two subsystems.

3.1 Two Subsystem of Nonlinear Active Suspension System

The purpose dividing state equation in (4) is to reduce disturbance and to track control variable, x_{t1c}. Then the system in equation (4) turns into:

$$\begin{cases} \begin{pmatrix}\dot{x}_{t1} \\ \dot{x}_{t2}\end{pmatrix} = A_1 \begin{pmatrix}x_{t1} \\ x_{t2}\end{pmatrix} + B_1 v, \\ y_{t1} = C_1 \begin{pmatrix}x_{t1} \\ x_{t2}\end{pmatrix}, \end{cases} \tag{5}$$

and

$$\begin{cases} \begin{pmatrix}\dot{x}_{t3} \\ \dot{x}_{t4}\end{pmatrix} = A_2 \begin{pmatrix}x_{t3} \\ x_{t4}\end{pmatrix} + B_2 x_{t1} + B_3 \dot{z}_r, \\ y_{t2} = C_2 \begin{pmatrix}x_{t3} \\ x_{t4}\end{pmatrix}, \end{cases} \tag{6}$$

where

$$v := \begin{bmatrix} -\frac{k_s l_C}{m_s} & -\frac{C_p l_C}{m_s} & 0 & \frac{C_p l_C}{m_s} \end{bmatrix} x + \frac{1}{m_s}(f_a - f_b) + \frac{(m_s+m_u)k_t}{m_u m_s}\frac{(m_s+m_s)C_p}{m_s m_u} x_{t3} \tag{7}$$

$$A_1 = \begin{bmatrix} 0 & 1 \\ a_{21} & 0 \end{bmatrix}, \quad A_2 = \begin{bmatrix} 0 & 1 \\ a_{22} & a_{23} \end{bmatrix}, \quad C_1 = [1 \quad 0], C_2 = [0 \quad 1]$$

$$a_{21} = \frac{k_t l_C sin^2(-\theta_0)}{D_1}$$

$$a_{22} = \frac{1}{D_2^2}\left\{\left[\frac{1}{2}(m_s + m_u) k_s \cos\alpha' \left(b_l + \frac{d_l}{(c_l - d_l\cos(\alpha'))^{1/2}}\right)(\cos(\alpha' + \theta_0)) - \frac{1}{2}(m_s + m_u) k_s\, sin\,\alpha' \left(\frac{d_l^2 \sin\alpha'}{2(c_l - d_l\cos\alpha')^{3/2}}\right) + m_s k_t l_C^2\, cos(-\theta_0)\right] \cdot [m_s m_u l_C^2 + m_u^2 l_C^2 sin^2(-\theta_0)] + \frac{1}{2}(m_s + m_u) m_u^2 k_s l_C^2 \quad \sin\alpha' \sin(-\theta_0) cos^2(-\theta_0)\left(b_l + \frac{d_l}{(c_l - d_l\, cos(\alpha'))^{1/2}}\right)\right\}$$

$$a_{23} = -\frac{1}{D_2}\cdot\frac{(m_s + m_u) c_p b_l^2 sin^2\alpha'}{4(a_l - b_l\cos\alpha')},$$

$$B_1 = \begin{bmatrix} 0 \\ \dfrac{(m_s + m_u) l_C}{m_s m_u l_C^2 + m_u^2 l_C^2 sin^2(-\theta_0)} \end{bmatrix},$$

$$B_2 = \begin{bmatrix} 0 \\ \dfrac{m_s k_t l_C \cos(-\theta_0)}{m_s m_u l_C^2 + m_u^2 l_C^2 sin^2(-\theta_0)} \end{bmatrix},$$

$$B_3 = \begin{bmatrix} 0 \\ \dfrac{m_u l_C \cos(-\theta_0)}{m_s m_u l_C^2 + m_u^2 l_C^2 sin^2(-\theta_0)} \end{bmatrix},$$

3.2 Linear Control Law

Equation (5) is used to design linear part in CCNF as follow to track the control variable x_{t1c}.as follow:

$$v_{1L} = F_1 \begin{pmatrix} x_{t1} \\ x_{t2} \end{pmatrix} + G_1 x_{t1c} \tag{8}$$

where

$$F_1 = -\frac{[4\pi^2 f_1^2 + a_1 \quad 4\pi f_1 \zeta_1]}{\frac{(m_s + m_u) l_C}{m_s m_u l_C^2 + m_u^2 l_C^2 sin^2(-\theta_0)}}$$

with frequency of f_1 = 10 Hz and ζ_1= 0.32 for quickly tracking. It can be checked that $(A_1 + B_1 F_1)$ is asymptotically stable.

$$G_1 = \frac{1}{C_1 (A_1 + B_1 F_1)^{-1} B_1}$$

For equation (6), the linear control law is designed as follows:

$$v_{2L} = F_2 \begin{pmatrix} x_{t3} \\ x_{t4} \end{pmatrix} \tag{9}$$

where the item of $G_2 r$ is ignored in the linear control law as the reference $r = 0$.

$$F_2 = \begin{bmatrix} 0 & \frac{\zeta_2}{\pi f_2} \end{bmatrix}$$

with frequency of $f_2 = 3.5684$Hz, the natural frequency of (6), and $\zeta_2 = 0.55$ to reduce the disturbance with the small control signal. It can be checked that $(A_1 + B_1 F_1)$ is asymptotically stable.

3.3 Nonlinear Control Law

For equation (5) the nonlinear part can be designed as follows:

$$v_{1N} = \rho_1(y_{t1}) N_1 \left[\begin{pmatrix} x_{t1} \\ x_{t2} \end{pmatrix} \quad -x_e \right] \tag{10}$$

where

$$N_1 = \frac{[4\pi^2 f_1^2 \quad \zeta_1/\pi f_1]}{\frac{(m_s + m_u) l_C}{m_s m_u l_C^2 + m_u^2 l_C^2 sin^2(-\theta_0)}},$$

$$\rho_1 = -4.53 \frac{e^{-15.685|y_{t1}|} - e^{-1}}{1 - e^{-1}}.$$

ρ_1 is to be designed. $N_1 = B_1' P_1$ in which $P_1 > 0$ is a solution of the Lyapunov equation given by

$$(A_1 + B_1 F_1)' P_1 + P_1 (A_1 + B_1 F_1) = -W_1,$$

$$W_1 = diag \left[\frac{40\pi^4 f_1^4}{\frac{(m_s + m_u) l_C}{m_s m_u l_C^2 + m_u^2 l_C^2 sin^2(-\theta_0)}} \quad 10^{-6} \right]$$

As x_{t1c} is a non-step reference, x_e is generated by a signal generator as follows

$$\dot{x}_e = A_e x_e + B_e x_{t1c}, \quad A_e = A_1 + B_1 F_1$$

$$B_e = B_1 G_1 \tag{11}$$

For equation (6) the nonlinear feedback part as follows:

$$v_{2N} = \rho_2(y_{t2}) N_2 \begin{pmatrix} x_{t3} \\ x_{t4} \end{pmatrix} \tag{12}$$

where $x_{e2} = 0$ as the reference $r = 0$.

$$N_2 = \begin{bmatrix} 0 & \frac{\zeta_2}{\pi f_2} \end{bmatrix}, \qquad \rho_2 = -1.4877 \frac{e^{-2.544|y_{t2}|} - e^{-1}}{1 - e^{-1}}.$$

ρ_2 is designed to remove the overshoot and $N_2 = B_2' P_2$ in which $P_2 > 0$ is a solution of the Lyapunov equation given by

$$(A_2 + B_2 F_2)' P_2 + P_2 (A_2 + B_2 F_2) = -W_2, \; W_2 = diag \begin{bmatrix} 10^{-6} & \frac{2\zeta_2^2}{\pi^2 f_2^2} \end{bmatrix}.$$

The complete CNF control law for equation (5) as follows,

$$v = v_{1L} + v_{1N} = F_1 \begin{pmatrix} x_{t1} \\ x_{t2} \end{pmatrix} + G_1 x_{t1c} + \rho_1(y_{t1}) N_1 \left[\begin{pmatrix} x_{t1} \\ x_{t2} \end{pmatrix} \quad -x_e \right] \tag{13}$$

and for equation (6) as follows,

$$x_{t1c} = v_{2L} + v_{2N} = (F_2 + \rho_2(y_{t2}) N_2) \begin{pmatrix} x_{t3} \\ x_{t4} \end{pmatrix} \tag{14}$$

4.0 RESULTS AND DISCUSSION

The numerical experiment is performed using the CarSim. Figure 2 shows the Simulink block used to design the control algorithm of CCNF and its integration with CarSim. All selected parameters are imported from CarSim and feed into Simulink to for CCNF controller design. In order to do a control performance for this numerical experimental, the LQR controller is defining as:

$$u(t) = -Kx(t), \tag{15}$$

where K is state feedback gain matrix.

The complete LQR equation as follows:

$$J_{LQR} = \frac{1}{2} \int_0^\infty (x^T Q x + u^T R u) dt, \tag{16}$$

The matrix gain K is represented

$$K = R^{-1}B'P, \tag{17}$$

The matrix P must satisfy the reduced-matrix Riccati equation.

$$A'P + PA - PBR'B'P + Q = 0 \tag{18}$$

Then the feedback regulator

$$U = -(R^{-1}B'P)x \tag{19}$$

In this research study, vertical acceleration is considered as a ride quality index. The criterion recommended by the International Standard Organization (ISO2631, 1985, 1989) provides the frequency weighting curve relating to vertical acceleration. The vertical weighting function is given in ISO2631. The weighted RMS vertical acceleration can be computed from the equation,

$$a_{RMS} = \left[\frac{1}{T}\int_0^T a_w^2(t)dt\right]^{\frac{1}{2}} \tag{4.2}$$

where $a_w(t)$ is the weighted vertical acceleration of a driver's seat. The vertical weighting function is given in ISO2631. Table 1 shows the relationship between the vertical acceleration level and the degree of comfort. Table 2 is shown as a parameter used in the simulation for CNF controller and passive system.

Table 1 RMS vertical acceleration level and degree of comfort (ISO2631-1, 1997)

RMS vertical acceleration level	Degree of comfort
(1) Less than 0.315 m/s^2	Not uncomfortable
(2) 0.315-0.63 m/s^2	A little uncomfortable
(3) 0.5-1 m/s^2	Fairly uncomfortable
(4) 0.8-1.6 m/s^2	Uncomfortable
(5) 1.25-2.5 m/s^2	Very uncomfortable
(6) Greater than 2 m/s^2	Extremely uncomfortable

Table 2 Physical parameters for quarter car model

Parameter	Value
Mass of a car body	455 kg
Mass of a car wheel	71 kg
Stiffness of a car body spring	17,659 N/m
Stiffness of a car tire	183,888 N/m
Damper coefficient	1950 Ns/m
Distance from point 0 to A	0.37 m
Distance from point 0 to B	0.64 m
Distance from point 0 to C	0.66 m

Before running the numerical experimental and numerical simualtion in both environments in MATLAB/Simulink and CarSim, the initial setting must be done first in CarSim. Figure 3 shows the independent suspension kinematic screen for active suspension system setting. The physical parameters involved as shown in Table 2 is used. The kinematics of the suspension linkages is described by the lateral and longitudinal motions of the wheel as the suspension deflects vertically. The lateral movement primarily affects the transfer of tire lateral force of the body and the resulting body roll. The longitudinal movement primarily affects the transfer of tire longitudinal force of the body and the resulting body pitch. The motion of the wheels is defined by the positions of the wheel centre, and by camber and caster changes as each wheel moves in jounce.

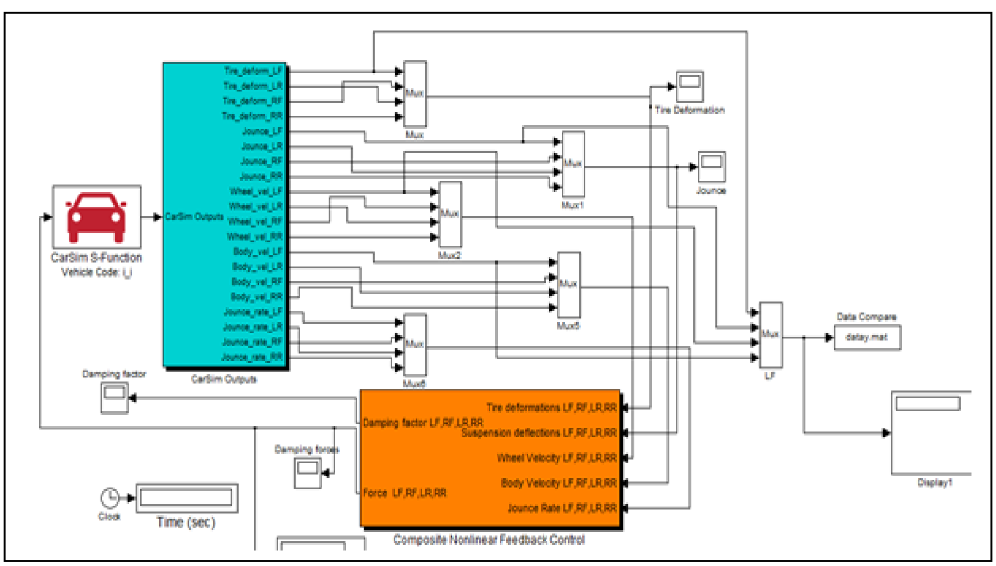

Figure 2 MATLAB/Simulink block diagram integrated with CarSim in CCNF

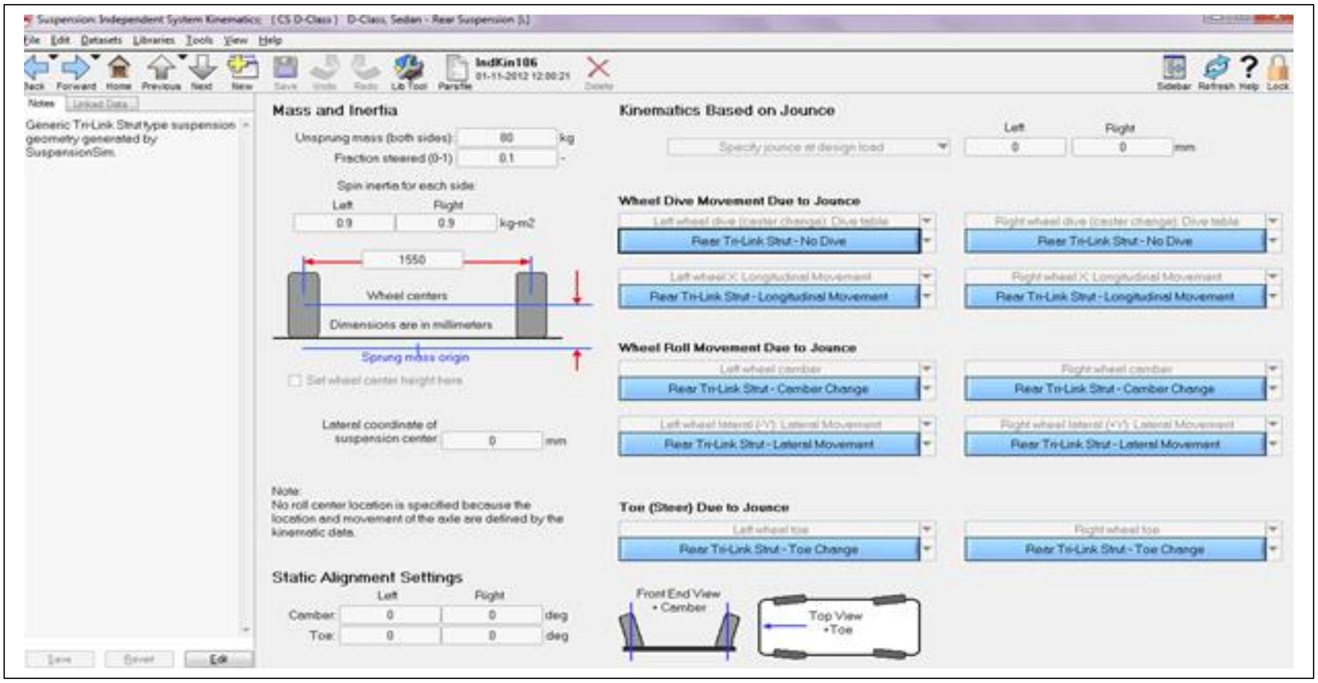

Figure 3 Control screen for active suspension parameter setup

The road profiles design also important factor to get the ride comfort quality and road handling performance in a nonlinear active suspension system. The geometry and friction setting is the important parameters that give effect when the ride quality test for active suspension system is running. The road profiles defined as a disturbance element in the CCNF controller design scheme. Figure 4 shows the control screen on the road profiles design parameter setting

The numerical experimental results can be shown in Figure 5 until Figure 9. The control performance of the nonlinear active suspension system can be observed by sprung mass and angular of control arm. In Figure 5, the displacement of sprung mass can be observed based on the CCNF controller, LQR controller and passive system. The transient response of CCNF is the best control performance in order to reduce overshoot and fast response due to the nonlinearity of the active suspension system. In Figure 6, the CCNF again gave a good transient response compared to the LQR controller and passive system. As mentioned by C.Sandu *et al.* [14] the greater angle of angular displacement of control arm change, the greater the nonlinearity. Therefore, the CCNF is succeeded to reduce the overshoot simultaneously the nonlinearity of angular displacement of control arm also less.

In Figure 7, the velocity of sprung mass controlled by CCNF is the best transient performance. Due to the influence of a road profile as in Figure 4, LQR controller and passive system is unsuccesful to overcome the high overshoot. The same situation happened in angular velocity of control arm as a result shown in Figure 8. The CCNF controller gave the best control performance due to the high damping ratio to reduce overshoot in the response. The acceleration of sprung mass is refered to the ISO 2631-1 in Table 1 as a guidline due to the human protection from the vibration noise. The results in Figure 9 shown that the acceleration of sprung mass controlled by CCNF controller gave the RMS vertical accelaration level 0.2856 m/s^2. This reading put the degree of comfort in comfortable level compared to the LQR controller and passive system. The angular accelaration of control arm represents the road handling performance of the car an active suspension system. The result in Figure 10 shows that the best transient response came from CCNF controller compared to the LQR controller and passive system.

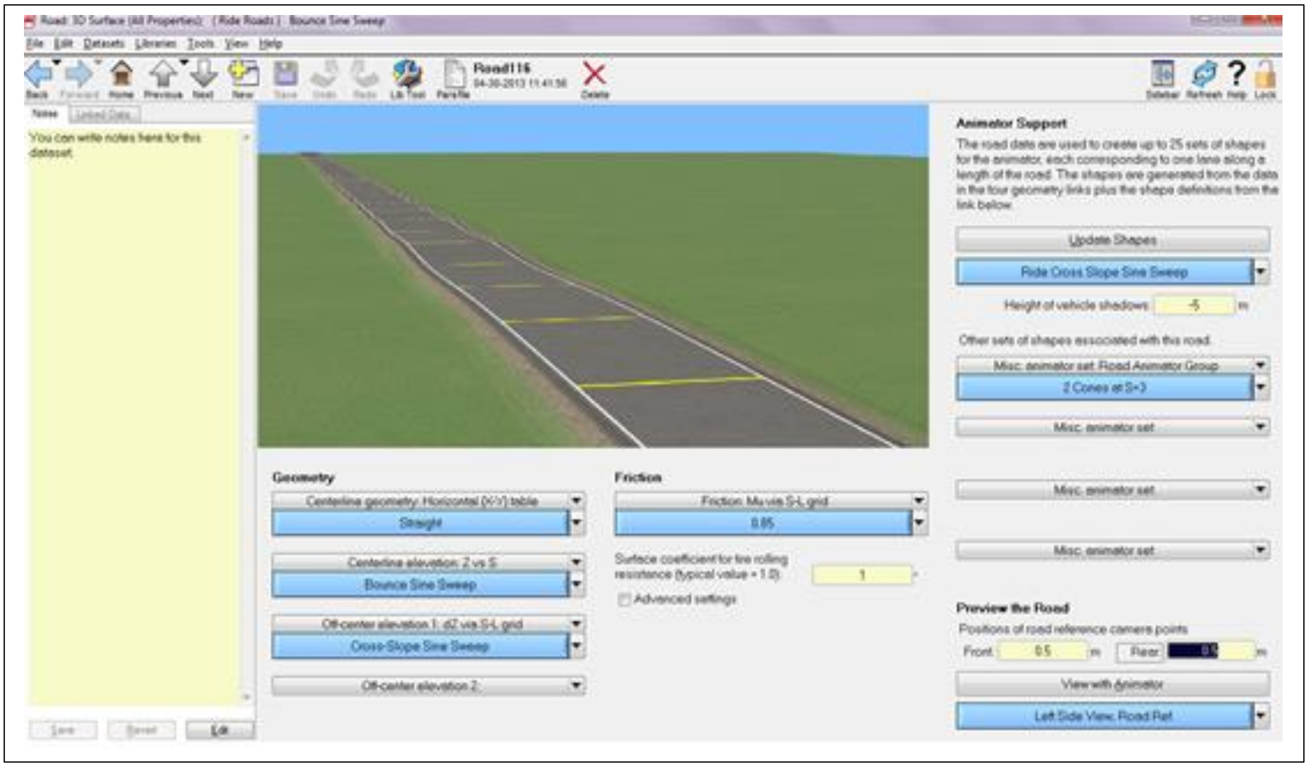

Figure 4 Control screen for road profile design

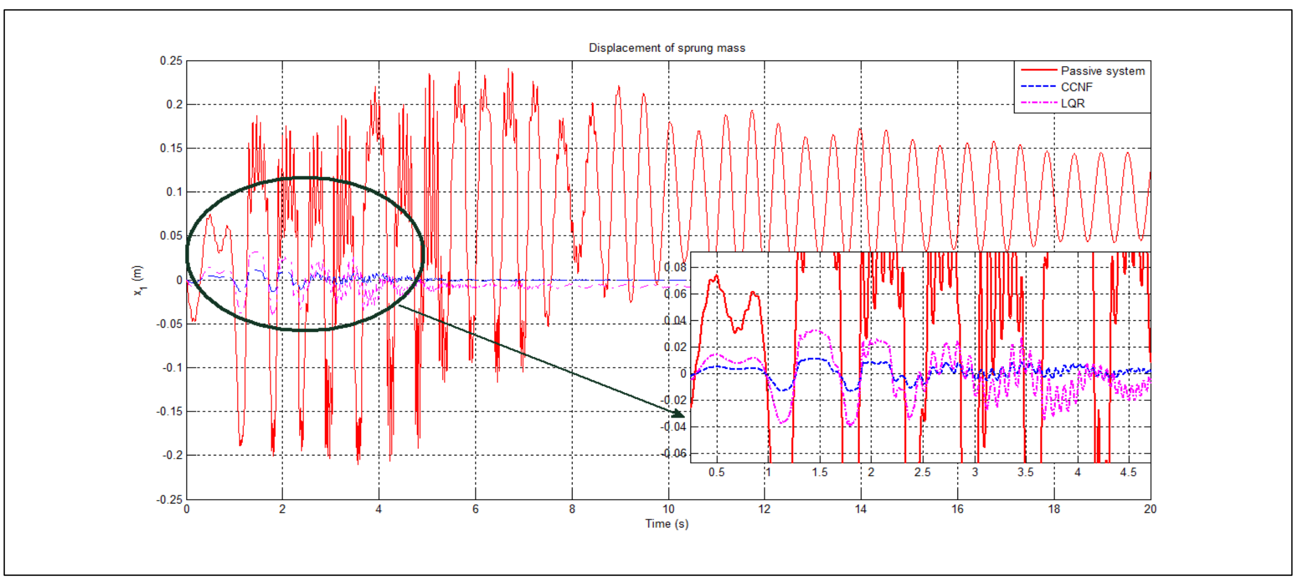

Figure 5 Displacement of sprung mass

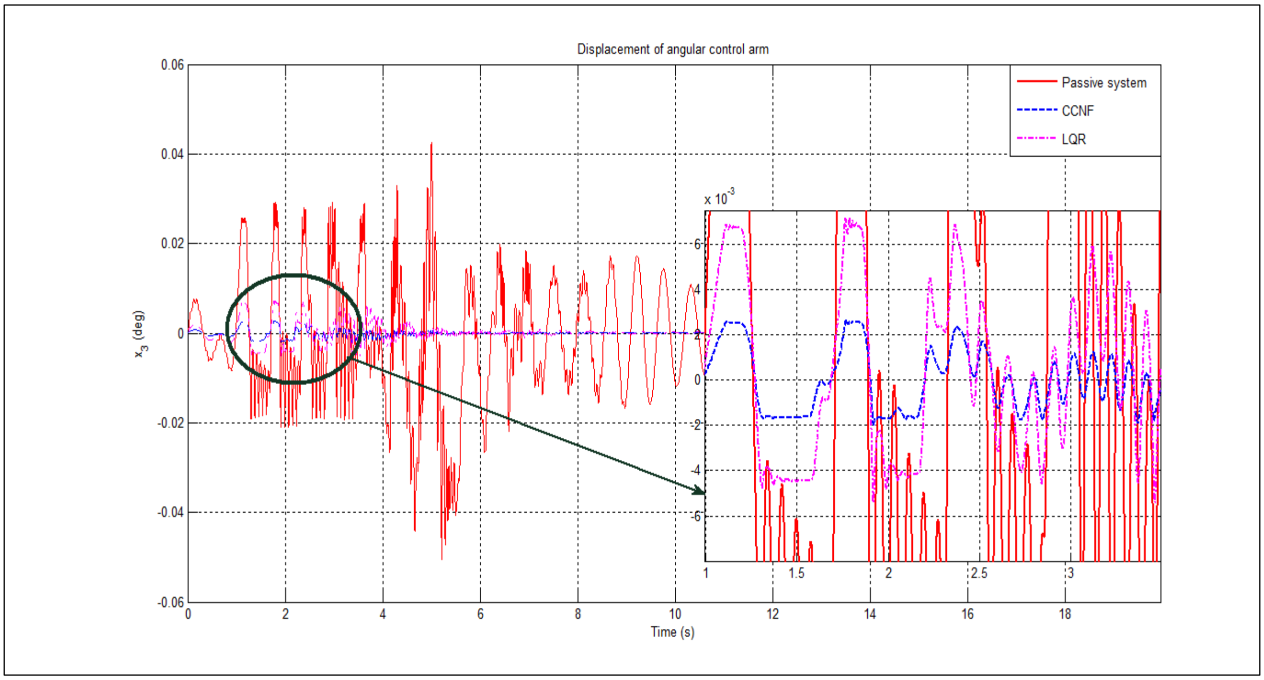

Figure 6 Angular displcament of control arm

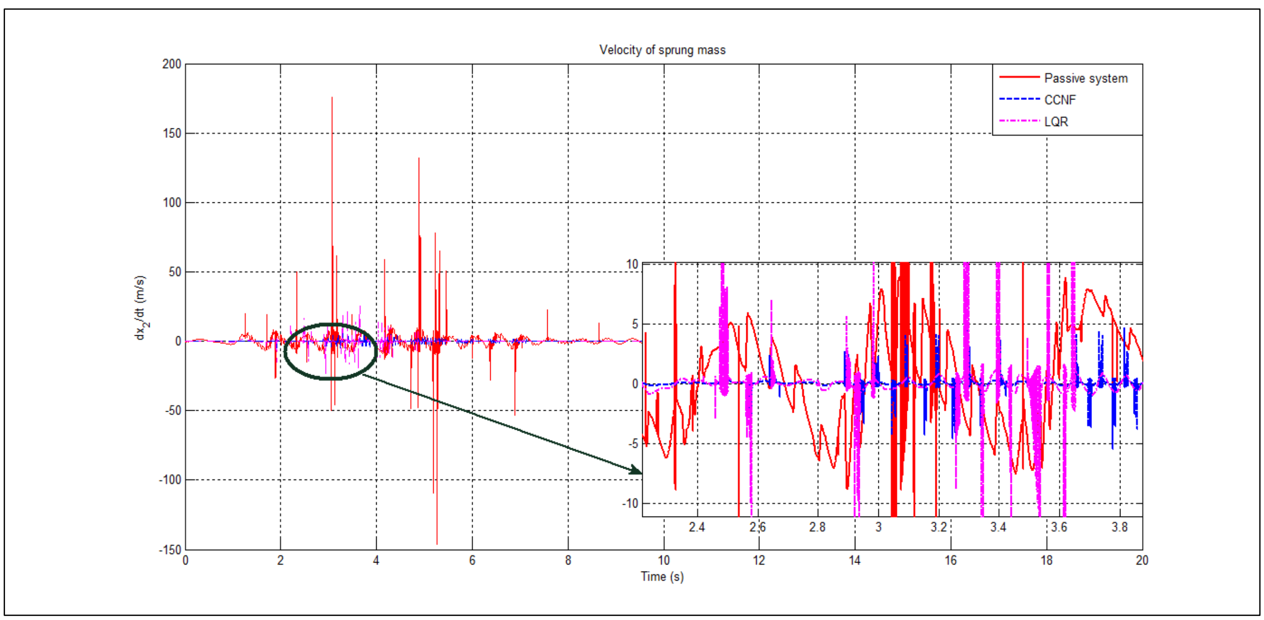

Figure 7 Velocity of sprung mass

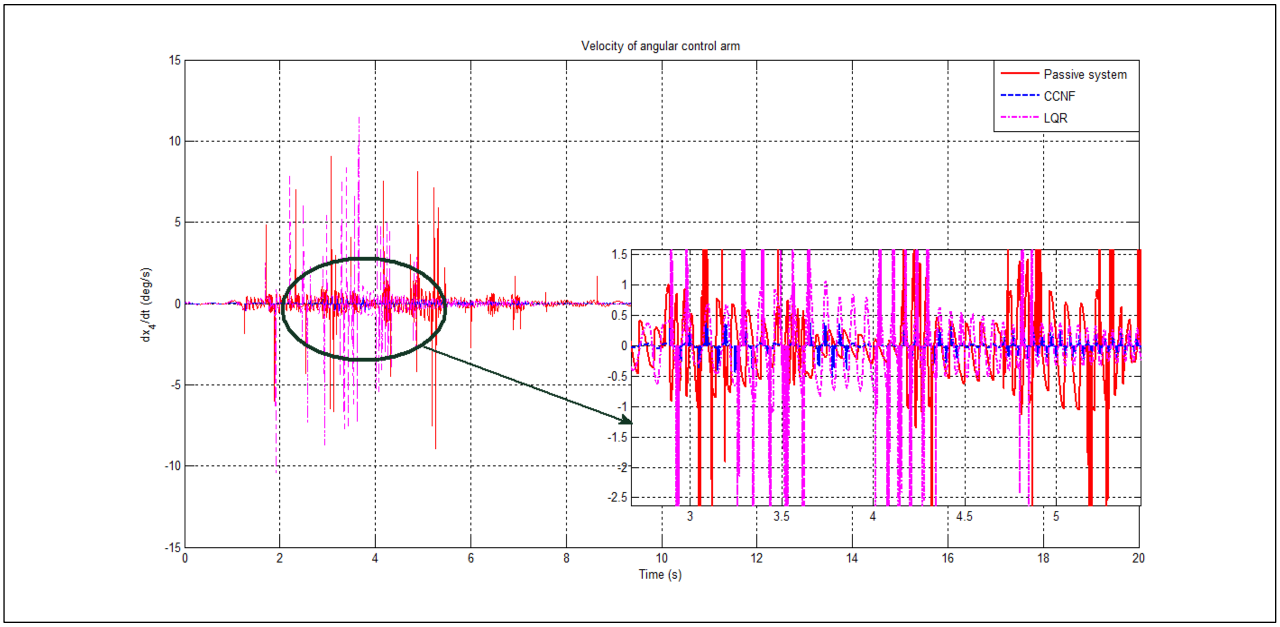

Figure 8 Angular velocity of control arm

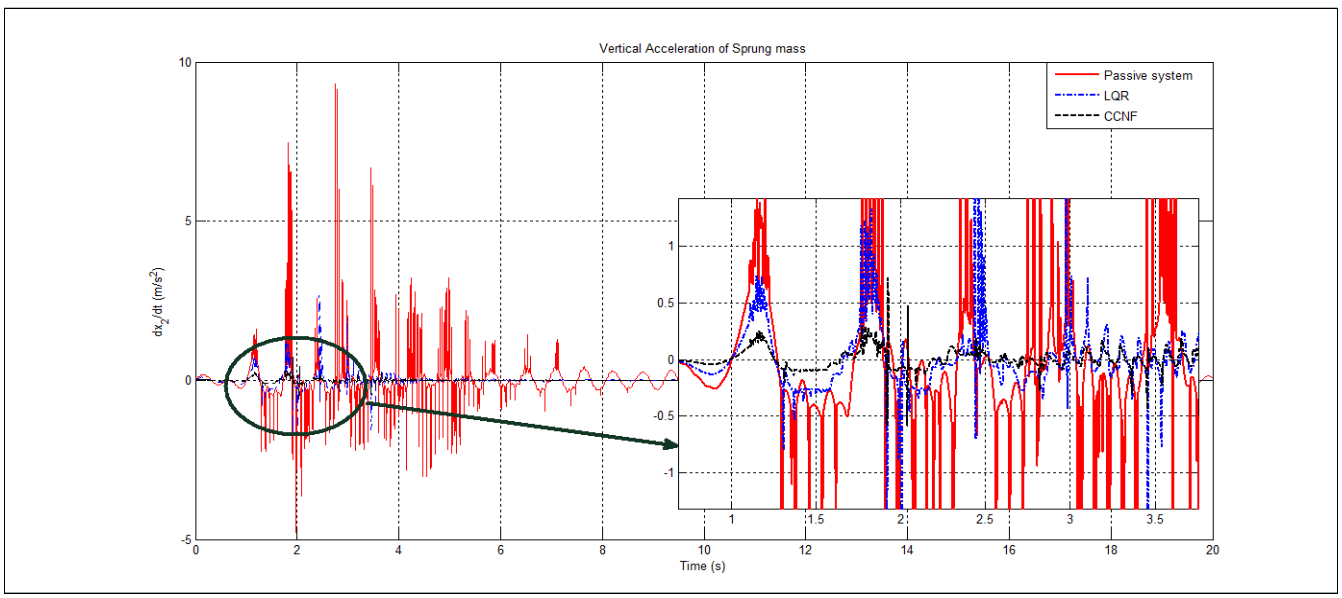

Figure 9 Vertical accleration of sprung mass

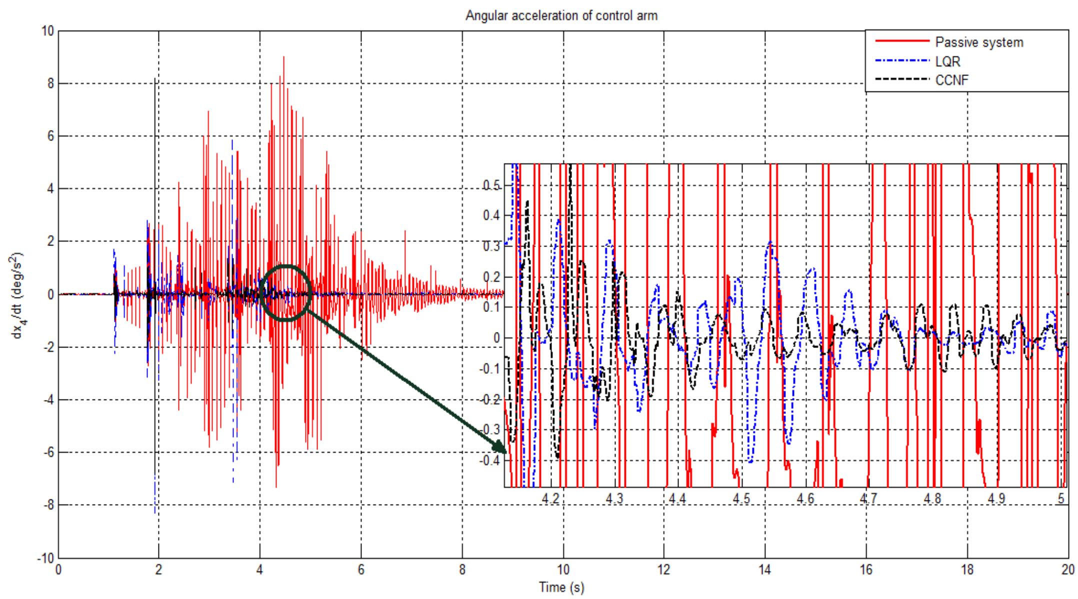

Figure 10 Angular acceleration of control arm

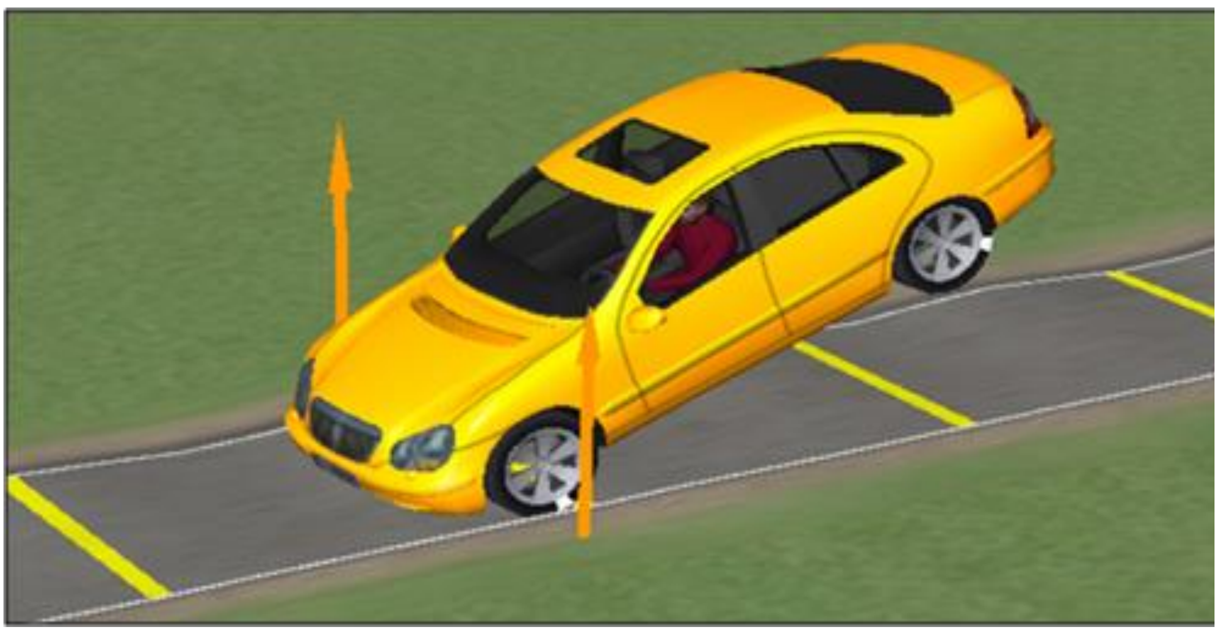

Figure 11 Virtual animation by CarSim for passive suspension system

Figure 12 Virtual animation by CarSim for active suspension system with CCNF

The effect of the control simulation can be observed in Figure 7 and Figure 8. The car in Figure 7 is considered to fail in terms of ride quality test and road handling performance. The car becomes unstable because of the effect on a passive suspension system without active force element. In Figure 8, the CCNF controller and LQR controller are successfully applied to the active suspension system. The different of control performance for both controllers can be evaluated from the results given from Figure 5 to Figure 10. The ride quality test and road handling performance made the car become stable at sine sweep road profile as validation process using CarSim.

■5.0 CONCLUSION

A CCNF controller designed is successfully applied to a nonlinear active suspension that resulted fast response and reduced to overshoot. The results for each variable proved that control strategy of CCNF gives the best control performance compared to the LQR controller and passive system. The CarSim can be used as a validation platform to nonlinear active suspension system and prove the control performance in a virtual animation environment. The CarSim also provided the best research tool for advance an automotive control system research field.

Acknowledgement

The authors would like to thank Universiti Teknologi Malaysia (UTM), Universiti Kuala Lumpur (UniKL) and Ministry of Higher Education (MOHE) Malaysia for their support.

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