RIDE PERFORMANCE ANALYSIS OF HALF CAR VEHICLE DYNAMIC SYSTEM SUBJECTED TO DIFFERENT ROAD PROFILES WITH WHEEL BASE DELAY AND NONLINEAR PARAMETERS

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ABSTRACT

The purpose of a vehicle suspension system is to improve ride comfort and road handling. Vehicle dynamic analysis has been a hot research topic due to its important role in ride comfort, vehicle safety and overall vehicle performance. For proper designing of suspension system, nonlinearities in suspension system parameters must be considered.In this paper, nonlinearities of spring and damper are considered while preparing half car model.The current article is simulated and analyzed the handling and ride performance of a vehicle with passive suspension system Half Car Model with Four Degree of Freedom,Since, the equations of the system cannot be solved mathematically has developed a scheme in MATLAB/Simulink that allows analyzing the behavior of the suspension system for different road profiles.

Keywords: 4 DoF, Half Car Model, MATLAB /Simulink, Passive Suspension System.

I. INTRODUCTION

The vibration of vehicle and seat leads to fatigue of driver and decreases driver safety and operation stability of vehicle. Hence developing improved suspension system to achieve high ride quality is one of the important ride challenges in automotive industry. Therefore the goal of vehicle suspension systems is to decrease the acceleration of car body as well as the passenger seat. In reality, some of the vehicle parameters are with uncertainties, so that it is an important issue to deal with vehicle suspension subjected touncertain parameters in engineering application [1].

The vehicle suspension system differs depending on the manufacturer which ensures a wide range of models. Whichever solution is adapted to design, a suspension system has the primary role to ensuring the safety function. It is known that road unevenness produce oscillations of the vehicle wheels which will transmitted to their axles. It becomes clear that the role of the suspension system witch connect the axles to the car body is to reduce as much vibrations and shocks occurring in the operation. This causes the necessity to use a suspension of a better quality. A quality suspension must achieve a good behavior of the vehicle and a degree of comfort depending on the interaction with uneven road surface [2, 3].

When the vehicle is requested by uneven road profile, it should not be too large oscillations, and if this occurs, they must be removed as quickly. The design of a vehicle suspension is an issue that requires a series of calculations based on the purpose. Suspension systems are classified in the well-known terms of passive, semiactive, active and various in between systems[4]. Passive systems are the most common. Suspension is subjected to various road conditions like a single step road profile, brake and release maneuver, sinusoidal road profile with pitching, heaving and mixed model excitation, broad band road profile etc. at constant or variable speed[5]. The measurement of road surface qualities is one of the important opportunities of vehicle manufactures all over the world. The operations of the measuring devices depend mainly on the use of displacement transducers [6]. This paper deals with the analysis of the vibrational effect when the vehicle is subjected to different road profiles. For this purpose half car vehicle model with linear and nonlinear parameters is developed. For analysis Hyundai Elantra 1992half car model with front suspension is experimented for different road profiles.

II. THE MATHEMATICAL MODEL OF HALF CAR SUSPENSION SYSTEM

The system shown in Fig.1 is half car system where m_s- Sprung Mass, m_{uf}- Front Unsprung Mass, m_{uf}- Rear Unstrung Mass, k_f- Front Suspension Stiffness, k_t- Rear Suspension Stiffness, c_f- Front Suspension Damping Coefficient, c_t-Rear Suspension Damping Coefficient k_t -Tyre (195/65R15) Stiffness, c_t-Tyre (195/65R15) Damping Coefficient, r- Radius of Gyration, b-Wheel Base.The model comprises of sprung mass and two unsprung masses. The x_s denotes the vertical displacement (bounce) at the center of gravity and θ is the pitch of the sprung mass. The front and rear displacements of the unsprung masses are denoted by $x_{uf}$and x_{ur}. In this model, the disturbances $q_f$and q_r are caused by road irregularities.

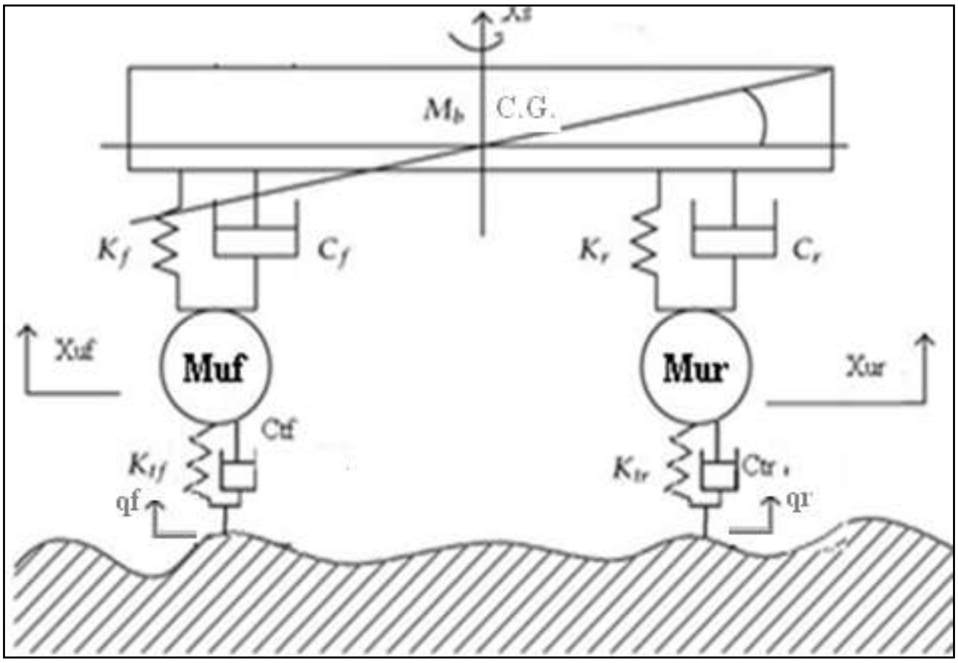


Fig.1: 4DoF Half Car Vehicle Model

The equations of motion can be obtained using the Newton's second law for each of the two masses in motion. These will be:

$$m_s \ddot{x}_s + k_f(x_s - l_f\theta - x_{uf}) + c_f(\dot{x}_s + l_f\dot{\theta} - \dot{x}_{uf}) + k_r(x_s + l_r\theta - x_{ur}) + c_r(\dot{x}_s + l_r\dot{\theta} - \dot{x}_{ur}) = 0 \quad \text{..... (1)}$$

$$I\ddot{\theta} - k_f(x_s - l_f\theta - x_{uf})l_f - c_f(\dot{x}_s - l_f\dot{\theta} - \dot{x}_{uf})l_f + k_r(x_s + l_r\theta - x_{ur})l_r$$
$$+ c_r(\dot{x}_s - l_r\dot{\theta} - \dot{x}_{ur})l_r = 0 \quad \text{...... (2)}$$

$$m_{uf}\ddot{x}_{uf} + k_{tf}(x_{uf} - q_f) + c_{tf}(\dot{x}_{ur} - \dot{q}_f) + k_f(-x_s + l_f\theta + x_{uf}) + c_f(-\dot{x}_s + l_f\dot{\theta} + \dot{x}_{uf}) = 0 \quad \text{.... (3)}$$

$$m_{ur}\ddot{x}_{ur} + k_{tr}(x_{ur} - q_r) + c_{tr}(\dot{x}_{ur} - \dot{q}_r) + k_r(-x_s - l_r\theta + x_{ur}) + c_r(-\dot{x}_s - l_r\dot{\theta} + \dot{x}_{ur}) = 0 \quad \text{...... (4)}$$

Where,

$q_f(t) = q_r(t+\tau_w)$

and, τ_w = Time Delay = $\frac{\text{wheel base (b)}}{\text{vehicle velocity(v)}} = \frac{2.5\times 3600}{1000\times 40} = 0.225$

By introducing nonlinearities in mass, suspension spring, damper and tyre, the equations of motion for body bouncing motion for nonlinear passive model are,

$$f_{Is} = -f_{sf} - f_{df} - f_{sr} - f_{dr} - m_{sg}$$

$$t_{Is} = t_{sf} + t_{df} - t_{sr} - t_{dr}$$

$$f_{Iuf} = -f_{stf} - f_{dtf} + f_{sf} + f_{df}$$

$$f_{Iur} = -f_{str} - f_{dtr} + f_{sr} + f_{dr}$$

In order to analyze the behavior of the half car suspension systemmodel is simulated in MATLAB/Simulink. The input parameters are as follows,

Table 1: Suspension Parameters for Hyundai Elantra 1992 Half Car Model

Sprung Mass (m_s)	515.45 Kg
Front Unsprung Mass (m_{uf})	23.61 Kg
Rear Unsprung Mass (m_{uf})	28 Kg
Front Suspension Stiffness (k_f)	12.394 kN/m
Rear Suspension Stiffness (k_r)	14.662 N/m
Front Suspension Damping Coefficient (c_f)	1.3854 kN-sec/m
Rear Suspension Damping Coefficient (c_r)	1.6352 kN-Sec/m
Tyre (195/65R15) Stiffness (k_t)	181.81888 kN/m
Tyre (195/65R15) Damping Coefficient (c_t)	0.0138 kN-sec/m
Radius of Gyration (r)	1.55 m
Wheel Base (b)	M

Vehicle is assumed to be traveling over a road with velocity of 40km/hr, during this travel the excitation frequency is calculated as

f = $\frac{2\pi V}{\lambda} = \frac{2\times\pi\times 40\times 1000}{6\times 3600}$ = 11.6rad/sec =1.85Hz

III. NON-LINEARITY IN SUSPENSION SPRING

The non-linear effects included in the spring force *f*sare due to two parts. One is bump stop which restricts the wheel travel within the given range and prevents the tire from contacting the vehicle body. And the other is strut bushing which connects the strut with the body structure and reduces the harshness from the road input. This non-linear effect can be included in spring force *f*swith non-linear characteristic versus suspension rattle space from the measured dataon SPMD (Suspension Parameter Measurement Device) shown in Fig.2.

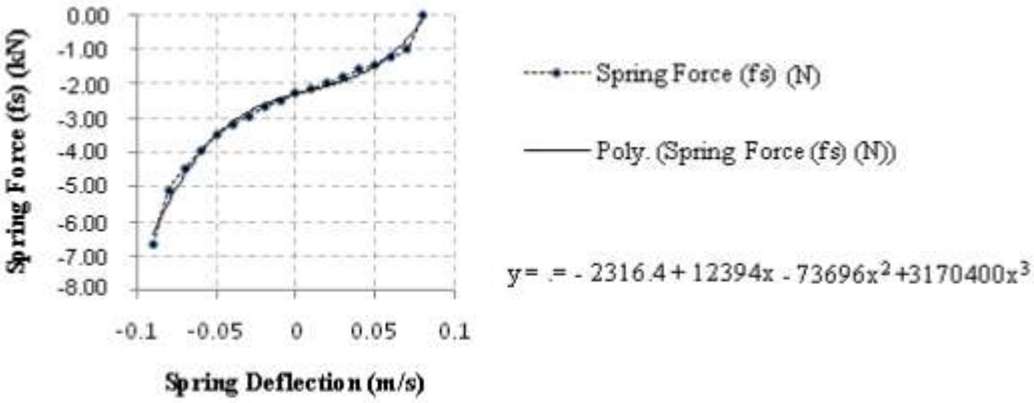


Fig2. Non-linear Spring Force Property of Hyundai Elantra 1992 Model Suspension Spring

The spring force *f*sis modeled as third order polynomial function.

$$f_s = k_0 + k_1 \Delta x + k_2 \Delta x^2 + k_3 \Delta x^3 \qquad \text{.......................... (5)}$$

Where the co-efficient are k_0 = -2.31 kN, k_1 = 12.39 kN/m, k_2 = -73.69 kN/m^2 and k_3 = 3170.40 kN/m^3 (The SPMD data from the 1992 Model Hyundai Elantra front suspension were used).

IV. NON-LINEARITY IN SUSPENSION DAMPER

Generally, the damping force is asymmetric with respect to speed of the rattle space:Damping force during bump is bigger than that during rebound in order to reduce the harshness from the road during bump while dissipating sufficient energy of oscillation during rebound at the same time. Measured data for the damping force versus relative velocity of upper and lower struts, shows the asymmetric property which is shown in Fig.3

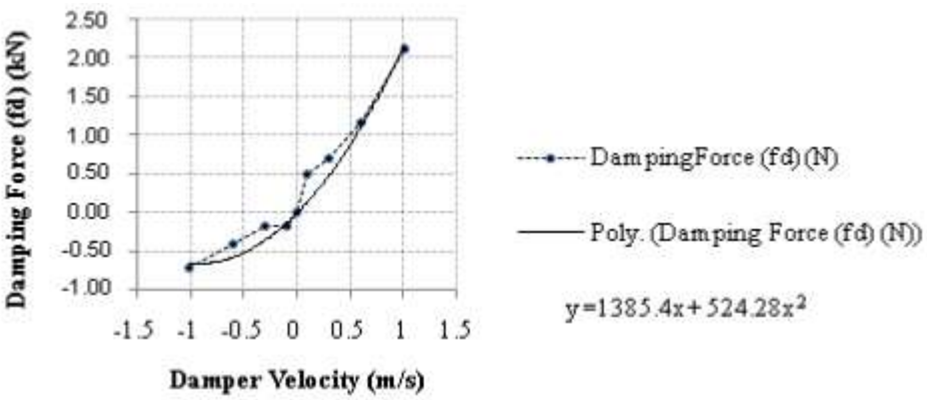


Fig.3.Non-linear Damping Force Property of Hyundai Elantra 1992 Model Suspension Damper

From the measured data the damping force f_d is modeled as second order polynomial function. The mathematical model for this is given by,

$$f_d = c_1 \Delta \dot{x} + c_2 \Delta \dot{x}^2 \quad \text{……………………. (6)}$$

Where the coefficient are obtained from fitting the experimental data, which resulted in

c_1 = 1385.4 N-s/m,

c_2 = 524.28 N-s/mand (The SPMD data from the 1992 model Hyundai Elantra front suspension were used).

V. MATLAB ANALYSIS

The MATLAB/Simulink model is prepared and the sprung mass displacement and pitch for different road profiles were obtained in time domain.The sprung massdisplacement and pitch for linear and non-linear passive suspension system for different road profiles are as below,

5.1 Bumpy (sinusoidal) Road Input

A single bump road input,y is described by (Jung-Shan Lin 1997),is used to simulate the road to verify the developed control system.The road input describedby Eq. (7)is shown in Fig.4.

y =a(1-coswt) for,0.4<t<0.9 ……….(7)

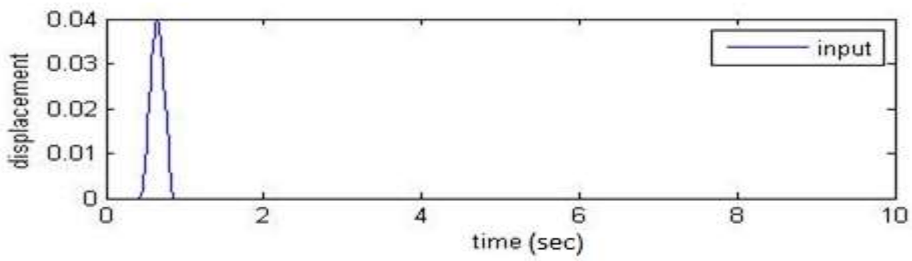


Fig.4Bumpy Road Input

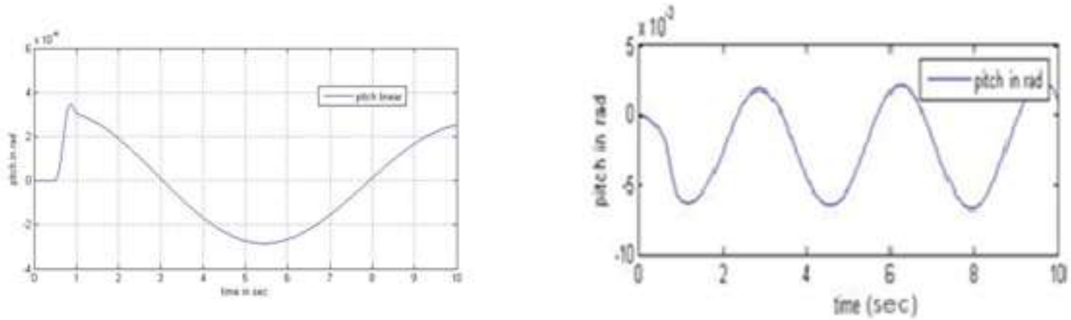


Fig.5. Pitch of Sprung Mass for Linear and Non-Linear Half Car Model

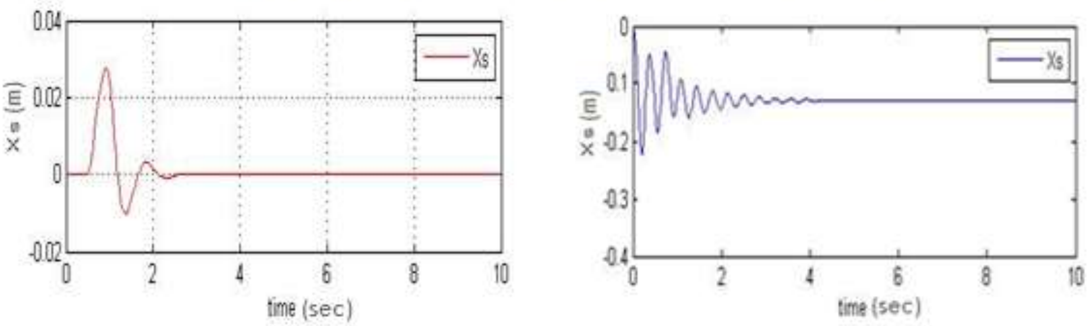


Fig.6. Sprung Mass Displacement (Xs) for Linear and Non-Linear Half Car Model

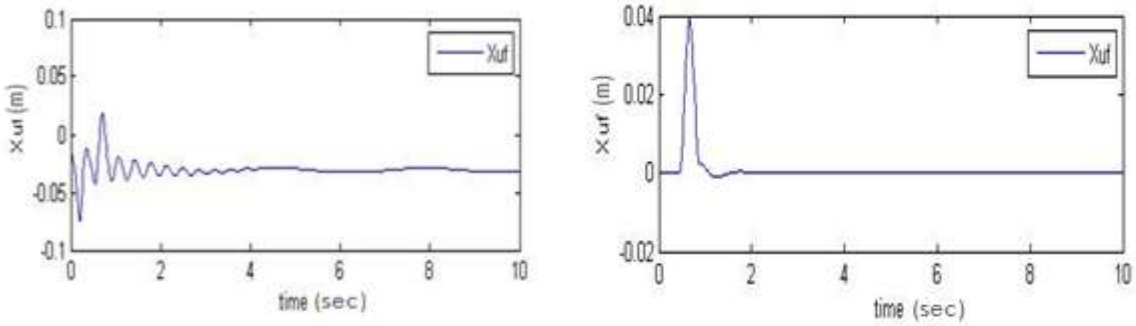


Fig.7. Unsprung Mass Displacement (Xuf) for Linear and Non-Linear Half Car Model

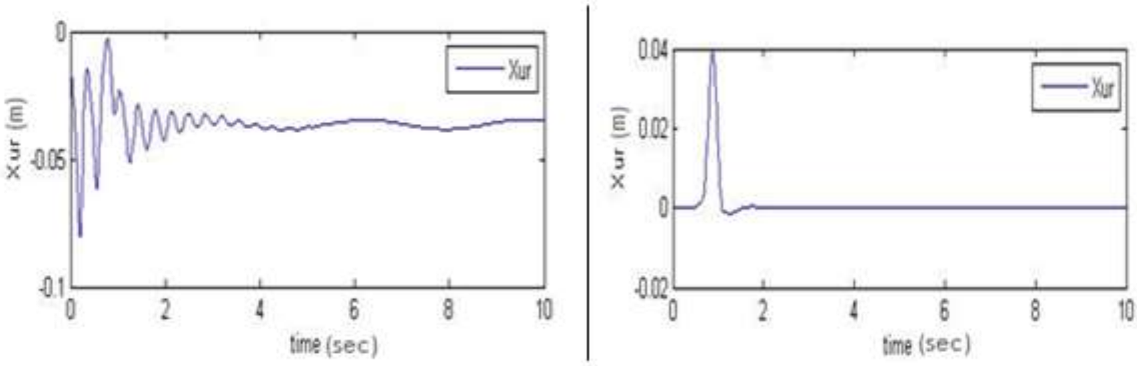


Fig.8. Unsprung Mass Displacement (Xur) for Linear and Non-Linear Half Car Model

5.2 Rectangular Input

The Rectangular Pulse is represented by equation (8)

y = -0.04 for 0.42 <t <0.88,

0.00 for otherwise ....(8)

Here also the height of the road disturbance is maintained at 4 cm.The road input is described by Eq. (8) is shown in

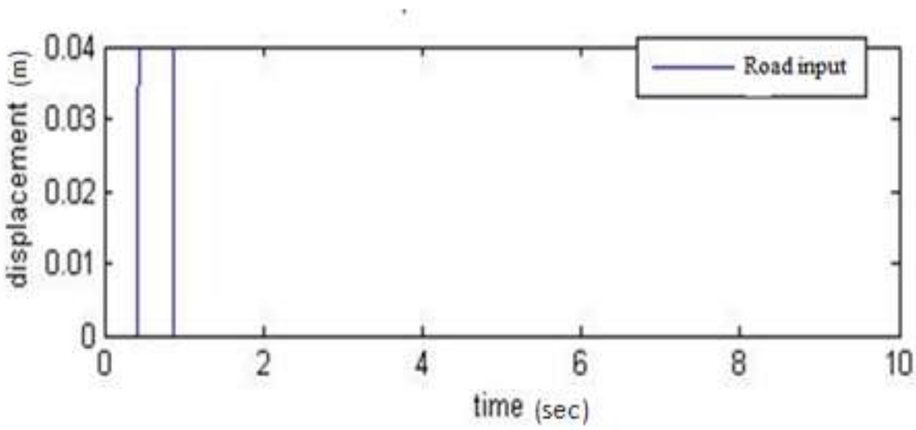


Fig.9.

Fig. 9.Rectangular Input

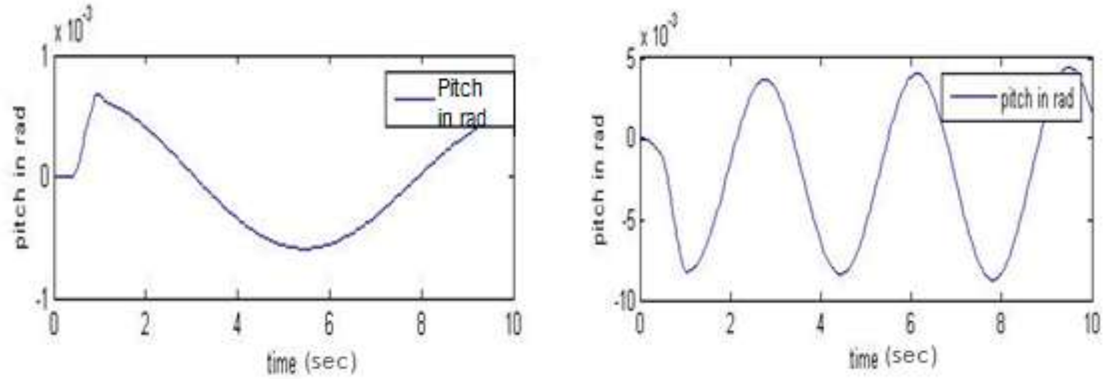


Fig.10. Pitch of Sprung Mass for Linear and Non-Linear Half Car Model

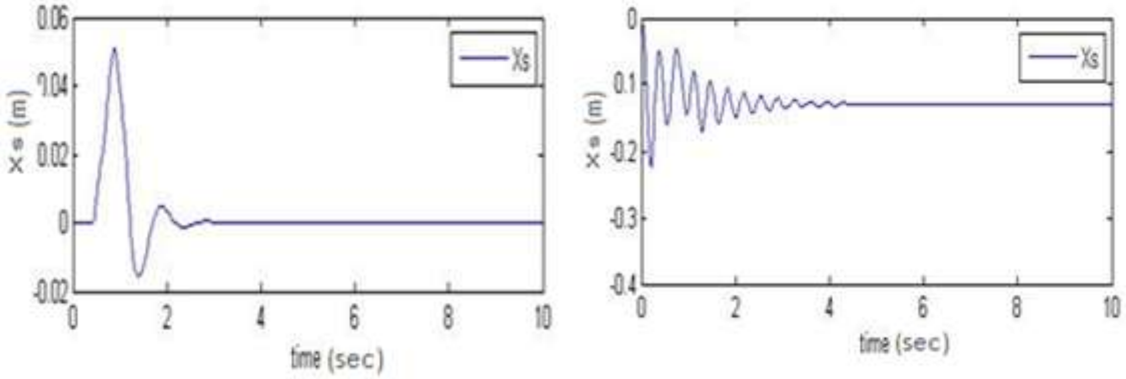


Fig.11.Sprung Mass Displacement (Xs) for Linear and Non-Linear Half Car Model

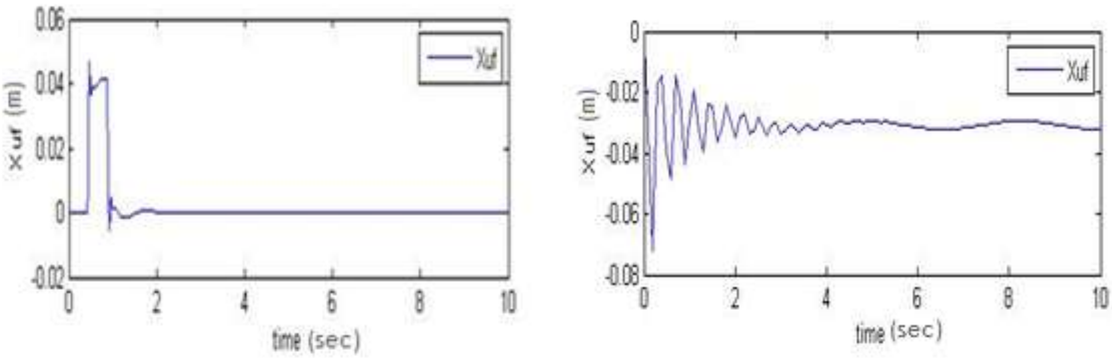


Fig.12.Unsprung Mass Displacement (Xuf) for Linear and Non-Linear Half Car Model

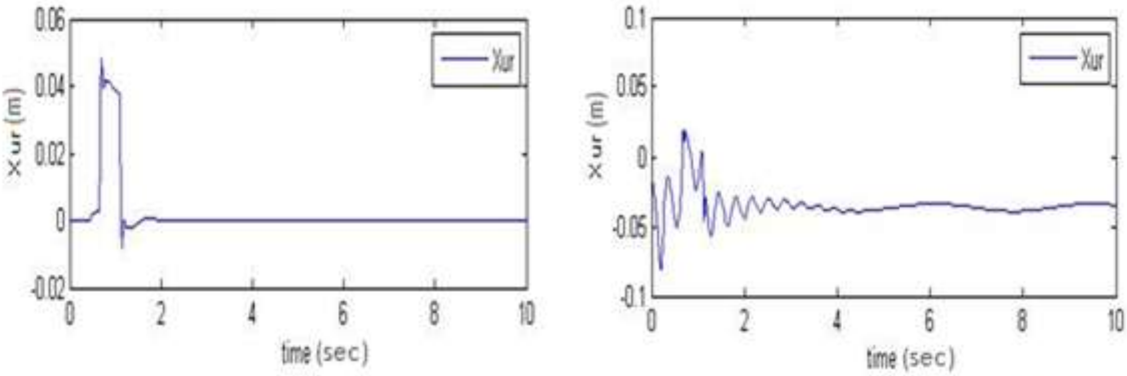


Fig.13.Unsprung Mass Displacement (Xur) for Linear and Non-Linear Half Car Model

5.3. Step Input

The Step excitation is represented by equation (9)

y = -0.04 for 0.42 <t <0.88,

0.00 for otherwise ……………(9)

The road input is described by Eq. (9) is shown in Fig.14.

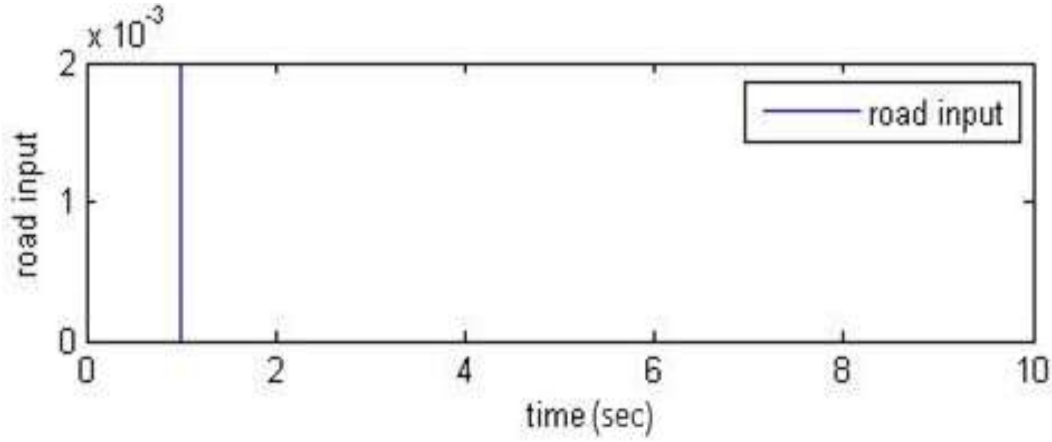
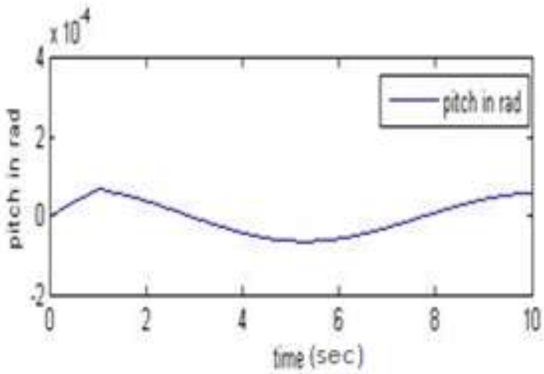


Fig.14.Step Input



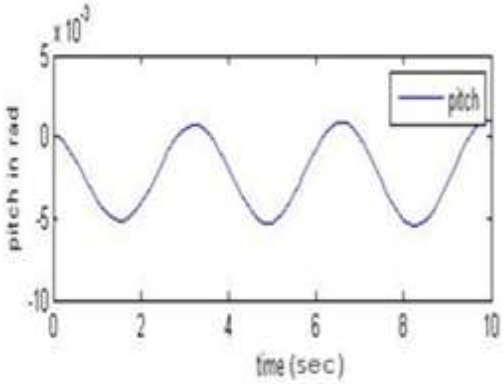
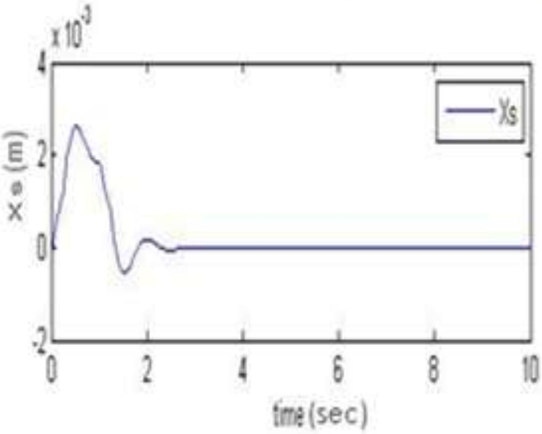


Fig.15.Pitch of Sprung Mass for Linear and Non-Linear Half Car Model



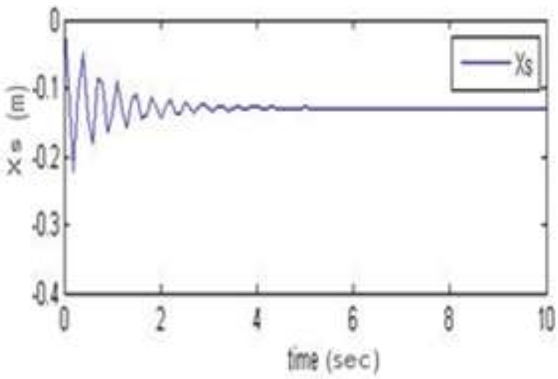


Fig.16.Sprung Mass Displacement (Xs) for Linear and Non-Linear Half Car Model

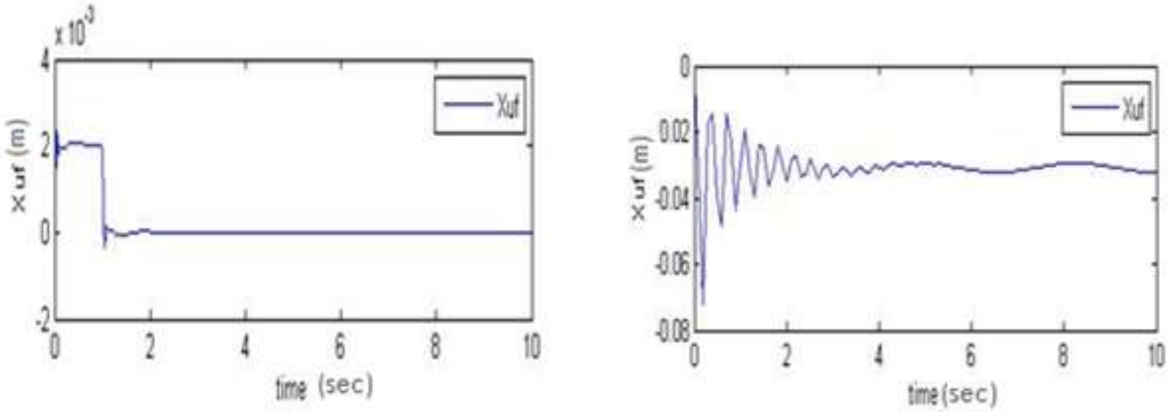


Fig.17.Unsprung Mass Displacement (Xuf) for Linear and Non-Linear Half Car Model

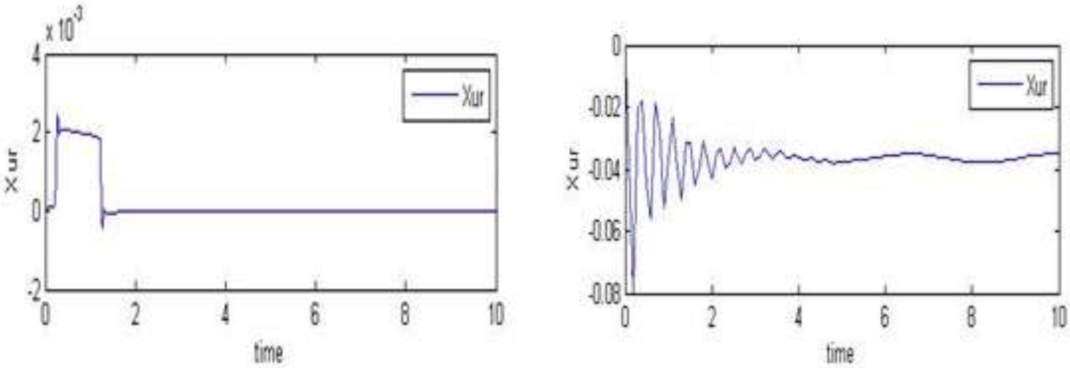


Fig.18.Unsprung Mass Displacement (Xur) for Linear and Non-Linear Half Car Model

VI. CONCLUSION

From the results obtained from simulation,it is seen that in the analysis of vehicle dynamic system consideration of nonlinear parameters is important.When graph of nonlinear passive suspension system is compared to linear suspension system for different road profiles,it is seen that behavior of nonlinear passive suspension system tends to move towards the actual behavior of the system. Comfort to the passengers can be achieved by choosing properspring and damper of the sprung mass which effects on its natural frequency. For Linear half car modelSprung mass displacement is less than for Non Linear half car model for all types of road profiles under consideration.

VII. ACKNOWLEDGEMENTS

The authors are grateful to Dr. J.J.Magdum College of Engineering, Jaysingpur for supporting this work.

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