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# Design and Development of PID Controller-Based Active Suspension System for Automobiles

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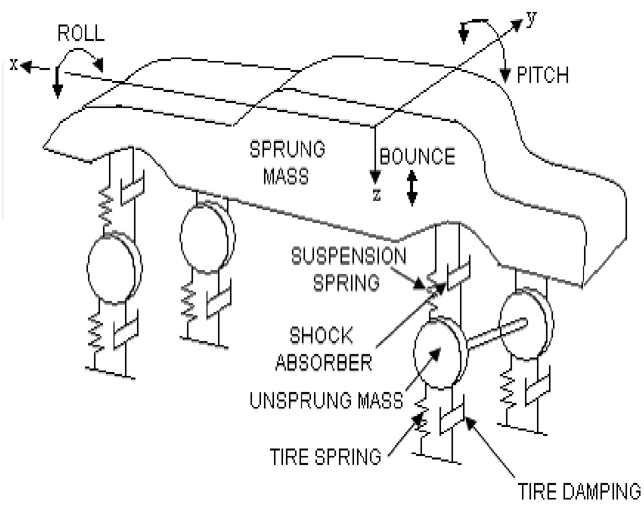
## 1. Introduction

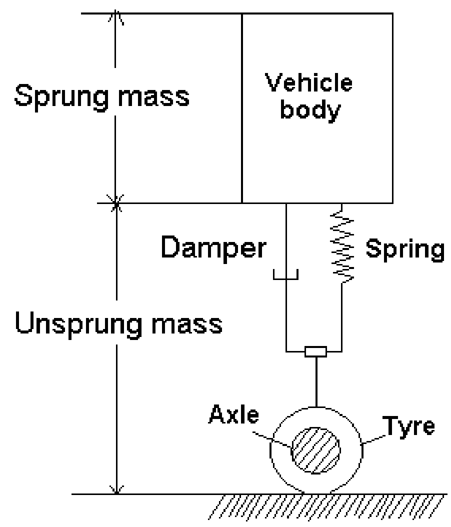
Suspension systems have been widely applied to vehicles, right from the horse-drawn carriage with flexible leaf springs fixed at the four corners, to the modern automobile with complex control algorithms. Every vehicle moving on the randomly profiled road is exposed to vibration which is harmful both for the passengers in terms of comfort and for the durability of the vehicle itself. Different disturbances occur when a vehicle leans over during cornering (rolling) and dives to the front during braking (pitching). Also, unpleasant vertical vibrations (bouncing) of the vehicle body can occur while driving over road irregularities. These dynamic motions do not only have an adverse effect on comfort but can also be unsafe, because the tyres might lose their grip on the road. 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 of the user.

A typical suspension system used in automobiles is illustrated in Figure 1. 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. Sports cars usually have stiff, hard suspension with poor ride quality while luxury sedans have softer suspensions but with poor road handling capabilities. From a system design point of view, there are two main categories of disturbances on a vehicle, namely road and load disturbances. Road disturbances have the characteristics of large magnitude in low frequency (such as hills) and small magnitude in high frequency (such as road roughness). Load disturbances include the variation of loads induced by accelerating, braking and cornering. Therefore, in a good suspension design, importance 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, whereas a lightly damped suspension will yield a more comfortable ride, but would significantly reduce the stability of the vehicle at turns, lane change maneuvers, or during negotiating an exit ramp. 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.




(a) Full car model




(b) Quarter car model

Fig. 1. Suspension system of a passenger car

The traditional engineering practice of designing a spring and a damper, are two separate functions that has been a compromise from its very inception in the early 1900's. Passive suspension design is a compromise between ride comfort and vehicle handling, as shown in Figure 2. In general, only a compromise between these two conflicting criteria can be obtained if the suspension is developed by using passive springs and dampers.. This also applies to modern wheel suspensions and therefore a break-through to build a safer and more comfortable car out of passive components is below expectation. The answer to this problem seems to be found only in the development of an active suspension system.

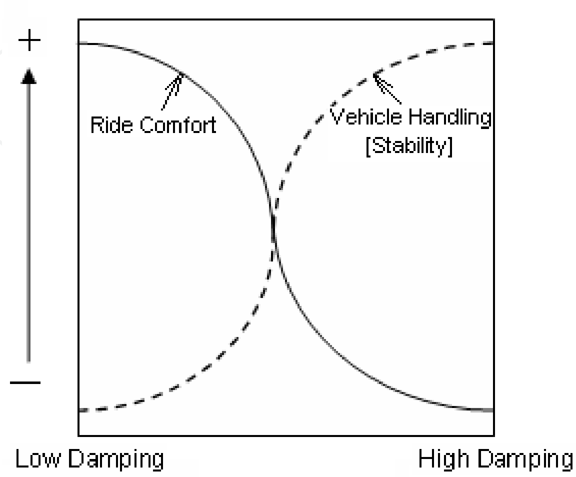



Fig. 2. Performance compromise of passive suspension system

In recent years, considerable interest has been generated in the use of active vehicle suspensions, which can overcome some of the limitations of the passive suspension systems. Demands for better ride comfort and controllability of road vehicles has motivated many automotive industries to consider the use of active suspensions. These electronically controlled active suspension systems can potentially improve the ride comfort as well as the road handling of the vehicle simultaneously. An active suspension system should be able to provide different behavioral characteristics depending upon various road conditions, and be able to do so without going beyond its travel limits.

Though the active suspension systems are superior in performance to passive suspension, their physical realization and implementation is generally complex and expensive, requiring sophisticated electronic operated sensors, actuators and controllers. Recent advances in adjustable dampers, springs, sensors and actuators have significantly contributed to the applicability of these systems. Consequently, the automobile has a better combination of ride and handling characteristics under various conditions, than cars with conventional suspension systems. Since electronic controlled suspension systems are more expensive than conventional suspension systems, they are typically found in luxury-class automobiles and high expensive sport utility vehicles. Therefore, a study has been made to develop an active suspension system for improved performance with less cost on light passenger vehicle.

Active suspension system is characterized by a built-in actuator, which can generate control forces to suppress the above-mentioned motions. In addition, the road holding has also been improved because of the dynamic behavior of the contact forces between the tyres and road.

Active vehicle suspensions have attracted a large number of researchers in the past few decades, and comprehensive surveys on related research are found in publications by (Elbeheiry et al, 1995), (Hedrick & Wormely 1975), (Sharp & Crolla 1987), (Karnopp 1995) and (Hrovat 1997). These review papers classify various suspension systems discussed in literature as passive, active (or fully active) and semi-active (SA) systems.

Some of potential benefits of active suspension were predicted decades ago by the first pioneer researchers. Indeed, the optimal control techniques that were launched with "Sputnik" and used in the aerospace industry since the 1950s and 1960s, have also been applied to the study of active suspensions, starting from about the same period by (Crossby & Karnopp 1973).

Fully active suspension system (FASS) is differentiated from semi-active suspension system(SASS) on the fact that it consists of a separate active force generator. The physical implementation of FASS is usually provided with a hydraulic actuator and power supply as shown in Figure 3. Fully active suspension systems have been designed by Wright and (Williams 1984) and (Purdy and Bulman 1993), which appear in formula one racing cars. Active suspension system has been compared with semi-active suspension system by (Karnopp 1992) and concluded that active suspensions have performance improvements, particularly in vehicle handling and control. In the process of enhancing passenger comfort and road handling, active suspensions introduce additional considerations of rattle space and power consumption, which must be factored into the overall design goals. While the ride/ handling tradeoff is prevalent in most approaches as pointed out by (Karnopp 1986), (Hrovat 1988) and (Elbeheiry 2000), the ride/ rattle space tradeoff is not explicitly addressed. Alternately, an active suspension system has been developed to improve ride comfort with rattle space limitations by (Jung-Shan Lin et al, 1995). But tire-road contact has not been studied.

However, not much literature is found on FASS as reviewed by (Pilbeam & Sharp 1996) and (Hrovat 1997). The advantage of FASS is that its bandwidth is more than that of SASS which is very much described by (Hrovat 1997). It has been shown that FASS requires considerable amount of energy to actuate and can be quite complex and bulky and therefore requires further stringent research. Recently, some research has been focused on the experimental development of the active suspension systems.The construction of an active suspension control of a one-wheel car model using fuzzy reasoning and a disturbance observer has been presented by (Yoshimura & Teramura 2005). (Senthilkumar & Vijayarangan 2007) presented the development of fully active suspension system for bumpy road input using PID controller. (Nemat & Modjtaba 2011) compared PID and fuzzy logic control of a quarter car suspension system. Non-linear active suspension systems have also been developed by (Altair & Wang 2010 & 2011). Different control strategies for developing active suspension systems have also been proposed by (Alexandru & Alexandru 2011, Lin & Lian 2011, Fatemeh Jamshidi & Afshin Shaabany 2011).

In active suspension systems, sensors are used to measure the acceleration of sprung mass and unsprung mass and the analog signals from the sensors are sent to a controller. The controller is designed to take necessary actions to improve the performance abilities already set. The controller amplifies the signals and the amplified signals are fed to the actuator to generate the required forces to form closed loop system (active suspension system), which is schematically depicted in Figure 3. The performance of this system is then compared with that of the open loop system (passive suspension system).

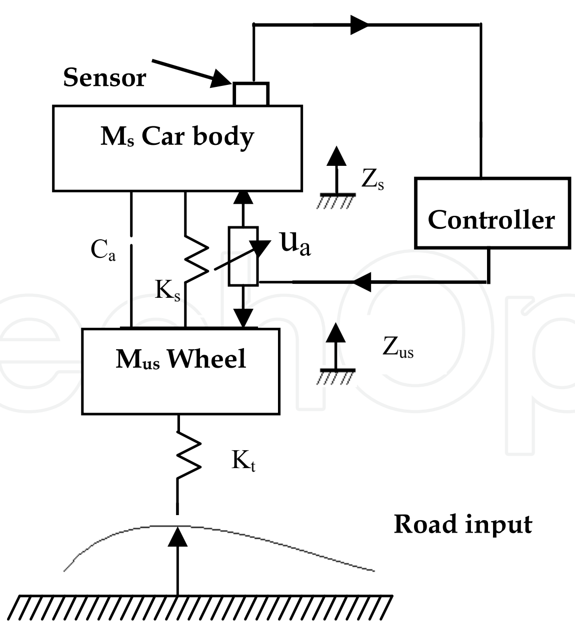



Fig. 3. Active suspension system

This chapter describes the development of a controller design for the active control of suspension system, which improves the inherent tradeoff among ride comfort, suspension travel and road-holding ability. The controller shifts its focus between the conflicting objectives of ride comfort, rattle space utilization and road-holding ability, softening the suspension when rattle space is small and stiffening it as it approaches the travel limits. The developed design allows the suspension system to behave differently in different operating conditions, without compromising on road-holding ability. The effectiveness of this control method has been explained by data from time domains. Proportional-Integral-Derivative (PID) controller including hydraulic dynamics has been developed. The displacement of hydraulic actuator and spool valve is modeled. The Ziegler – Nichols tuning rules are used to determine proportional gain, reset rate and derivative time of PID controller (Ogata 1990). Simulink diagram of active suspension system is developed and analyzed using MATLAB software. The investigations on the performance of the developed active suspension control are demonstrated through comparative simulations in this chapter.

## 2. Active suspension system

Active suspension systems add hydraulic actuators to the passive components of suspension system as shown in Figure 3. The advantage of such a system is that even if the active hydraulic actuator or the control system fails, the passive components come into action. The equations of motion are written as,

$$\begin{aligned} M_s\ddot{z}_s + K_s(z_s - z_{us}) + C_a(\dot{z}_s - \dot{z}_{us}) - u_a = 0 \\ M_{us}\ddot{z}_{us} + K_s(z_{us} - z_s) + C_a(\dot{z}_{us} - \dot{z}_s) + K_t(z_{us} - z_r) + u_a = 0 \end{aligned} \tag{1}$$

where $u_a$ is the control force from the hydraulic actuator. It can be noted that if the control force $u_a = 0$, then Equation (1) becomes the equation of passive suspension system.

Considering $u_a$ as the control input, the state-space representation of Equation (1) becomes,

$$\begin{aligned} \dot{z}_1 &= z_2 \\ \dot{z}_2 &= -\frac{1}{M_s}[K_s(z_1 - z_3) + C_a(z_2 - z_4)] \\ \dot{z}_3 &= z_4 \\ \dot{z}_4 &= \frac{1}{M_{us}}[K_s(z_1 - z_3) + C_a(z_2 - z_4) + K_t(z_3 - z_r)] \end{aligned} \tag{2}$$

where $z_1 = z_S$, $z_2 = \dot{z}_S$, $z_3 = z_{us}$ and $z_4 = \dot{z}_{us}$

## 3. Proportional - Integral - Derivative (PID) controller

PID stands for proportional, integral and derivative. These controllers are designed to eliminate the need for continuous operator attention. In order to avoid the small variation of the output at the steady state, the PID controller is so designed that it reduces the errors by the derivative nature of the controller. A PID controller is depicted in Figure 4. The set-point is where the measurement to be. Error is defined as the difference between set-point and measurement.

(Error) = (set-point) – (measurement), the variable being adjusted is called the manipulated variable which usually is equal to the output of the controller. The output of PID controllers will change in response to a change in measurement or set-point. Manufacturers of PID controllers use different names to identify the three modes. With a proportional controller, offset (deviation from set-point) is present. Increasing the controller gain will make the loop go unstable. Integral action was included in controllers to eliminate this offset. With integral action, the controller output is proportional to the amount of time the error is present. Integral action eliminates offset. Controller Output = (1/Integral) (Integral of) e(t) d(t). With derivative action, the controller output is proportional to the rate of change of the measurement or error. The controller output is calculated by the rate of change of the measurement with time. Derivative action can compensate for a change in measurement. Thus derivative takes action to inhibit more rapid changes of the measurement than proportional action. When a load or set-point change occurs, the derivative action causes the controller gain to move the "wrong" way when the measurement gets near the set-point. Derivative is often used to avoid overshoot. The different between the actual acceleration and desired acceleration is taken as error in this study.

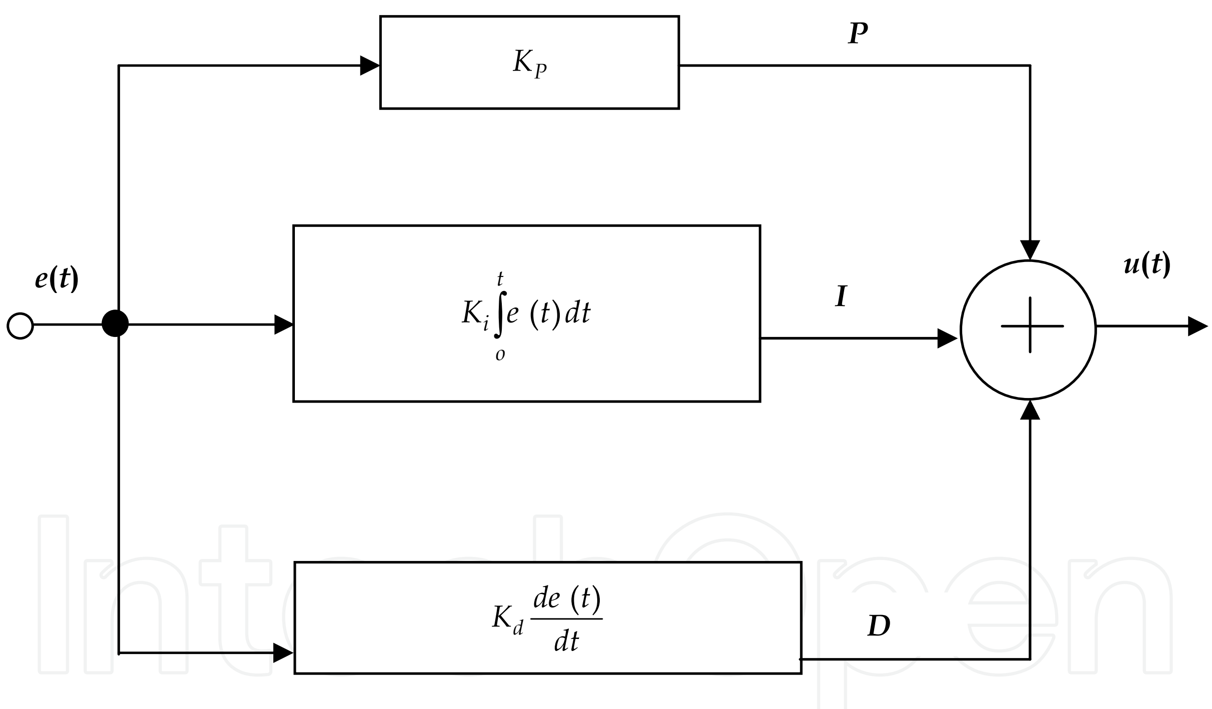



Fig. 4. PID controller

## 4. Hydraulic active suspension system

Block diagram of control system used to develop active suspension system is shown in Figure 5. In order to develop an active suspension system, the following hydraulic components are used.

- Pressurized hydraulic fluid source

- Pressure relief valve to control the pressure of hydraulic fluid
- Direction control valve
- Hydraulic cylinder (active actuator) to convert the hydraulic pressure into force to be transmitted between the sprung and the unsprung mass

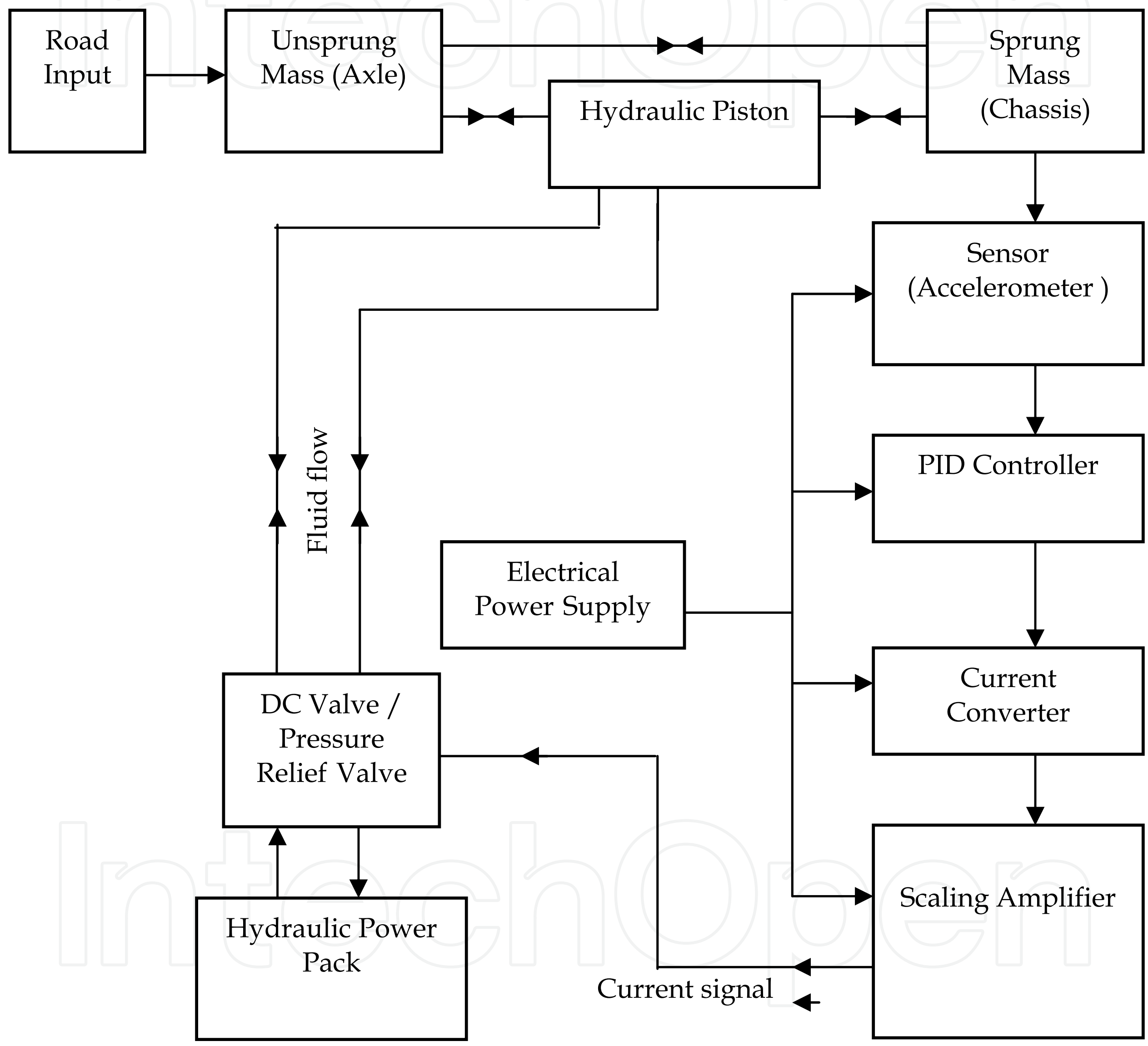



Fig. 5. Block diagram of control system

Figure 6 shows the hydraulic actuator installed in between sprung mass and unsprung mass, including a valve and a cylinder, where $U_h$ is the actuator force generated by the hydraulic piston and $x_{act}$ (= $x_1$-$x_3$) is the actuator displacement. $U_h$ (equal to Ua) is applied dynamically in order to improve ride comfort as and when the road and load input vary.

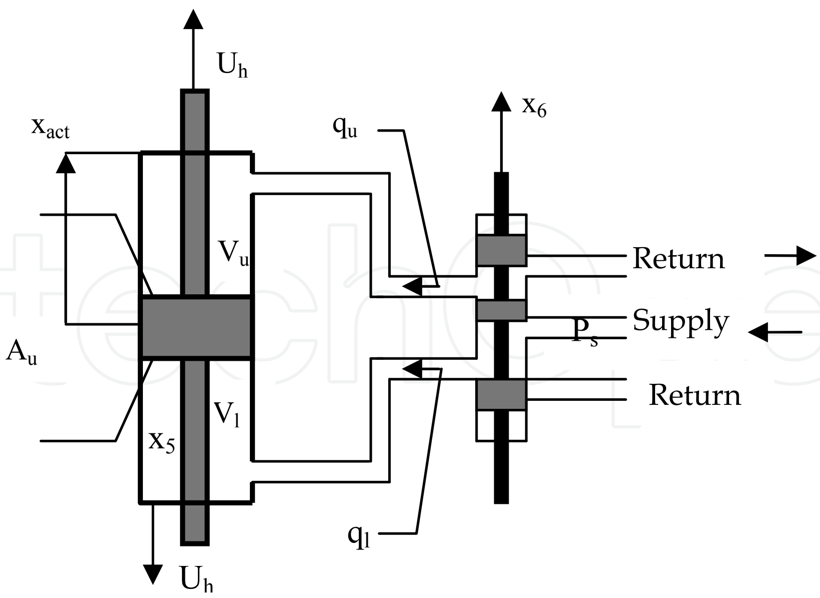


Fig. 6. Hydraulic valve and cylinder

## 5. Controller design

The design of controller is given by,

$$U_c = K_p e(t) + \frac{K_p}{T_i}\int_0^t e(t)dt + K_p T_d \frac{de(t)}{dt} \tag{3}$$

$U_c$ is the current input from the controller, $K_p$ is the proportional gain, $T_i$ and $T_d$ is the integral and derivative time constant of the PID controller respectively.

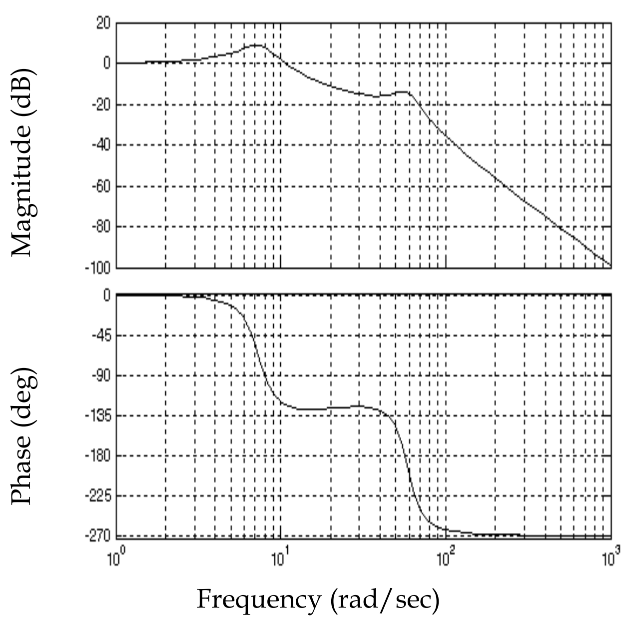



Fig. 7. Bode plot of passive suspension system

The values of gain margin and phase margin obtained from the frequency response plot of car body displacement of the passive suspension system shown in Figure 7 are used to determine the tuning parameters of the PID controller for the active quarter car model. The Ziegler-Nichols tuning rules are used to determine proportional gain, reset rate and derivative time of PID controller.

### 5.1 Tuning of PID controller

The process of selecting the controller parameters to meet given performance specification is known as controller tuning. Zeigler and Nichols suggested rules for tuning PID controllers (meaning to set values $K_p$, $T_i$, $T_d$) based on experimental step responses or based on the values of $K_p$ that results in marginal stability when only proportional control action is used. Ziegler-Nichols rules, which are briefly presented in the section 5.1.1 are very much useful. Such rules suggest a set of values of $K_p$, $T_i$, and $T_d$ that will give a stable operation of the system. However, the resulting system may exhibit a large maximum overshoot in the step response, which is unacceptable. In such a case we need a series of fine tunings until an acceptable results is obtained. In fact, the Zeigler-Nichols tuning rules give an educated guess for the parameter values and provide a starting point for fine tuning, rather than giving the final settings for $K_p$, $T_i$, and $T_d$ in a single shot.

#### 5.1.1 Zeigler-Nichols rules for tuning PID controllers

Zeigler and Nichols proposed rules for determining the proportional gain $K_p$, integral time $T_i$, and derivative time $T_d$ based on the transient response characteristics of a given system. In this method, we first set $T_i = \infty$ and $T_d = 0$. Using the proportional control action only, increase $K_p$ from 0 to a critical value $K_{cr}$ at which the output first exhibits sustained oscillations. Thus the critical gain $K_{cr}$ and the corresponding period $P_{cr}$ are experimentally determined. Zeigler and Nichols suggested that we set the values of the parameter $K_p$, $T_i$, and $T_d$ according to the formula shown in Table 1.

| Type of Controller | Kp | Ti | Td |
|---|---|---|---|
| P | 0.5 $K_{cr}$ | $\infty$ | 0 |
| PI | 0.45 $K_{cr}$ | 0.83 $P_{cr}$ | 0 |
| PID | 0.6 $K_{cr}$ | 0.5$P_{cr}$ | 0.125 $P_{cr}$ |

Table 1. Zeigler-Nichols tuning rules

It can be noted that if the system has a known mathematical model, then we can use the root-locus method to find the critical gain $K_{cr}$ and the frequency of the sustained oscillations $\omega_{cr}$, where $2\pi/\omega_{cr} = P_{cr}$. These values can be found from the crossing points of the root locus branches with the $j\omega$ axis. The passive suspension (open loop) system of the quarter car model is analyzed and the bandwidth and gain margin of the system are found to be 1.92 Hz and –13.9 db respectively as shown in Figure 8. Gain margin is the gain, at which the active suspension (closed loop) system goes to the verge of instability; (Gain margin is the gain in db at which the phase shift of the system is $-180^0$). The gain margin of the system is found to be 4.91. It is the value of the gain, which makes the active suspension (closed loop) system to exhibit sustained oscillation (the vibration of car body of the active suspension (closed loop) system is maximum for this value of gain).

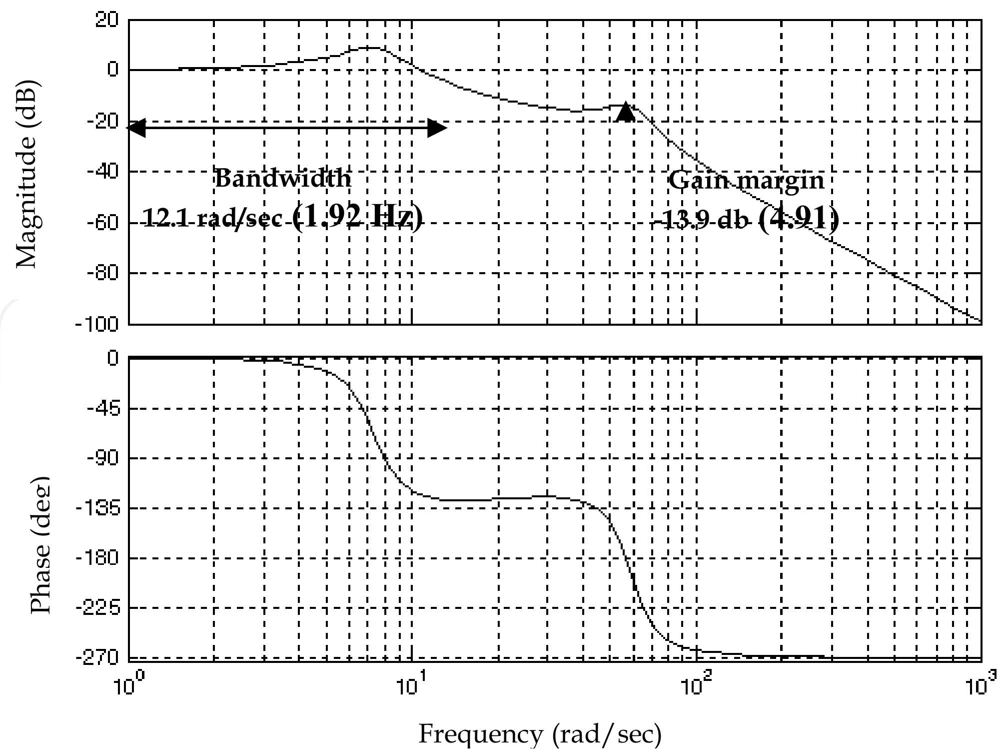



Fig. 8. Open loop unit step response

When the gain of the system is increased beyond 4.915 the response (vibration of car body displacement) of the active suspension (closed loop) system is increased instead of being reduced. The system becomes unstable when the gain of the system is increased beyond 4.915 which is shown in Figure 9.

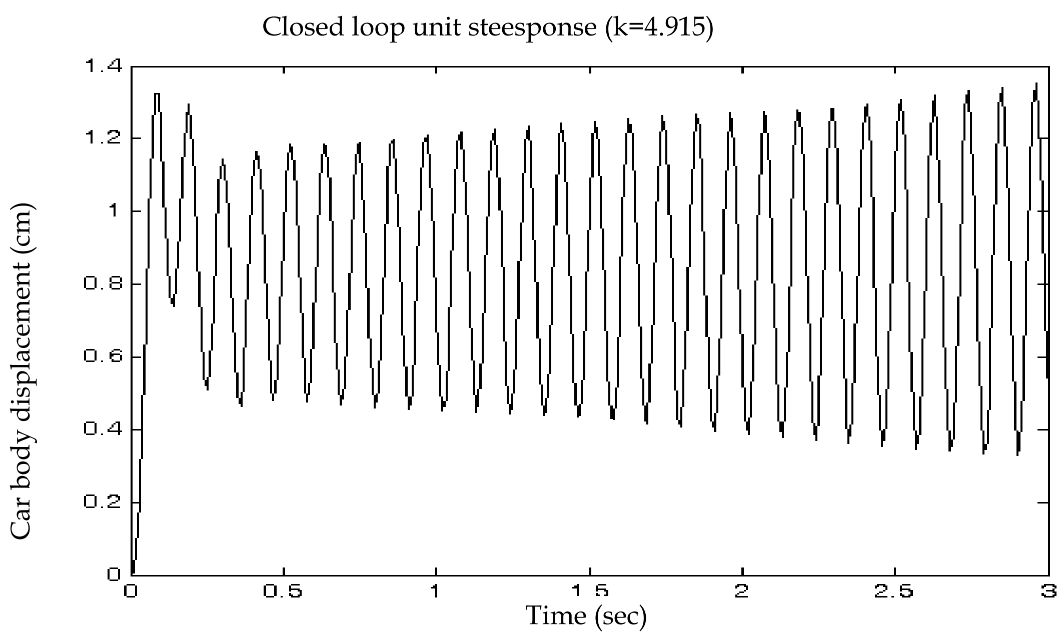



Fig. 9. Closed loop unit step response (k>4.915)

The response of the active suspension (closed loop) system of the quarter car model for the critical gain value ($K_{cr}$ = 4.915) is as shown in Figure 10 and the time period of the sustained oscillation for this value of critical gain $K_{cr}$ is called critical period $P_{cr}$, which is determined from the step response of the closed loop system and is found to be $P_{cr}$ = 0.115 sec.

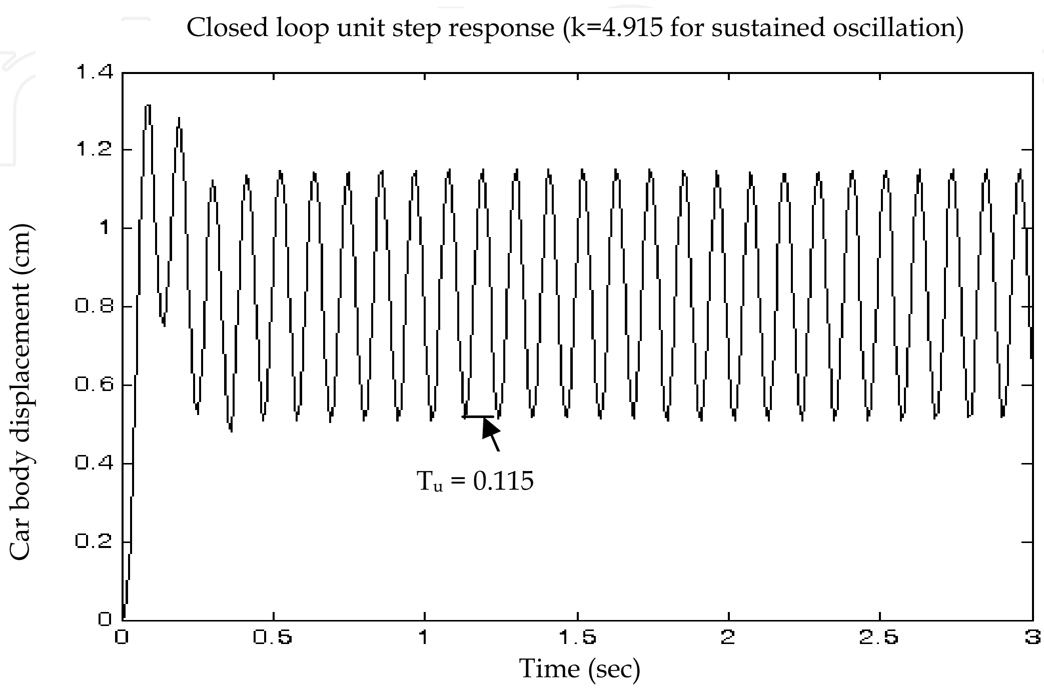



Fig. 10. Closed loop unit step response (k=4.915) for sustained oscillation

The critical gain ($K_{cr}$) and critical time period ($P_{cr}$), determined above are used to set the tuning rules for the quarter car model using the Zeigler-Nichols tuning rules. As discussed, the values of the P, PI and PID controller are obtained and are tabulated in Table 2.

| Type of Controller | Kp | Ti | Td |
|---|---|---|---|
| P | 2.4575 | $\infty$ | 0 |
| PI | 2.212 | 0.096 | 0 |
| PID | 2.95 | 0.0575 | 0.014 |

Table 2. Zeigler-Nichols tuning values

### 5.2 Hydraulic dynamics

Three-land four-way valve-piston system as shown in Figure 6 is used in the hydraulic controller design. The force $U_h$ from the actuator is given by,

$$U_h = AP_L \tag{4}$$

where A (=$A_u$, Area of upper chamber; = $A_l$, Area of lower chamber) is the piston area and $P_L$ is the pressure drop across the piston. Following Merritt (1967), the derivative of $P_L$ is given by

$$\frac{V_t}{4\beta}\dot{P}_L = Q - C_{tp}P_L - A(x_2 - x_4) \tag{5}$$

where $V_t$ is the total actuator volume, $\beta$ is the effective bulk modulus of the fluid, Q is the hydraulic load flow ($Q = q_u + q_l$), where $q_u$ and $q_l$ are the flows in the upper and lower chamber respectively and $C_{tp}$ is the total leakage coefficient of the piston. In addition, the valve load flow equation is given by

$$Q = C_d \omega x_6 \sqrt{\frac{1}{\rho}\left[P_s - sgn(x_6)x_5\right]} \tag{6}$$

where $C_d$ is the discharge coefficient, $\omega$ is the spool valve area gradient, $x_5$ is the pressure inside the chamber of hydraulic piston and $x_6 = x_{sp}$ is the valve displacement from its closed position, $\rho$ is the hydraulic fluid density and $P_s$ is the supply pressure, Since, the term $\left[P_s - sgn(x_6)x_5\right]$ may become negative, Equation (6) is replaced with the corrected flow equation as,

$$Q = sgn\left[P_s - sgn(x_6)x_5\right] C_d \omega x_6 \sqrt{\frac{1}{\rho}\left[P_s - sgn(x_6)x_5\right]} \tag{7}$$

Finally, the spool valve displacement is controlled by the input to the valve $U_c$, described by Equation (3), which could be a current or voltage signal. Equations (2) to (7) used to derive the equation of the active suspension system, including the hydraulic dynamics are rewritten as Equation (8).

$$\begin{aligned}
\dot{x}_1 &= x_2 \\
\dot{x}_2 &= \frac{K_s}{M_s}x_1 + \frac{C_a}{M_s}x_2 + \frac{K_s}{M_s}x_3 + \frac{C_a}{M_s}x_4 + \frac{A_l}{M_s}x_5 \\
\dot{x}_3 &= x_4 \\
\dot{x}_4 &= \frac{K_s}{M_{us}}x_1 + \frac{C_a}{M_{us}}x_2 + \frac{K_s + K_t}{M_{us}}x_3 + \frac{C_a}{M_{us}}x_4 + \frac{A_l}{M_{us}}x_5 + \frac{K_t}{M_{us}}r \\
\dot{x}_5 &= \beta x_5 + A(x_2 - x_4) + x_6\omega_3 \\
\dot{x}_6 &= \frac{x_6}{\tau} + U_c
\end{aligned} \tag{8}$$

where, $\omega_3 = sgn\left[P_s - sgn(x_6)x_5\right]\sqrt{\left|P_s - sgn(x_6)x_5\right|}$

Thus Equation (8) becomes state feedback model of active suspension system including hydraulic dynamics. Figure 11 represents the Simulink model of active suspension system.

## 6. Simulation

To ensure that our controller design achieves the desired objective, the open loop passive and closed loop active suspension system are simulated with the following values.

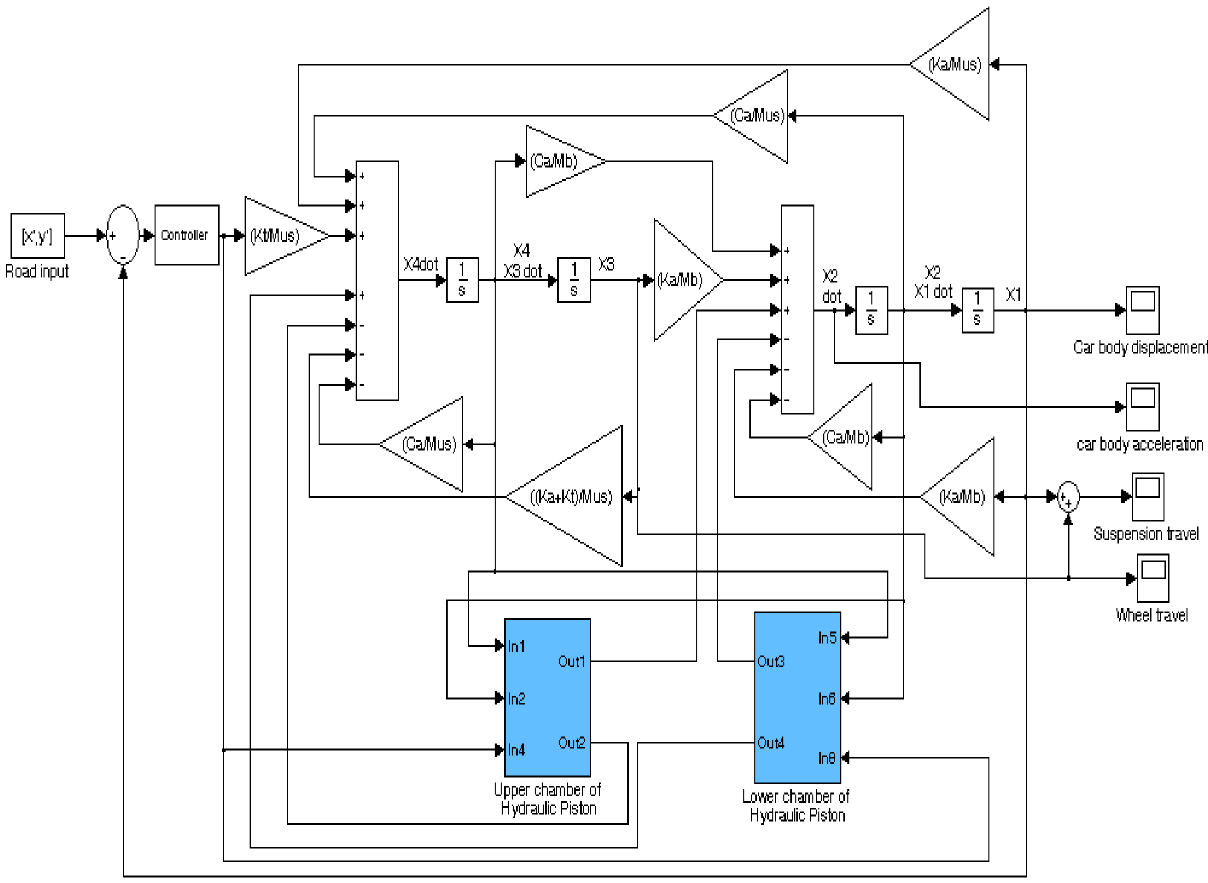



Fig. 11. Simulink model of active suspension system

| | |
|---|---|
| $M_b$ = 300 Kg | $K_t$ = 190000 N/m |
| $M_{us}$ = 60 Kg | $\beta$ = 1 $sec^{-1}$ |
| $K_a$ = 16850 N/m | $P_s$ = 10.55 MPa |
| $C_a$ = 1000 N/(m/sec) | |

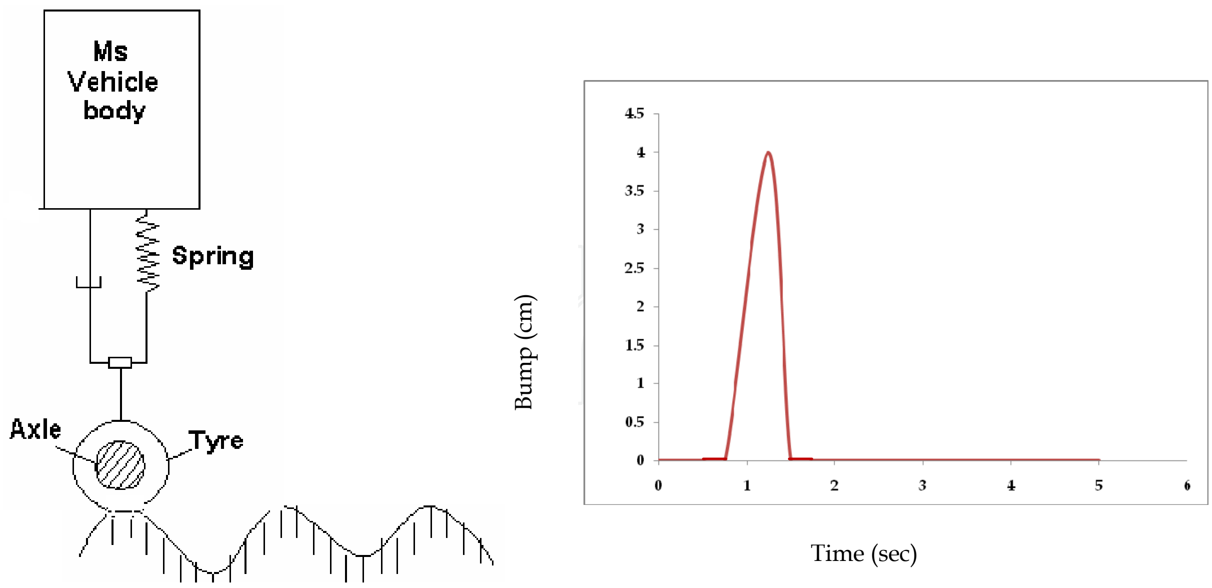
### 6.1 Bumpy road (sinusoidal input)

A single bump road input, $Z_r$ as described by (Jung-Shan Lin 1997), is used to simulate the road to verify the developed control system. The road input described by Equation (9) is shown in Figure 12.

$$Z\,r = \begin{cases} a(1-\cos\omega t) & 0.5 \le t \le 0.75 \\ 0, & otherwise, \end{cases} \tag{9}$$

In Equation (9) of road disturbance, 'a' is set to 0.02 m to achieve a bump height of 4 cm. All the simulations are carried out by MATLAB software. The following assumptions are also made in running the simulation.

a. Suspension travel limits: $\pm$ 8 cm
b. Spool valve displacement $\pm$ 1 cm




(a) Actual bumpy road (b) Bumpy road input

Fig. 12. Road input disturbance

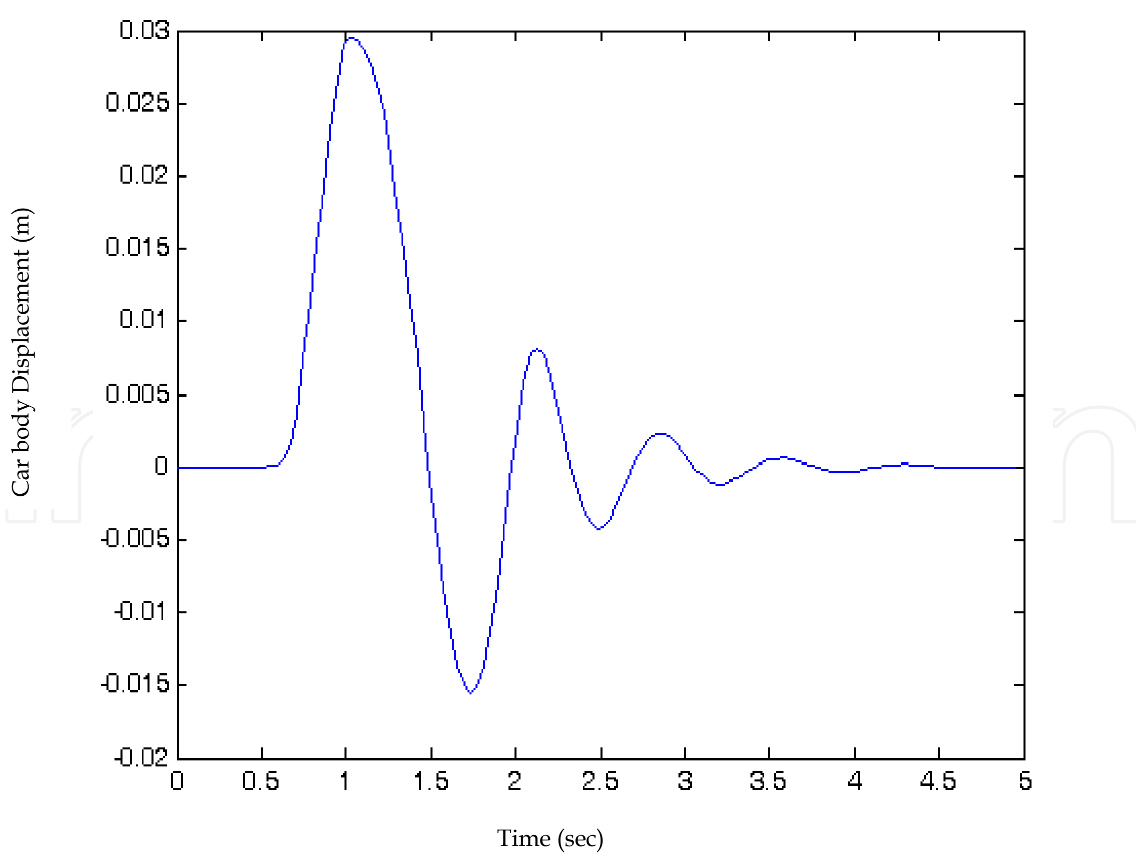



Fig. 13. Car body displacement of passive suspension system

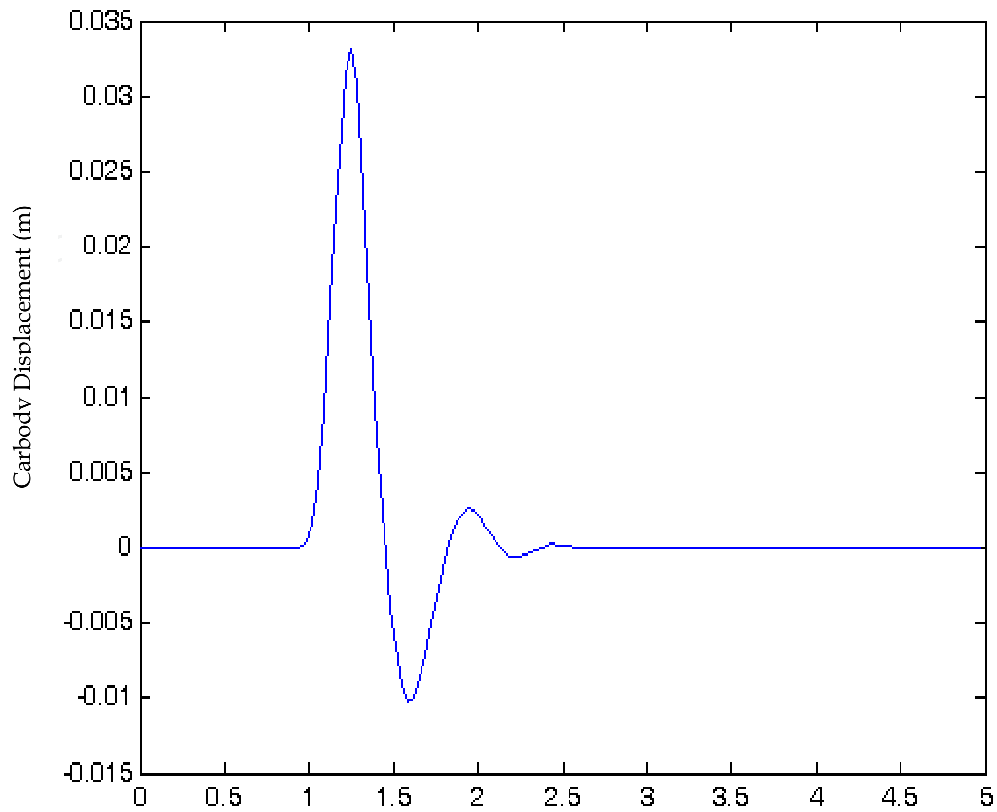



Fig. 14. Car body displacement of active suspension system

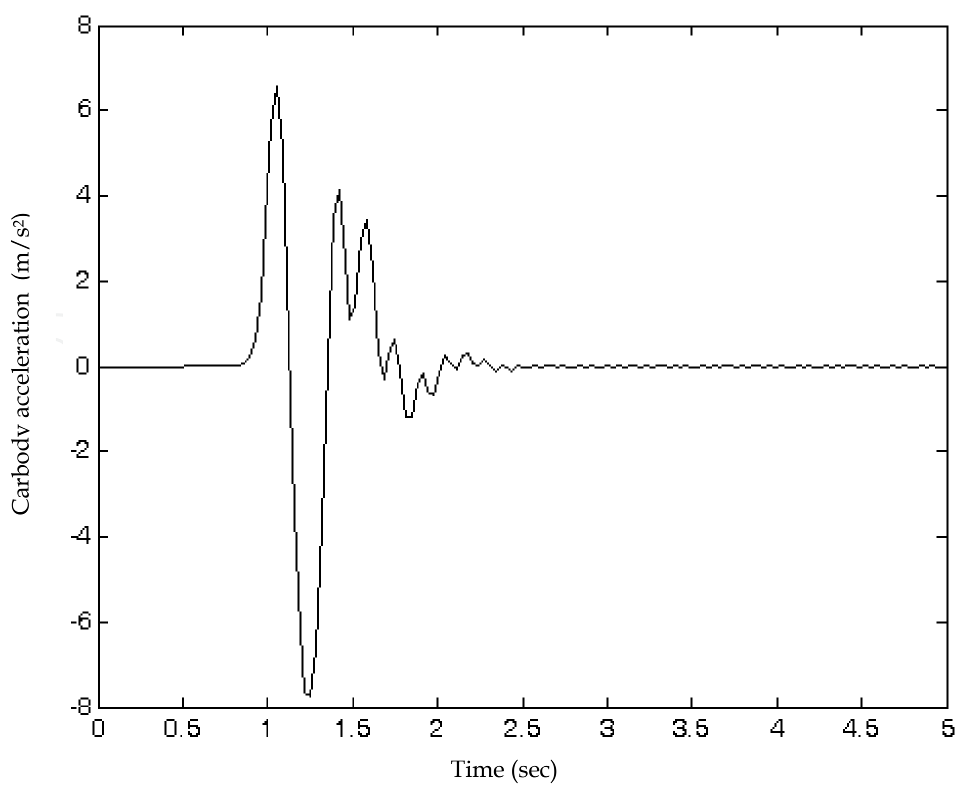



Fig. 15. Car body acceleration of passive suspensıon system

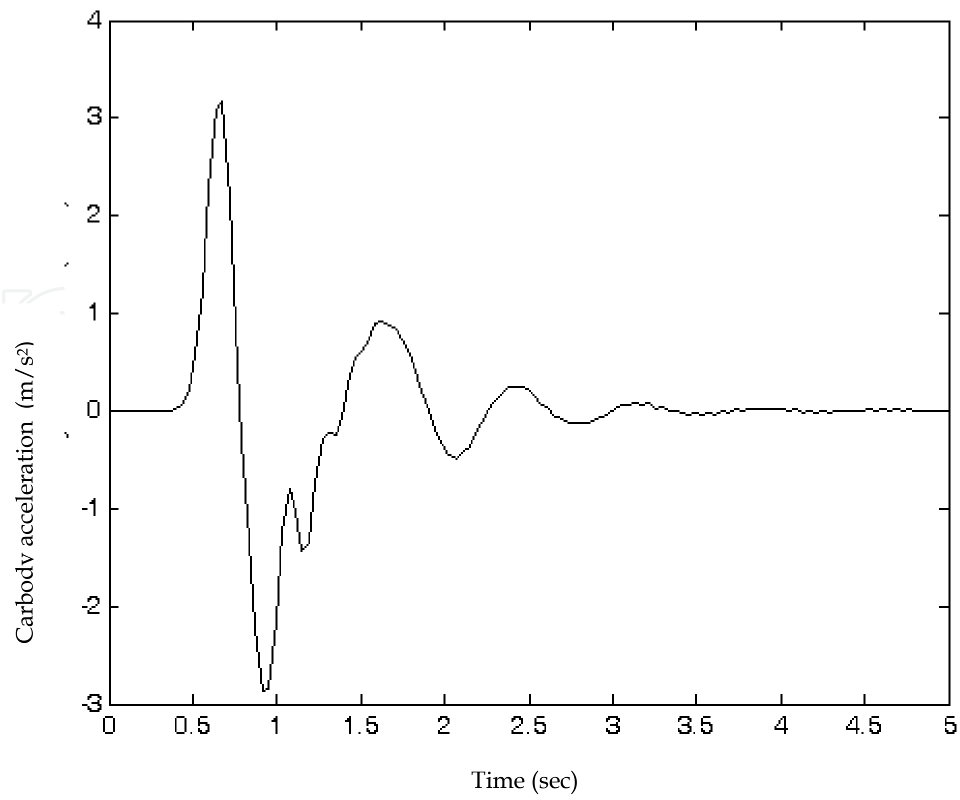



Fig. 16. Car body acceleration of active suspension system

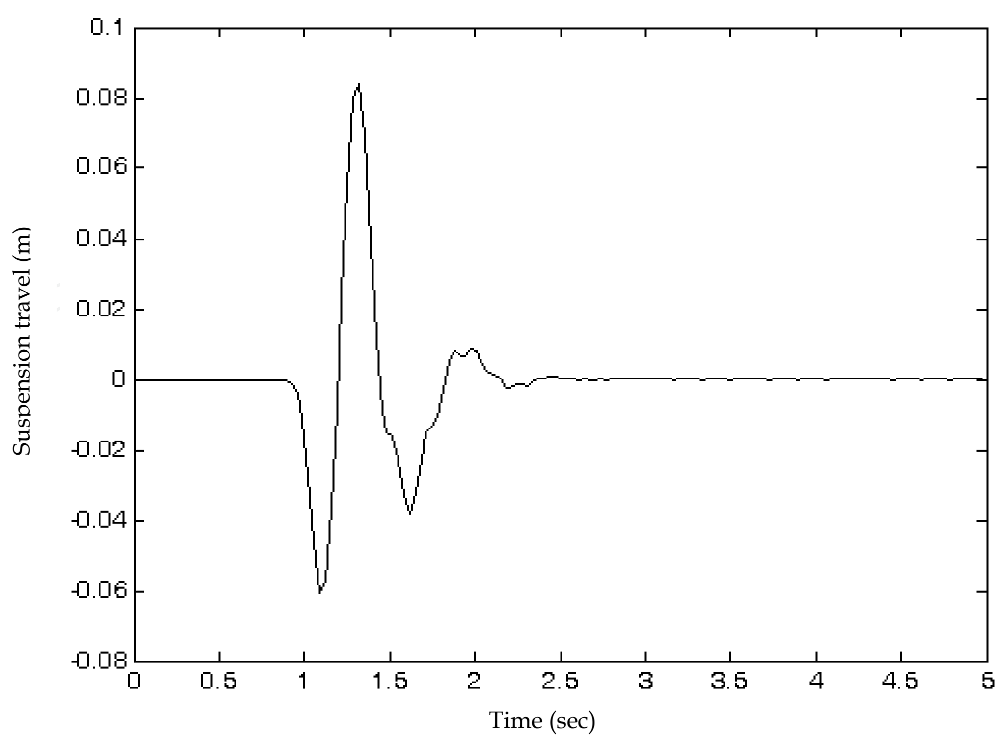



Fig. 17. Suspension travel of passive suspension system

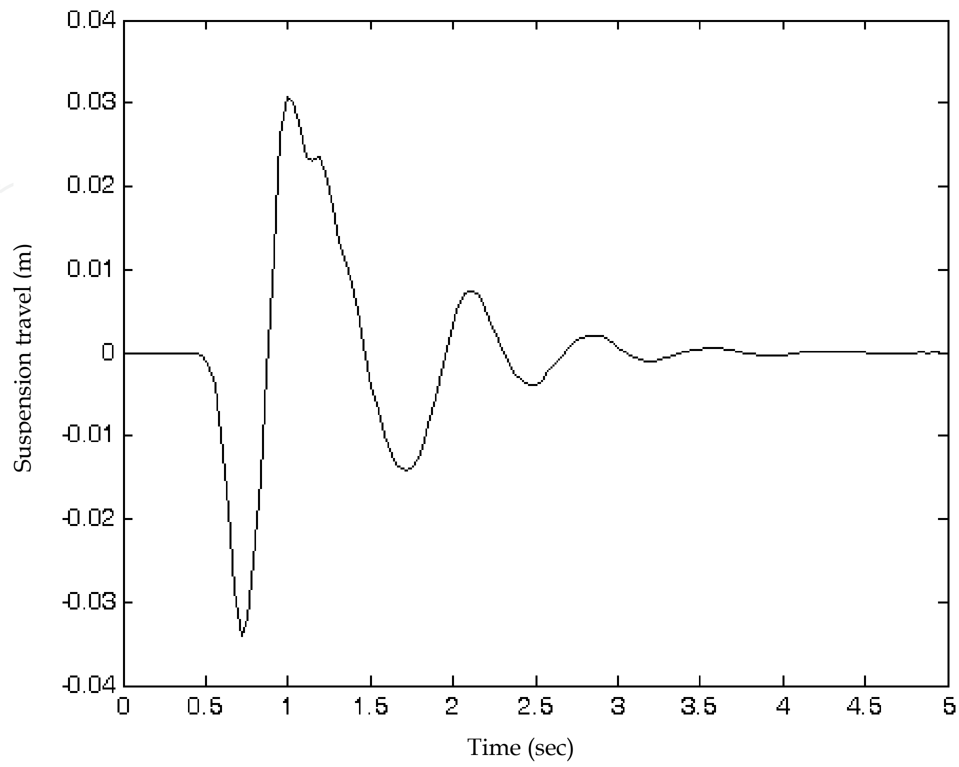



Fig. 18. Suspension travel of active suspension system

Figures 13-18 represent the time response plots of car body displacement, car body acceleration and suspension travel of both passive and active suspension system without tuning of controller parameters respectively. The PID controller designed produces a large spike (0.0325 m) in the transient portion of the car body displacement response of active suspension system as shown in Figure 14, compared to the response (0.03 m) of passive suspension system shown in Figure 13. The spike is due to the quick force applied by the actuator in response to the signal from the controller. Even though there is a slight penalty in the initial stage of transient vibration in terms of increased amplitude of displacement, the vibrations are settled out faster as it takes only 2.5 sec against 4.5 sec taken by the passive suspension system as found from Figure 14.

The force applied between sprung mass and unsprung mass would not produce an uncomfortable acceleration for the passengers of the vehicle, which is depicted in Figure 15 (3.1 $m/s^2$ in active system), in comparison with the acceleration (6.7 $m/s^2$) of passenger experienced in passive system as shown in Figure 15. Also, it is found that the suspension travel (0.031 m) is very much less as seen in Figure 18 compared with suspension travel (0.081 m) of passive suspension system as seen in Figure 17. Therefore rattle space utilization is very much reduced in active suspension system when compared with passive suspension system in which suspension travel limit of 8 cm is almost used.

Figures 19-22 represent the behavior of both active suspension systems with tuned parameters.

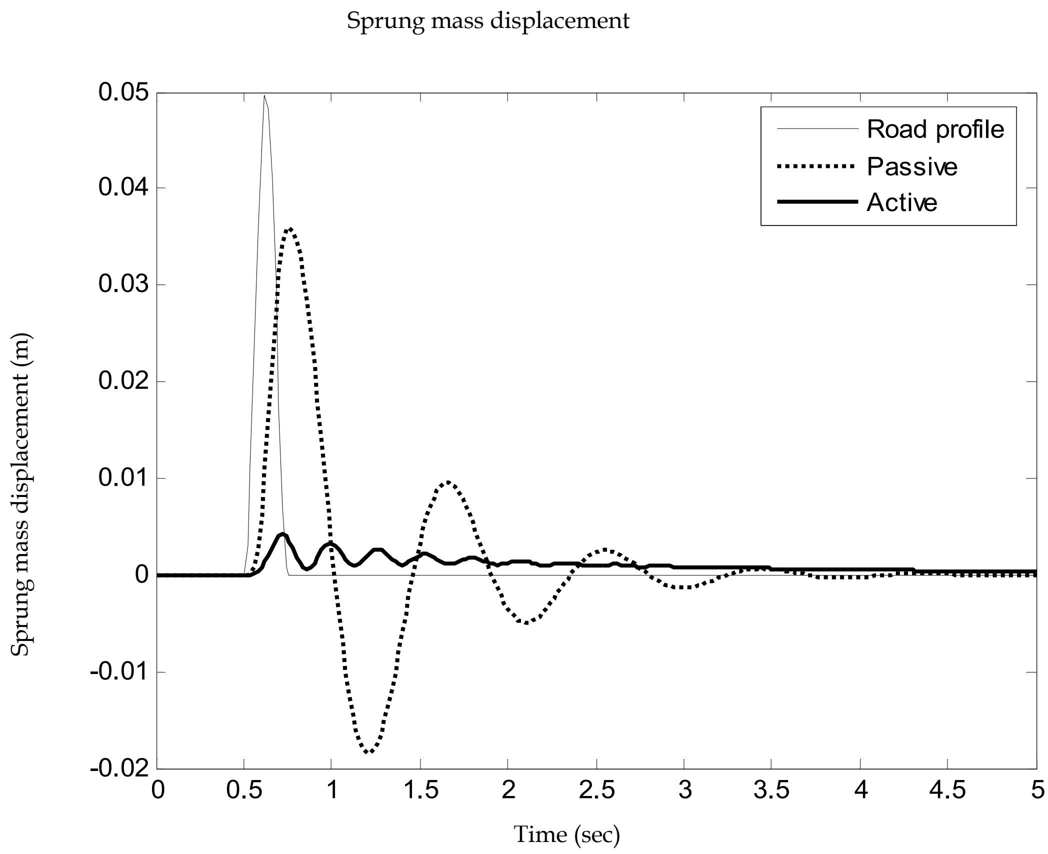



Fig. 19. Sprung mass displacement Vs time (Bumpy road)

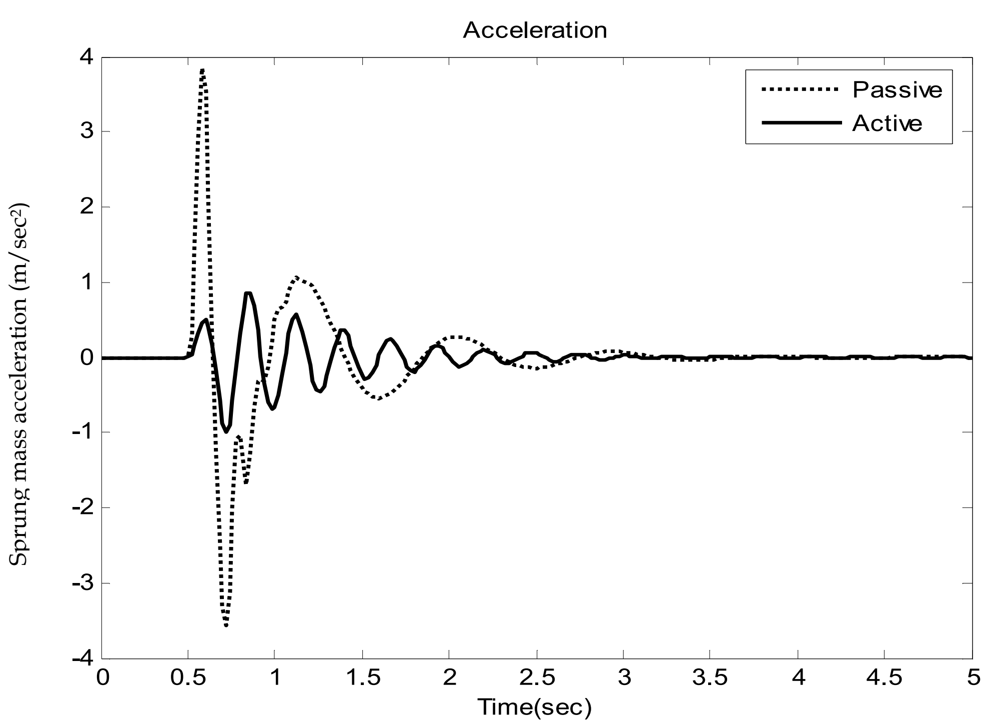



Fig. 20. Sprung mass acceleration Vs time (Bumpy road)

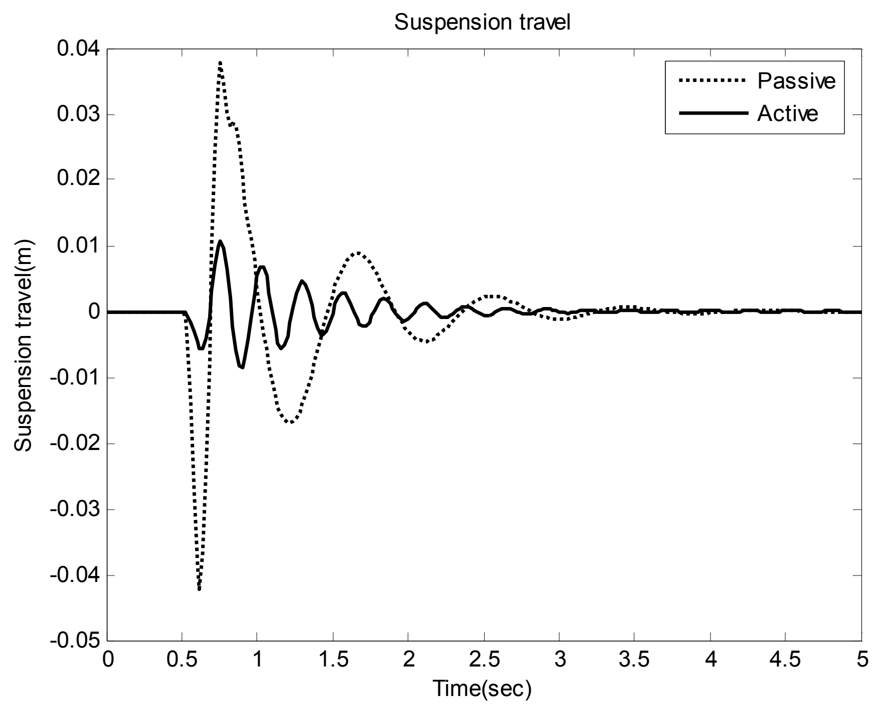



Fig. 21. Suspension travel Vs time (Bumpy road)

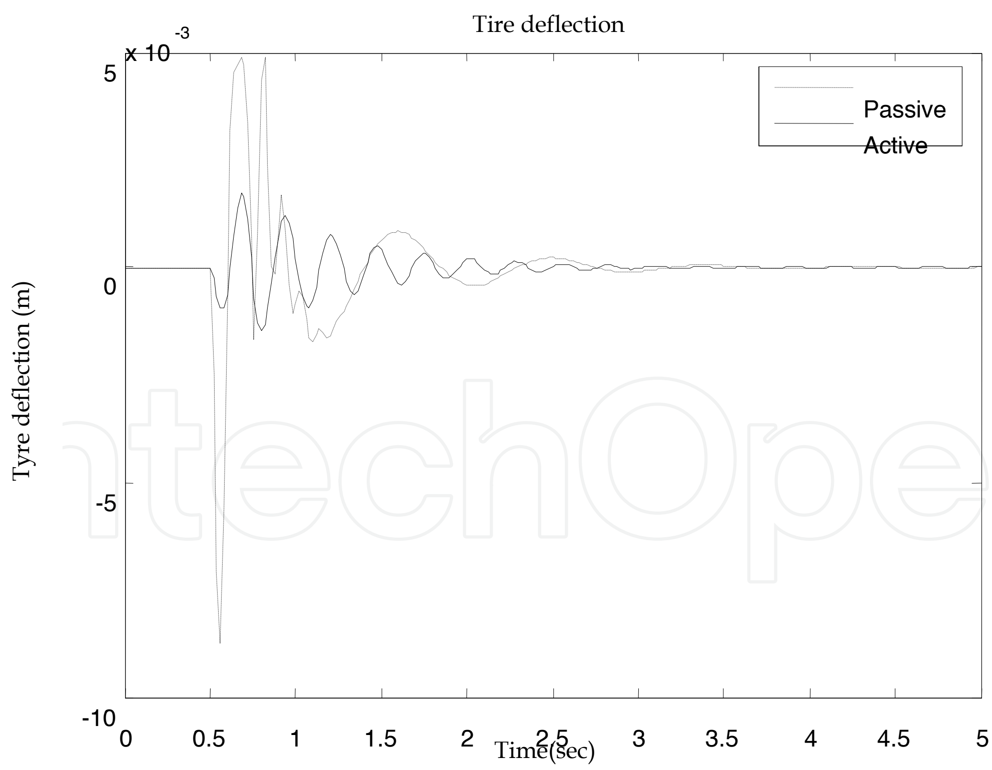



Fig. 22. Tyre deflection Vs time (Bumpy road)

Figures 19–22 illustrate that both peak values and settling time have been reduced by the active system compared to the passive system for all the parameters of sprung mass displacement, sprung mass acceleration (ride comfort), suspension travel and tyre deflection (road holding). Table 3 gives the percentage reduction in peak values of the various parameters for the sinusoidal bumpy road input.

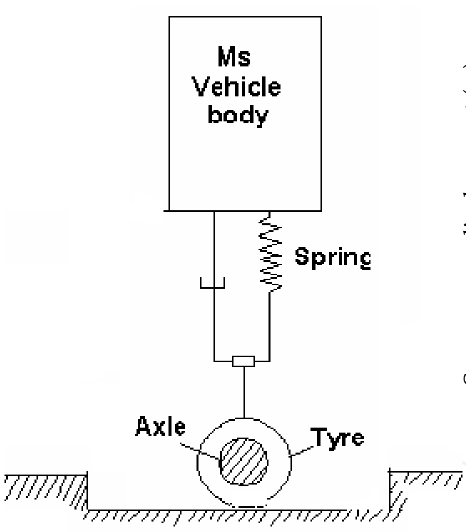
| Parameter | Passive | Active | % Reduction |
|---|---|---|---|
| Sprung Mass Acceleration | 3.847 m/s² | 0.845m/s² | 78.03 |
| Suspension Travel | 0.038 m | 0.011 m | 71.05 |
| Tyre Deflection | 0.005 m | 0.002 m | 60.00 |

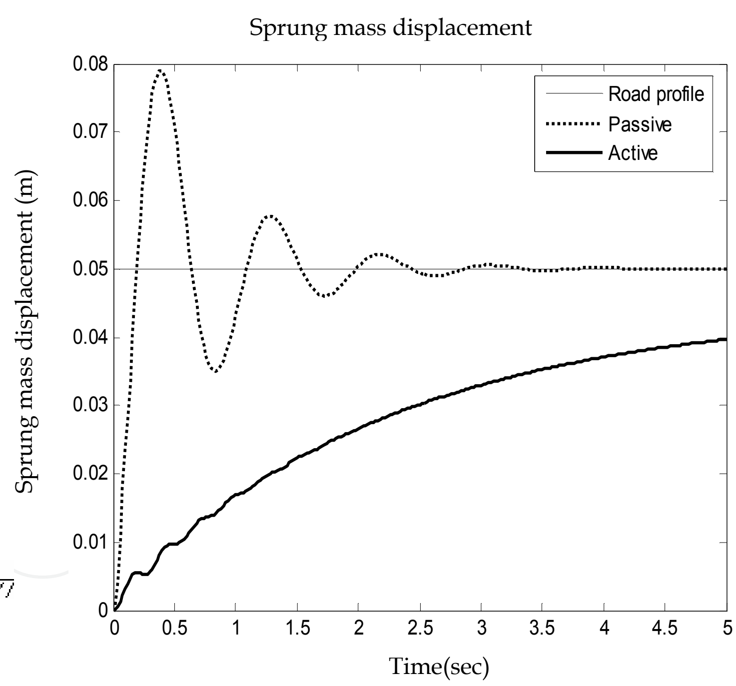
Table 3. Reduction in peak values different parameters (Bumpy road)

### 6.2 Pot-hole (step input)

The step input characterizes a vehicle coming out of a pothole. The pothole has been represented in the following form.

$$Z_r = \begin{cases} 0 & \rightarrow \ t \le 1 \\ 0.05 & \rightarrow \ t > 1 \end{cases} \tag{10}$$




(a) Pot hole (b) Sprung mass displacement Vs time

Fig. 23. Sprung mass displacement (Pot hole)

Figures 23-26 illustrates the performance comparison between passive and active suspension system for the vehicle coming out of a pot-hole of height 0.05 m.

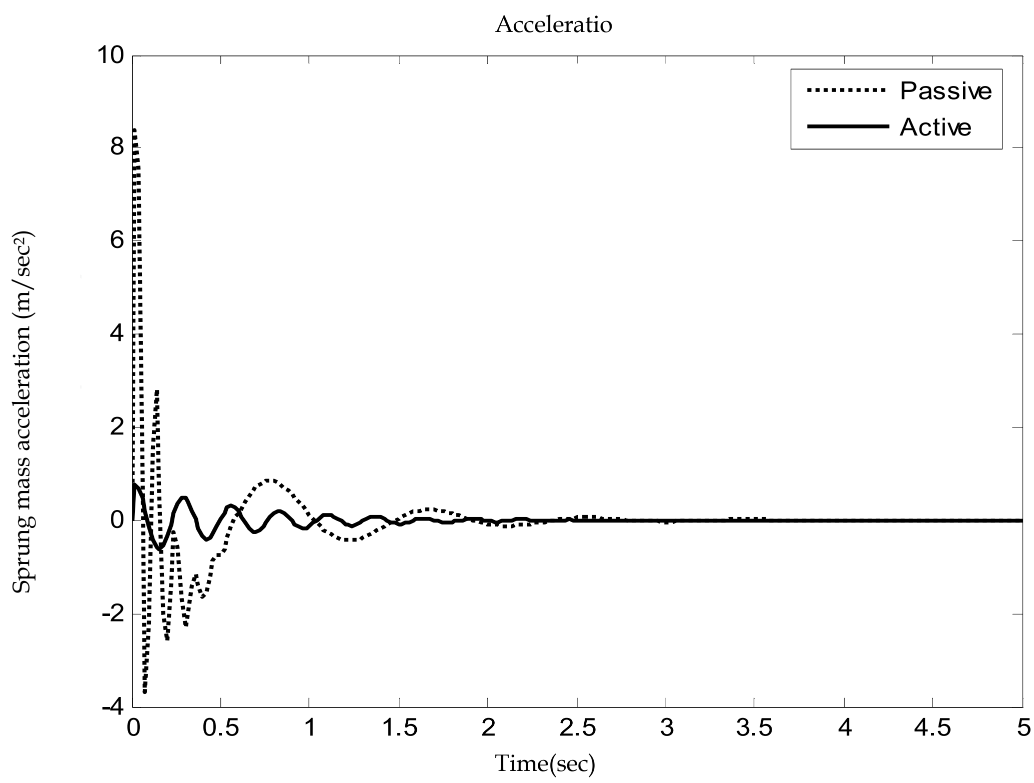



Fig. 24. Sprung mass acceleration Vs time (Pot hole)

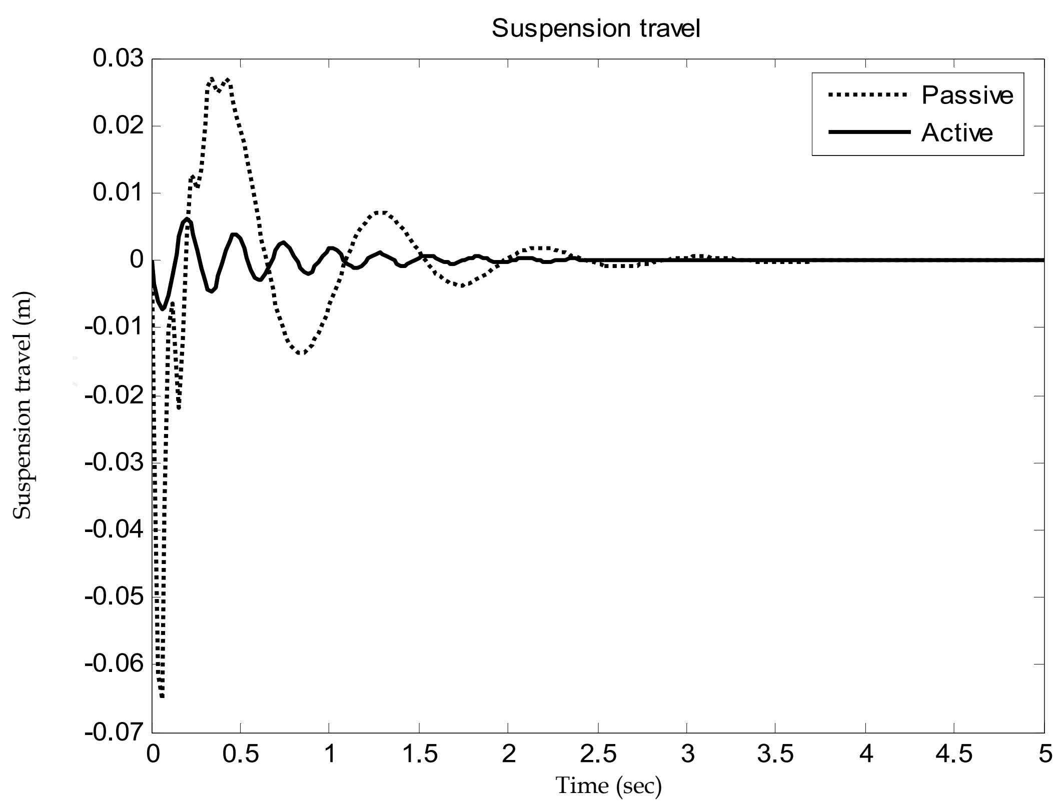



Fig. 25. Suspension travel Vs time (Pot hole)

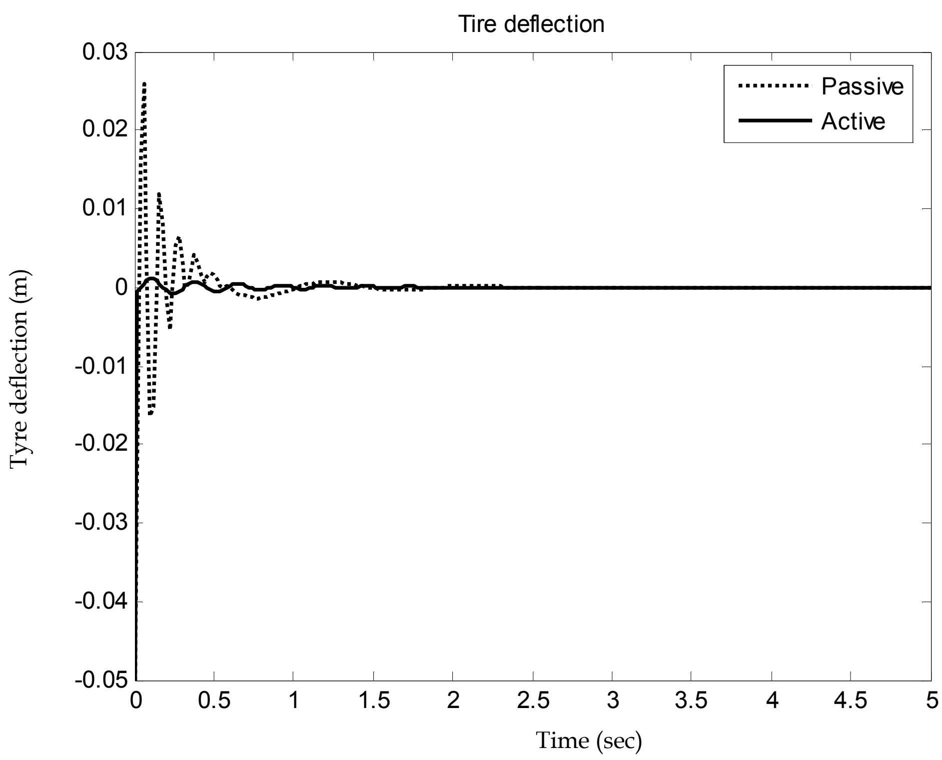



Fig. 26. Tyre deflection Vs time (Pot hole)

| Parameter | Passive | Active | % Reduction |
|---|---|---|---|
| Sprung Mass Acceleration | 8.3793 $m/s^2$ | 0.7535 $m/s^2$ | 91.01 |
| Suspension Travel | 0.06510 m | 0.00732 m | 88.75 |
| Tyre Deflection | 0.02597 m | 0.00102 m | 96.04 |

Table 4. Reduction in peak values of different parameters (Pot-hole)

From Figures 23-26, it could be observed that both peak overshoot and settling time have been reduced by the active system compared to the passive system for sprung mass acceleration, suspension travel, and tyre deflection. Table 4 shows the percentage reduction in various parameters which guarantees the improved performance by active suspension system.

### 6.3 Random road

Apart from sinusoidal bumpy and pot-hole type of roads, a real road surface taken as a random exciting function is used as input to the vehicle. It is noted that the main characteristic of a random function is uncertainty. That is, there is no method to predict an exact value at a future time. The function should be described in terms of probability statements as statistical averages, rather than explicit equations. In road model power spectral density has been used to describe the basic properties of random data.

Several attempts have been made to classify the roughness of a road surface. In this work, classifications are based on the International Organization for Standardization (ISO). The ISO has proposed road roughness classification (classes A-H) based on the power spectral density values is as shown in Figure 27.

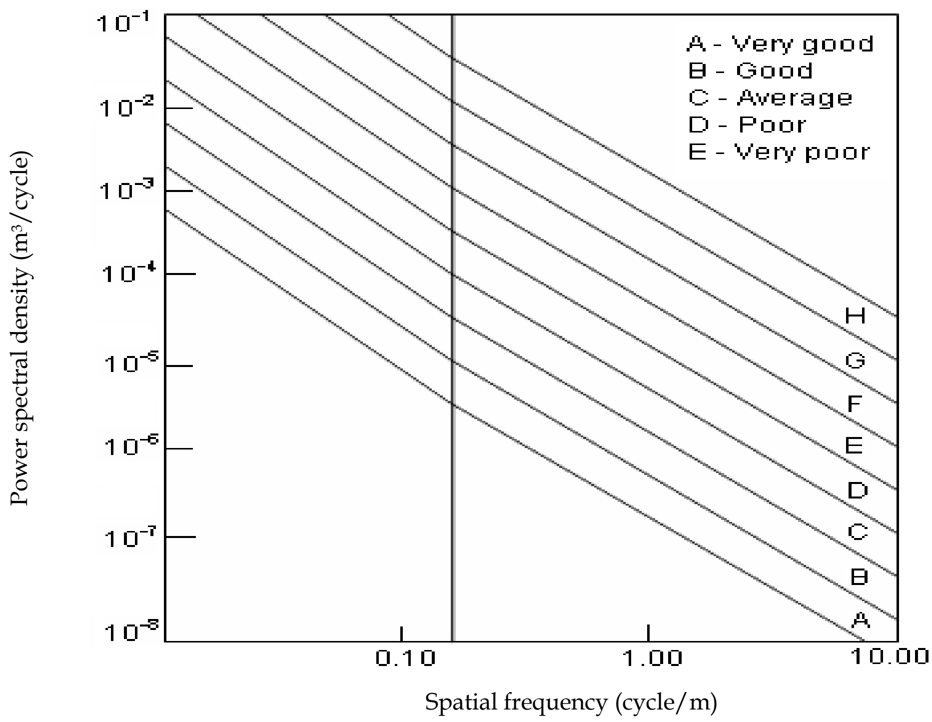



Fig. 27. Road roughness classification by ISO (2000)

### 6.3.1 Sinusoidal approximation

A random profile of a single track can be approximated by a superposition of $N \rightarrow \infty$ sine waves

$$Z_r(s)=\sum_{i=1}^{N} A_i \sin\left(\Omega_i s-\psi_i\right) \tag{11}$$

where each sine wave is determined by its amplitude $A_i$ and its wave number $\Omega_i$. By different sets of uniformly distributed phase angles $\psi_i$, i = 1(1) N in the range between 0 and 2π different profiles can be generated which are similar in the general appearance but different in details.

A realization of the class E road is shown in Figure 28. According to Equation (11) the profile z = z(s) was generated by N = 10 sine waves in the frequency range from 0.1cycle/m (0.628rad/m) to 1cycle/m (6.283rad/m). The amplitudes $A_i$, i = 1(1)N were calculated and the MATLAB function 'rand' was used to produce uniformly distributed random phase angles in the range between 0 and 2π. Figure 28 shows road profile input in time domain.

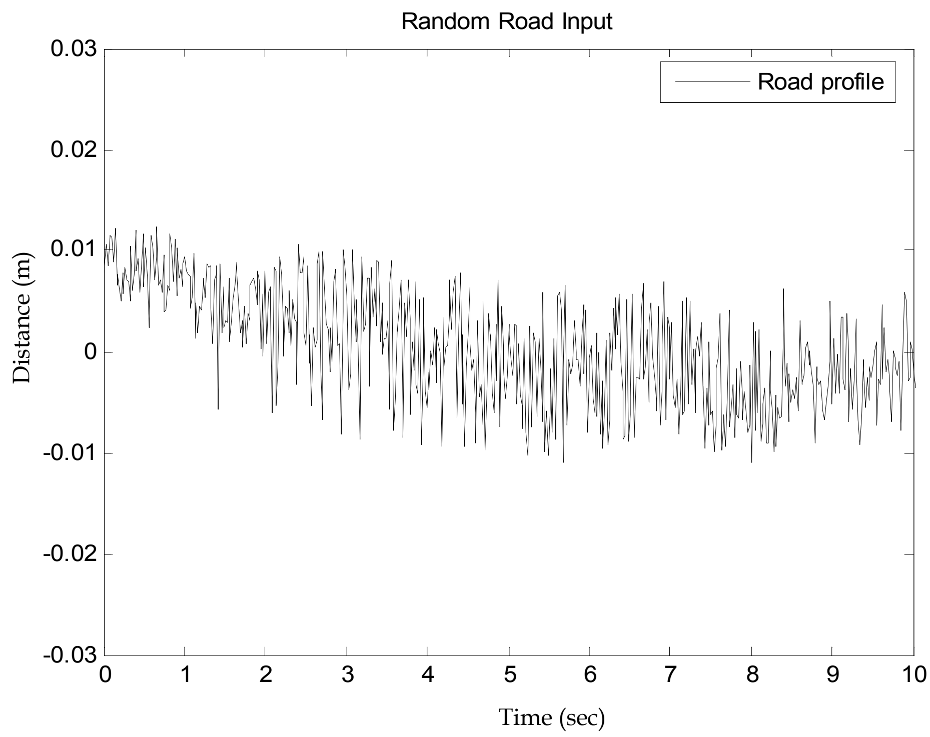



Fig. 28. Road disturbance Vs time (Random)

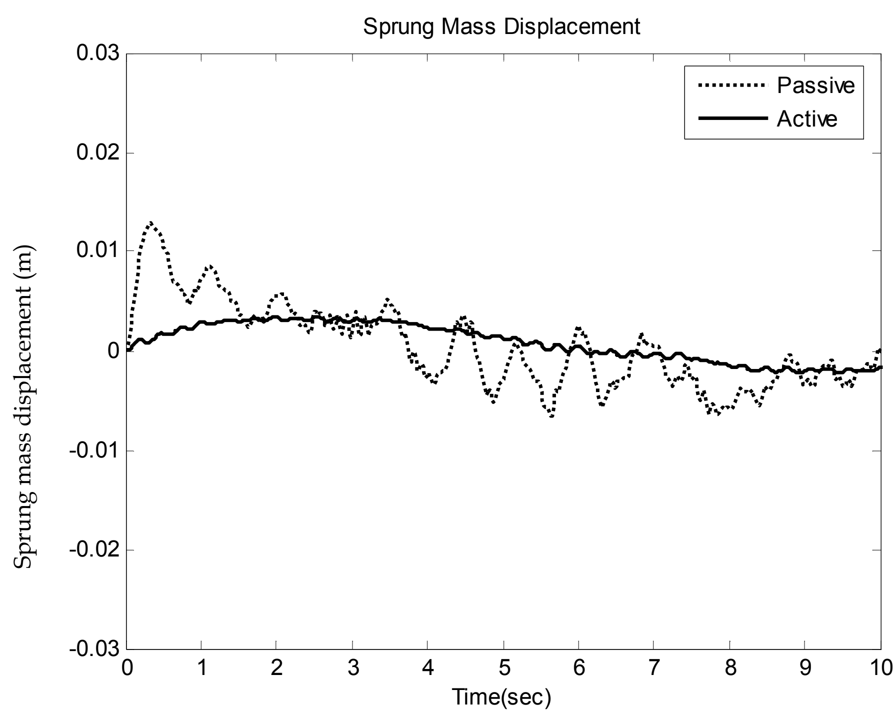



Fig. 29. Sprung mass displacement Vs time (Random road)

Figures 29-32 represent the behaviour of both passive and active suspension systems subjected to random road profile. Table 5 shows the percentage reduction in peak values of suspension parameters.

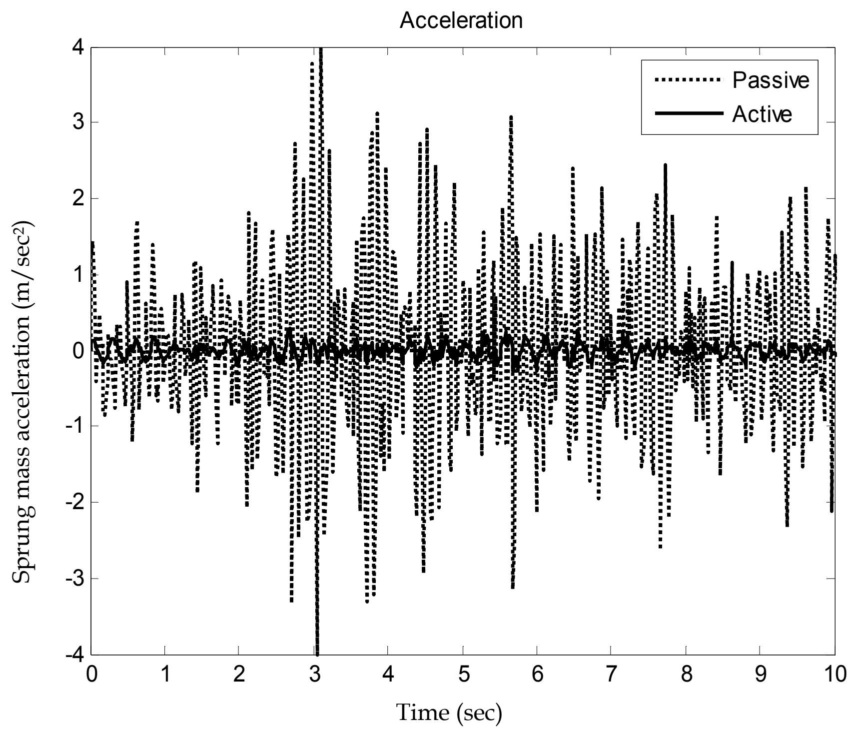



Fig. 30. Sprung mass acceleration Vs time (Random road)

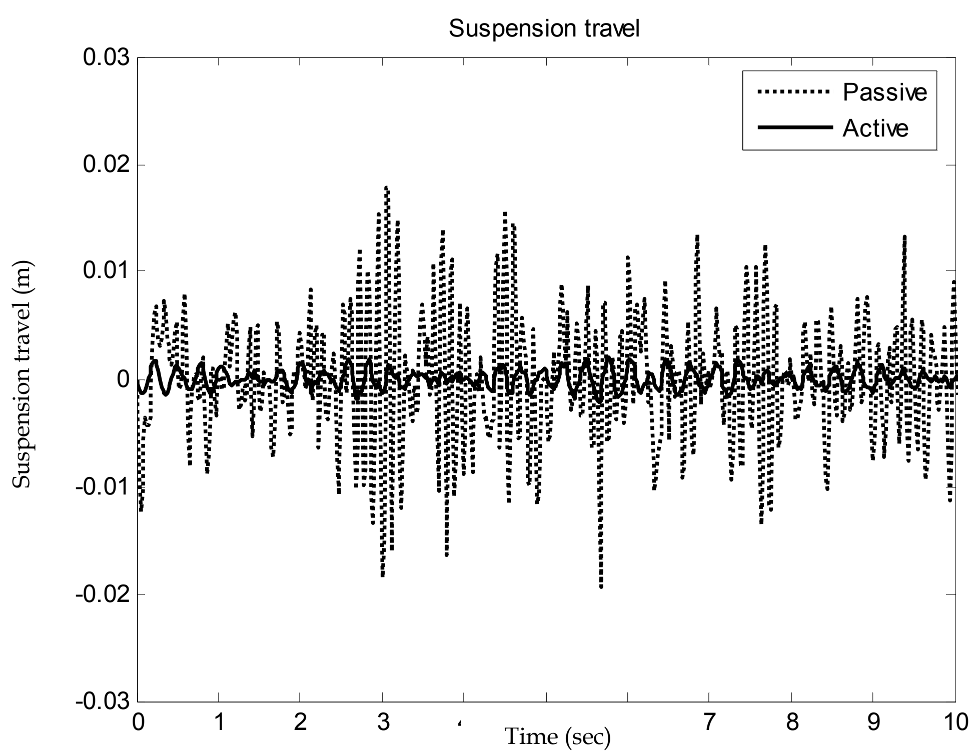



Fig. 31. Suspension travel Vs time (Random road)

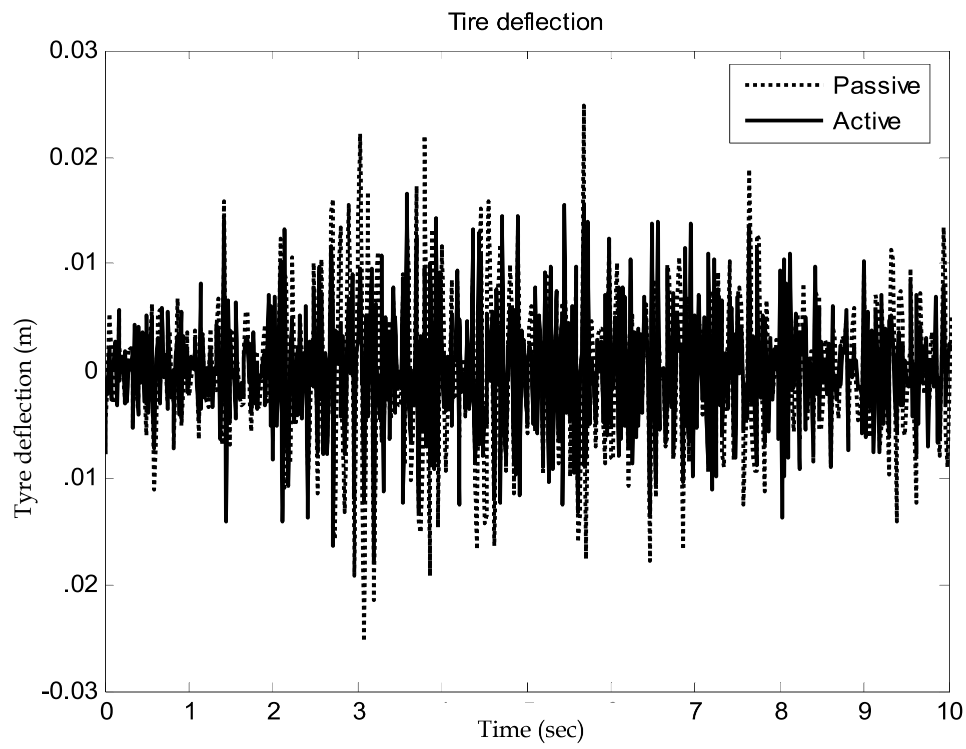



Fig. 32. Tyre deflection Vs time (Random road)

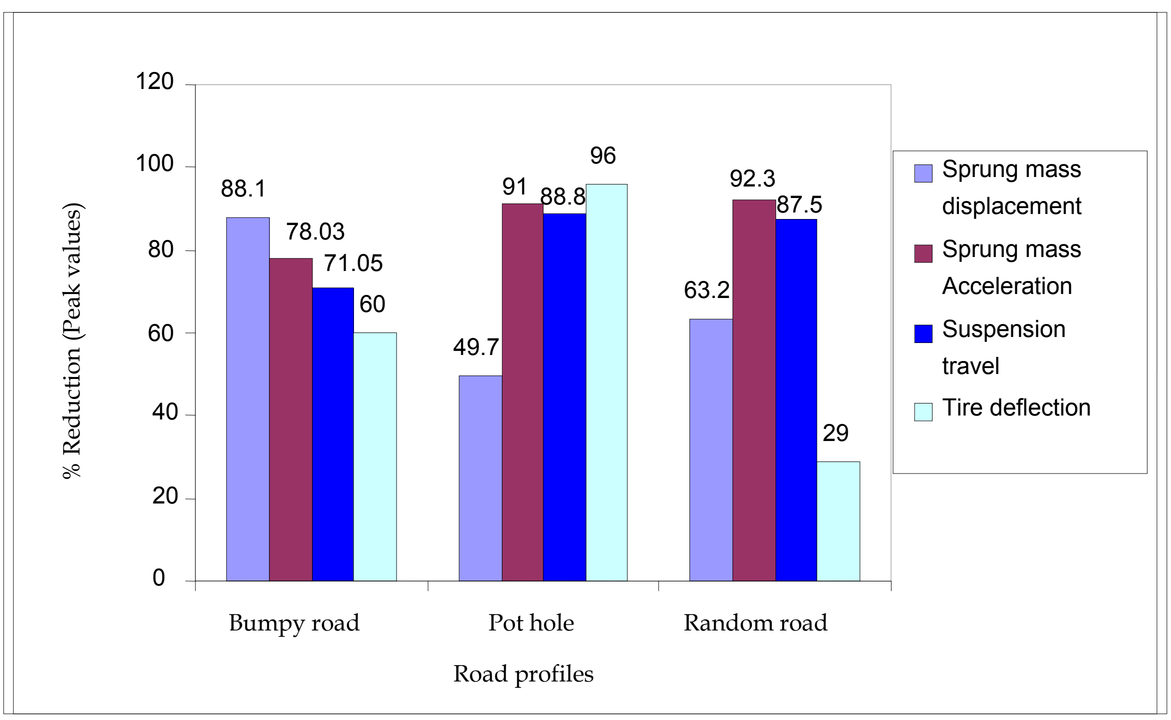



Fig. 33. Active system performance for various road inputs

| Parameter | Passive | Active | % Reduction |
|---|---|---|---|
| Sprung Mass Acceleration | 4.1551 $m/s^2$ | 0.3203 $m/s^2$ | 92.29 |
| Suspension Travel | 0.0195 m | 0.0024 m | 87.45 |
| Tyre Deflection | 0.0254 m | 0.018 m | 29.02 |

Table 5. Reduction in peak values of different parameters (Random road)

It is illustrated that both peak overshoot and settling time have been reduced by the active system compared to the passive system for all the parameters of sprung mass displacement, sprung mass acceleration (ride comfort), suspension travel. Moreover, there is no significant decrease in tyre deflection, but still it is lesser than the static spring deflection. The reason for no improvement in the tyre deflection is the wheel oscillations due to sudden variations of road profile due to randomness.

Figure 33 shows the percentage reduction of the peak values of sprung mass displacement, sprung mass acceleration, suspension travel and tyre deflection for active system for three road inputs of sinusoidal bump, step and random road profiles. The peak values of sprung mass acceleration have reduced for all the road profiles, which show the improved ride performance of active suspension system. The peak values of sprung mass displacement and suspension travel have also reduced significantly. As the ride comfort and road holding are mutually contradicting parameters the tyre deflection peak value has reduced only by 29% for random road profile.

## 7. Concluding remarks

The PID controller is designed for active suspension system. A quarter car vehicle model with two-degrees-of-freedom has been modeled. Hydraulic dynamics is also considered while simulated. Ziegler-Nichols tuning rules are used to determine proportional gain, reset rate and derivative time of PID controllers. The system is developed for bumpy road, pothole and random road inputs. The simulated results prove that, active suspension system with PID control improves ride comfort. At the same time, it needs only less rattle space. However, there is no significant improvement in road holding ability observed especially for random road surface. Besides its relative simplicity in design and the availability of well-known standard hardware, the viability of PID controller as an effective tool in developing active suspension system has been proved.

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First placed on the market in 1939, the design of PID controllers remains a challenging area that requires new approaches to solving PID tuning problems while capturing the effects of noise and process variations. The augmented complexity of modern applications concerning areas like automotive applications, microsystems technology, pneumatic mechanisms, dc motors, industry processes, require controllers that incorporate into their design important characteristics of the systems. These characteristics include but are not limited to: model uncertainties, system's nonlinearities, time delays, disturbance rejection requirements and performance criteria. The scope of this book is to propose different PID controllers designs for numerous modern technology applications in order to cover the needs of an audience including researchers, scholars and professionals who are interested in advances in PID controllers and related topics.