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Constructing Control System for Active Suspension System

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Abstract

This paper presents a control system for active Suspension systems which have been widely applied to vehicles, right from the horse- drawn carriages with flexible leaf springs fixed at the four corners, to the modern automobiles with complex control algorithms. This implementation includes constructing the transfer function of the control system then making a simulation using MATLAB software. It is found that active suspension system improves ride comfort even at resonant frequency. For step input the of 0.08 m, the sprung mass displacement has been reduced by 25 % which shows the improvement in ride comfort and sprung mass acceleration reduced by 89.93% .The suspension travel has been reduced by 74.64% and tire deflection has reduced by 89.73% .For the real application in vehicles, the proposed active suspension structure faces inevitably some challenges including the cost, the required space in vehicle and power consumption .

Keywords: control systems, active suspension system, MATLAB simulation

1 INTRODUCTION

Every vehicle moving on the randomly profiled road is exposed to vibrations which are harmful both for the passengers in terms of comfort and for the durability of the vehicle itself. Therefore the main task of a vehicle suspension is to ensure ride comfort and road holding for a variety of road conditions and vehicle maneuvers. This in turn would directly contribute to the safety. In general, a good suspension should provide a comfortable ride and good handling within a reasonable range of deflection. Moreover, these criteria subjectively depend on the purpose of the vehicle. Therefore, in a good suspension design it is important is given to fairly reduce the disturbance to the outputs (e.g. vehicle height etc). A suspension system with proper cushioning needs to be “soft” against road disturbances and “hard” against load disturbances. A heavily damped suspension will yield good vehicle handling, but also transfers much of the road input to the vehicle body. Therefore, a suspension design is an art of compromise between these two goals. A good design of a passive suspension can work up to some extent with respect to optimized riding comfort and road holding ability, but cannot eliminate this compromise. There were many articles and papers discussed this issue, Leegwater,M. (2007), an active suspension is investigated which is capable of leveling the car during cornering theoretically without consuming energy. Simulations using a full-car model show that this maximizes the car’s cornering velocity. As extreme cornering may be required to remain on the road or to avoid an obstacle, implementing the active suspension system improves safety. As the active part of the suspension takes care of realizing good cornering behavior and of static load variations, the primary suspension springs can be tuned purely for optimizing comfort and road holding. Simulations show that the required energy for leveling the car during cornering is negligible, it is concluded that the active suspension system is able to economically level the car, [8]. Yoshimura T. et al.(2005), presented the construction of an active suspension control of a one-wheel car model using fuzzy reasoning and a disturbance observer. The one-wheel car model to be treated here can be approximately described as a nonlinear two degrees of freedom system subject to excitation from a road profile. The active control is designed as the fuzzy control inferred by using single input rule modules fuzzy reasoning, and the active control force is released by actuating a pneumatic actuator. The excitation from the road profile is estimated by using a disturbance observer, and the estimate is denoted as one of the variables in the precondition part of the fuzzy control rules. A compensator is inserted to counter the performance degradation due to the delay of the pneumatic actuator. The experimental result indicates that the proposed active suspension system improves much the vibration suppression of the car model, [14]. Mouleeswaran S. et al.(2008), The present work aimed at developing an active suspension for the quarter car model of a passenger car to improve its performance by using a proportional integral derivative (PID) controller. The controller design dealt with the selection of proportional, derivative gain and integral gain parameters (Kp, Ki, and Kd). The results show that the active suspension system has reduced the peak overshoot of sprung mass displacement, sprung mass acceleration, suspension travel and tire deflection compared to passive suspension system[12]. The presented work aims to construct control system simulated by using MATLAB SW.

2 ACTIVE SUSPENSION SYSTEMS

Some of the advanced adaptive suspension systems may be called active suspensions. This concept refers to those controlled by double-acting hydraulic cylinders or solenoids (usually called actuators) that are mounted at each wheel. Each actuator maintains a sort of hydraulic equilibrium with the others to carry the vehicle's weight, while maintaining the desired body attitude. At the same time, each hydraulic actuator serves as its own shock absorber, eliminating the need for yet another traditional suspension component. In other words, each hydraulic actuator acts as both a spring (with variable-rate damping characteristics) and a variable –rate shock absorber. This is accomplished in an active suspension system by varying the hydraulic pressure within each cylinder and the rate at which it increases or decreases. By bleeding or adding hydraulic pressure from the individual actuators, each wheel can react independently to changing road conditions shown in Fig (1). [2].

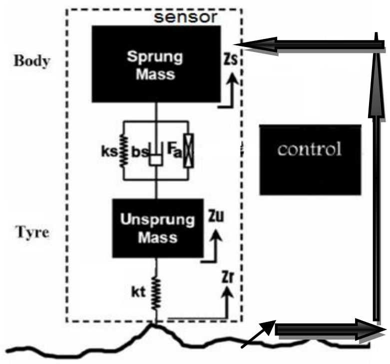

Fig.1 Active suspension system

3 CONTROL SYSTEMS

In this work, a quarter car models are considered instead of full model. It is not only leads to simplify the analysis but also represents most of the features of the full model. A quarter models shown in Figure (2) which is the most commonly used model in the design studies for passive suspension systems. According to Newton's second law and free body diagram approach, the equations of motion for the system are written as,[7,9,10]
-For sprung mass Ms

$$Ms * \ddot{Z}s + Ca * (\dot{Z}_s - \dot{Z}_{us}) + Ks * (Z_s - Z_{us}) = 0 \tag{1}$$

-For unsprung mass Mus

$$Mus * \ddot{Z}us + Ca * (\dot{Z}us - \dot{Zs}) + Ks * (Zus - Zs) + Kt * (Zus - Zr) = 0 \qquad (2)$$

Active suspension systems employ hydraulic or pneumatic actuators, which provide the desired force in the suspension system. Hence, clearly external force is introduced into the system by the means of hydraulic actuator. For active suspension shown in Figure (2), using the Newton's second law of motion and free-body diagram concept, the following equations of motion are derived.

-For sprung mass Ms

$$Ms * \ddot{Z}s + Ca * (\dot{Z}_s - \dot{Z}us) + Ks * (Zs - Zus) = F(t) \qquad (3)$$

-For unsprung mass Mus

$$Mus * \ddot{Z}us + Ca * (\dot{Z}us - \dot{Zs}) + Ks * (Zus - Zs) + Kt * (Zus - Zr) = -F(t) \qquad (4)$$

Taking Laplace transform for the equations of active suspension

$$Zs(S) * [Ms * S^2 + Ca * S + Ks] - Zus(S) * [Ca * S + Ks] = F(S)$$

$$Zs(S) * [-Ca * S - KS] + Zus * [Mus * S^2 + Ca * S + Ks + Kt]$$
$$= Kt * Zr(S) - F(S) \qquad (5)$$

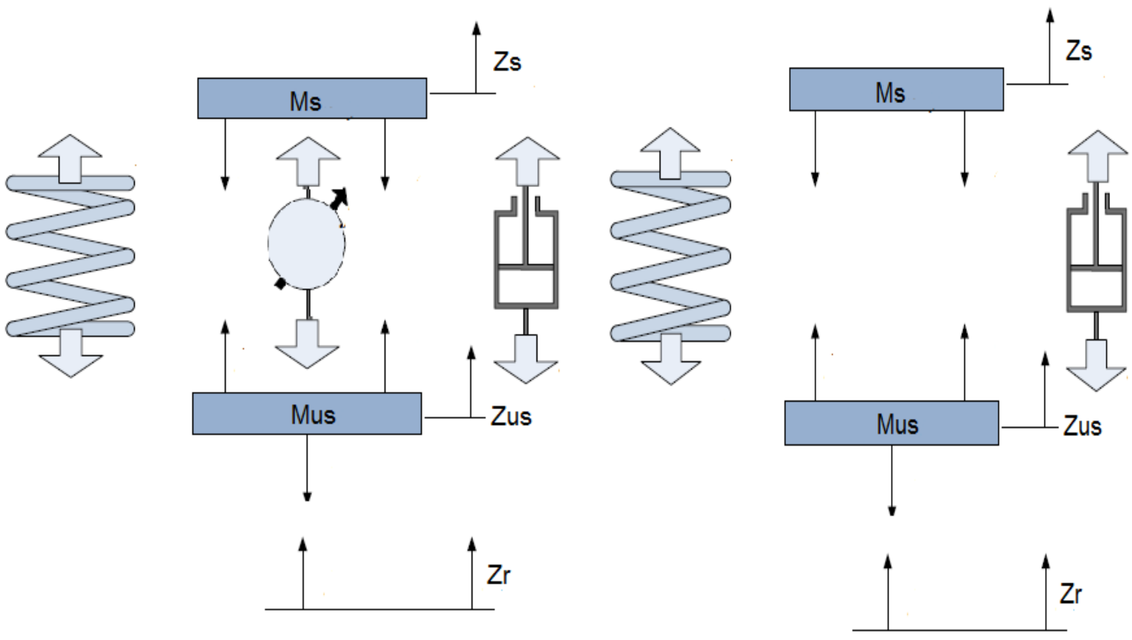

Fig. (2) Active suspension and passive suspension system

where: Ca: damping coefficient = 1000 N s/m, ks : spring stiffness =18600 N/m., kt : tire stiffness = 196000 N/m., Ms: quarter car sprung mass =250 kg ., Mus: unsprung

mass = 50 kg ., F : actuator force (N) , Zr : road profile, m, Zs : sprung mass vertical displacement, m, Zus : unsprung mass vertical displacement, m .
In order to find the transfer functions we have to find the following terms

$\frac{Zs(S)}{F(S)}$, $\frac{Zus(S)}{F(S)}$, $\frac{Zs-Zus}{F(S)}$, $\frac{Zs(S)}{Zr(S)}$, $\frac{Zus(S)}{Zr(S)}$, and $\frac{Zs-Zus}{Zr(S)}$

Using the super position principle we set $(Zr = 0)$ first and find the first (3) terms (1, 2, and 3). Then we set $(F = 0)$ and find the rest terms (4, 5, and 6). The First matrix to find first three terms:-

$$\begin{bmatrix} (Ms * s^2 + Ca * s + Ks) & -(Ca * s + Ks) \\ -(Ca * s + Ks) & (Mus * s^2 + Ca * s + Ks + Kt) \end{bmatrix} \begin{bmatrix} Zs(s) \\ Zus(s) \end{bmatrix} = \begin{bmatrix} F(s) \\ -F(s) \end{bmatrix} \quad (6)$$

this yields to

$$(Ms * s^2 + Ca * s + Ks)\times(Mus * s^2 + Ca * s + Ks + Kt) - (Ca * s + Ks)^2 \quad (7)$$

$$Zs(s) = \frac{F(s)\times \left(Mus*s^2+Ca*s+Ks+Kt\right)-F(s)\times(Ca*s+Ks)}{(Ms*s^2+Ca*s+Ks)\times(Mus*s^2+Ca*s+Ks+Kt)-(Ca*s+Ks)^2} \quad (8)$$

$$\frac{Zs(s)}{F(s)} = \frac{(Mus*s^2+Kt)}{(Ms*s^2+Ca*s+Ks)\times(Mus*s^2+Ca*s+Ks+Kt)-(Ca*s+Ks)^2} \quad (9)$$

$$Zus(s) = \frac{-F(s)*\left(Ms*s^2+Ca*s+Ks\right)+F(s)*(Ca*s+Ks)}{(Ms*s^2+Ca*s+Ks)\times(Mus*s^2+Ca*s+Ks+Kt)-(Ca*s+Ks)^2} \quad (10)$$

$$\frac{Zus(s)}{F(s)} = \frac{-Ms*s^2}{(Ms*s^2+Ca*s+Ks)\times(Mus*s^2+Ca*s+Ks+Kt)-(Ca*s+Ks)^2} \quad (11)$$

$Zs(S) - Zus(S) = car\ body\ displacement$

$$\frac{Zs(s)-Zus(s)}{F(S)} = \frac{Mus*s^2+Kt+Ms*s^2}{(Ms*s^2+Ca*s+Ks)\times(Mus*s^2+Ca*s+Ks+Kt)-(Ca*s+Ks)^2} \quad (12)$$

From the Second matrix we find the rest:-

$$\begin{bmatrix} (M_s s^2 + C_a S + k_s) & -(C_a S + k_s) \\ -(C_a S + k_s) & (M_{us} s^2 + C_a S + k_s + K_t) \end{bmatrix} \begin{bmatrix} z_s(s) \\ z_{us}(s) \end{bmatrix} = \begin{bmatrix} 0 \\ k_t z_r(s) \end{bmatrix} \quad (13)$$

$$Zs(s) = \frac{0+k_t z_r(C_a S+k_s)}{det} \quad (14)$$

$$\frac{z_s(s)}{z_r(s)} = \frac{k_t (C_a S+k_s)}{det} \quad (15)$$

$$Zus(s) = \frac{k_t z_r(M_s s^2+C_a S+k_s)}{det} \quad (16)$$

$$\frac{z_{us}(s)}{z_r(s)} = \frac{k_t (M_s s^2+C_a S+k_s)}{det} \quad (17)$$

$$\frac{z_s(s)-z_{us}(s)}{z_r(s)} = \frac{-k_t\, M_s s^2}{det} \tag{18}$$

- Finding car body displacement

$$\frac{F(s)}{z_r(s)} = \frac{F(s)}{z_s - z_{us}} \times \frac{z_s - z_{us}}{z_r(s)} \tag{19}$$

$$\frac{F(s)}{z_r(s)} = \frac{det}{s^2(M_s+M_{us})+k_t} \times \frac{-k_t\, M_s s^2}{det} \tag{20}$$

$$\frac{F(s)}{z_r(s)} = \frac{-k_t\, M_s s^2}{s^2(M_s+M_{us})+k_t} \tag{21}$$

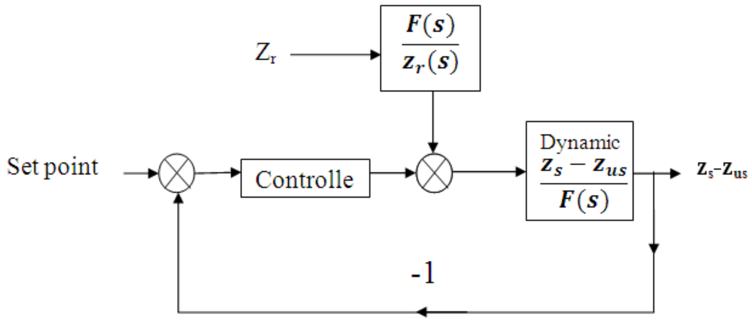

Fig. 3 car body displacement

Figure (3) shows the control loop for the active suspension system .where Z_r represents the road input which is considered as disturbance signal .This loop is used to find the car body displacement.

- Finding the sprung mass displacement

$$\frac{F(s)}{z_r(s)} = \frac{F(s)}{z_s(s)} \times \frac{z_s(s)}{z_r(s)} \tag{22}$$

$$\frac{F(s)}{z_r(s)} = \frac{det}{M_{us}s^2+k_t} \times \frac{k_t(C_a S+k_s)}{det} \tag{23}$$

$$\frac{F(s)}{z_r(s)} = \frac{k_t(C_a S+k_s)}{M_{us}s^2+k_t} \tag{24}$$

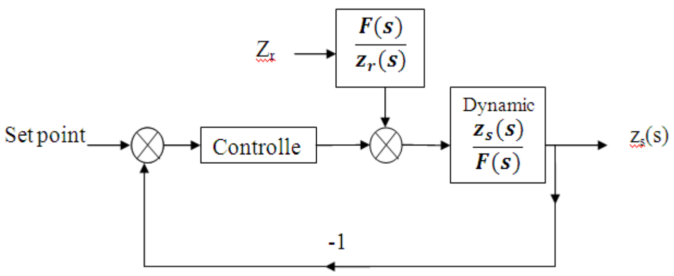

Fig 4 sprung mass displacement

Figure (4) shows the control loop for the active suspension system .where Z_r represents the road input which is considered as disturbance signal .This loop is used to find the sprung mass displacement (Zs). But the control loop must be more obvious and more practical to implement. So the control block must be divided into two parts which are the servo valve and the **controller**. The servo valve system also contains two parts which are the **spool** and the **hydraulic actuator** . The rest of this project will show how we obtained the transfer function for each component and some theoretical explanation about each one [4],[5].

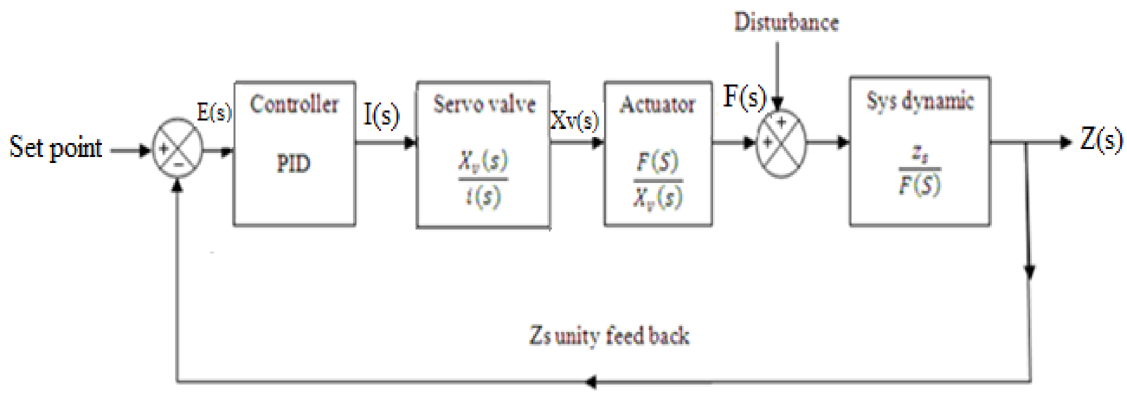

Fig 5 sprung mass displacement

Figure (5) shows the actual control loop for the active suspension system. The PID controller controls the servo valve according to the output from the displacement sensor. The controller varies the output current which is sent to the windings of the servo valve. This current shifts the spool according to its direction, this will create a force proportional to the displacement this force will oppose the unsprung mass displacement so it will damp the oscillations quickly and increase the ride comfort.

4 RESULTS AND DISCUSSION

The step input characterizes a vehicle coming out of a pot hole. The pot hole has been represented in the following manner for the step height of 0.08m.

$$W = \begin{cases} 0, & t < 1 \\ .08, & t \geq 1 \end{cases}$$

Figure (6) shows the peak overshoot of sprung mass displacement for passive system is .127 m where as for the active suspension system it is .09 m. Reduction in peak value = (passive value – active value) / passive value = 29.1% .

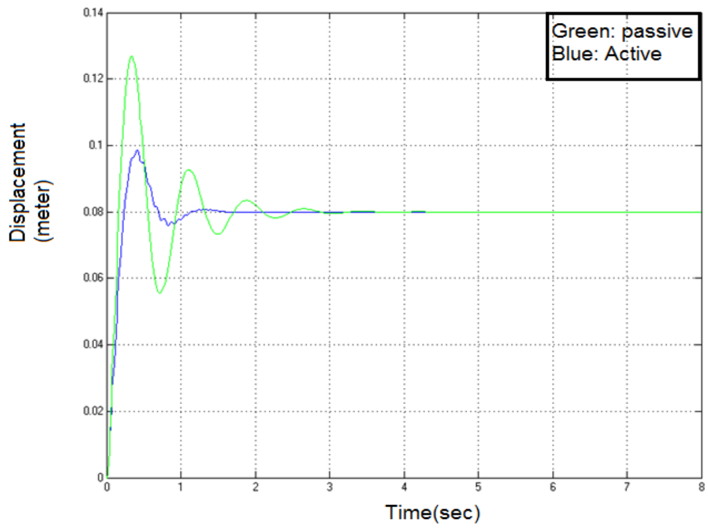

Fig.6 Sprung Mass displacement for active and passive

Figure (7) shows the peak overshoot of sprung mass acceleration for passive system is17.99 m/s^2 where as for the active suspension system it is 1.81 m/s^2.
Reduction in peak value = 89.9% ,which indicates that the active suspension has more riding comfort than active suspension system. We can also notice that the oscillation of active suspension system is less than passive system and settling time is also less.

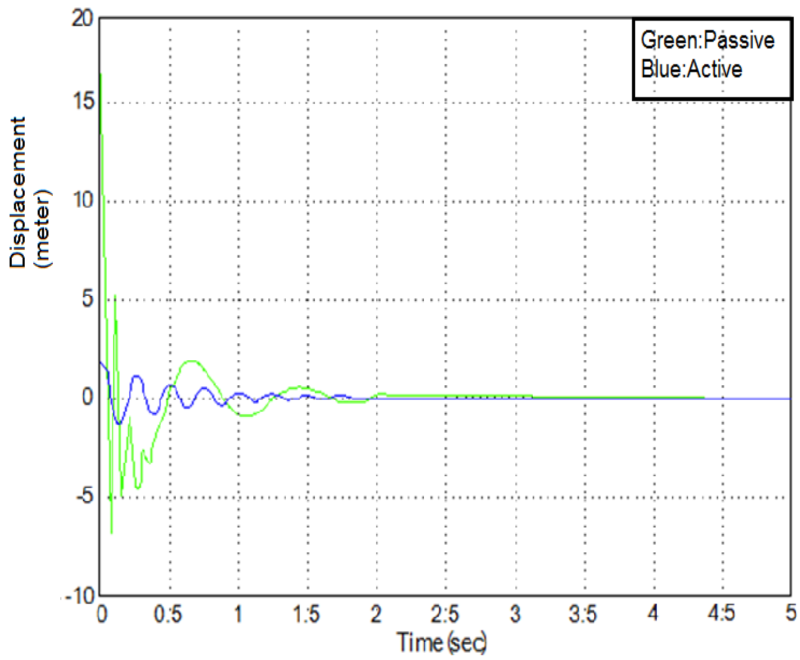

Fig.7 Sprung Mass acceleration for active and passive

Figure (8) shows the maximum tire deflection for passive system is 0.03246m where as for the active suspension system it is 0.0033 m. Reduction in peak value = 90% . This is an indication on consumption or the depreciation of the tires. We can conclude that the life of the tires on the active suspension system is more than passive system.

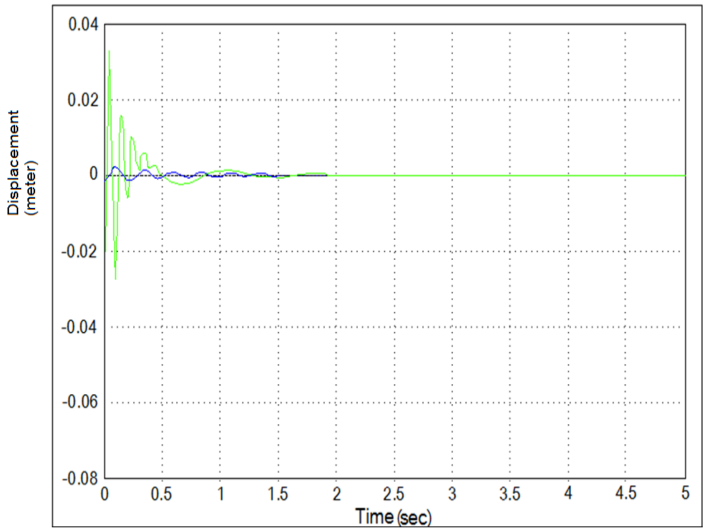

Fig.8 maximum tire deflection

Figure (9) describes the maximum elongation of suspension spring for passive system is 0.04362 m where as for the active suspension system it is 0.01106m. We can also notice that the oscillation of active suspension system is less than passive system and settling time is also less.

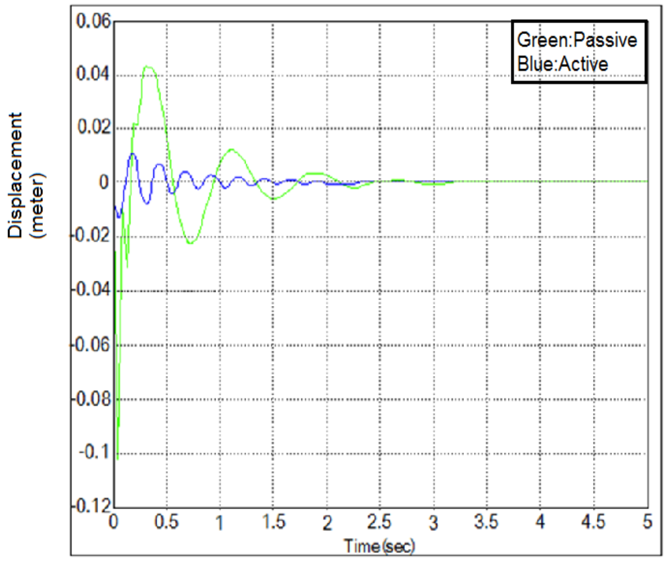

Fig.9Elongation for suspension spring for active and passive

2) flat road surface with a sinusoidal concave bump followed by a sinusoidal convex bump

(Fig 10) shows the road input which represents a concave down pump and a concave up pump. There is a flat road surface between of them. This road test is used to measure the response of active suspension system and compare it with the response of passive suspension system. The step input was used to explain a theoretical response of the system. This is a simulated model of a real road input.

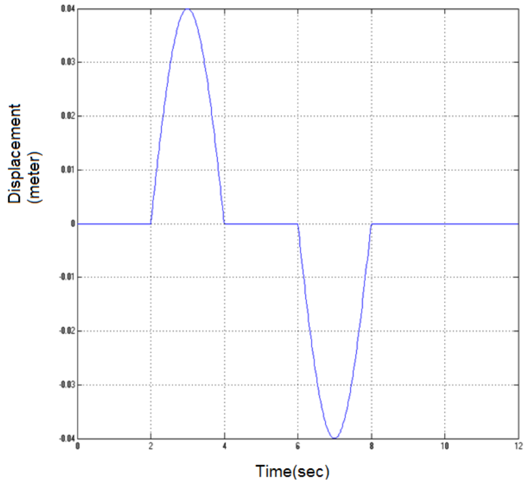

Fig.10 road input concave up and concave down

Figure (11) describes the peak overshoot of sprung mass displacement for passive system is .04 m where as for the active suspension system it is .017 m. Reduction in peak value = (passive value – active value) / passive value. If we want to find the reduction in the peak value for the sprung mass displacement, we find the sprung mass displacements for both active and passive suspension systems. Reduction in peak value = $\frac{.04-.017}{.04} = 57\%$. This is a direct indication on the superiority of the active suspension on the passive suspension system. The ride comfort also can be noticed from this graph.

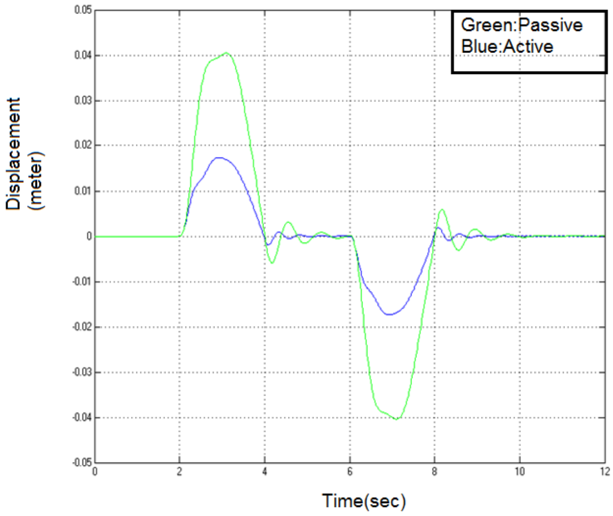

Fig.11 Sprung Mass displacement for active and passive

5 CONCLUSIONS

-PID controller including hydraulic dynamics has been designed for a quarter car model of a passenger car to improve the ride comfort and road holding ability.
-It is also found that active suspension system improves ride comfort even at resonant frequency.
-For step input the of 0.08 m, the sprung mass displacement has been reduced by 25 % which shows the improvement in ride comfort and sprung mass acceleration reduced by 89.93% .The suspension travel has been reduced by 74.64% and tire deflection has reduced by 89.73% .
-For the real application in vehicles, the proposed active suspension structure faces inevitably some challenges including the cost, the required space in vehicle and power consumption .
-For the power consumption problem. This power consumption is lowest when system is least active, as when driving on a smooth road. Rough roads and hard maneuvers, on other hand; put more of a demand on the system. The hydraulic pump works harder and thus requires more power.

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