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Experimental Verification of Passive Quarter Car Vehicle Dynamic System Subjected to Harmonic Road Excitation with Nonlinear Parameters.

Prof. S. P. Chavan[1], Prof. S. H. Sawant[2], Dr. J. A. Tamboli[3]
[1]*(Asst.Professor, Mechanical Engineering Department, Dr. J.J. Magdum College of Engg. Jaysingpur, India)*
[2]*(Asso.Professor, Mechanical Engineering Department, Dr. J.J. Magdum College of Engg. Jaysingpur, India)*
[3]*(Principal, NanashebMahadik College of Engg. Peth-Naka,,Sangli India)*

ABSTRACT: *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 parameters must be considered. In this paper, nonlinearities of spring and damper are considered while preparing quarter car model. For this a simplified model and experimental set up for the same is developed. The deterministic impulses due to road profile are given by harmonic shaker which gives input motion to shock absorber. The sprung mass acceleration response obtained by FFT analyzer at sprung mass of quarter car model is compared with the results obtained by linear and nonlinear MATLAB Simulink models.*

KEYWORDS:*Passive Suspension System, Deterministic, Quarter Car Model, Nonlinear, MATLAB Simulink.*

I. INTRODUCTION

Suspension system is one of the important part of the vehicle. Therefore, it is quite necessary to design finer suspension system in order to improve the quality of vehicles. Since the disturbances from the road may include uncomfortable shake and noise in the vehicle body, it is important to study the vibrations of the vehicle [1]. An automobile is a nonlinear system in practical terms because it consists of suspensions, tires and other components having nonlinear properties. Therefore, the chaotic response may appear as the vehicle moves over a road [2]. 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[3]. 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 [4].

This paper deals with the analysis of the vibrational effect when the vehicle is subjected to harmonic road excitation by the road profile. For this purpose quarter car vehicle model with linear and nonlinear parameters is developed. For analysis Hyundai Elantra 1992, suspension quarter car model with front suspension is experimented for sinusoidal road profile. For this quarter car experimental model, road surface excitations are provided with an electrodynamic shaker and the acceleration responses are obtained with FFT analyzer. These experimental results are compared with the MATLAB Simulink Simulation results.

II.ROAD PROFILE AND WHEEL TRAVEL

Road is considered as an infinite cam with wavy profile of harmonic waves and wheel of quarter car model is considered as follower. The road is considered as cam which will give harmonic road excitation to suspension system. The road profile is approximated by a sine wave as shown in Fig.1 represented by the equation

$$q = Y \sin \omega t$$

Where,

q = Road surface excitation at time t (m)
Y= Amplitude of sine wave = 0.02 m.
λ = Wavelength of road surface = 6 m.

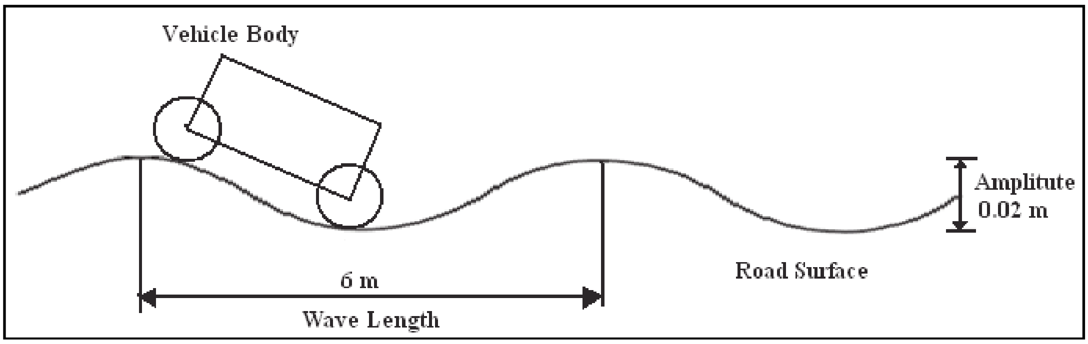

Fig. 1 : Harmonic Road Surface Profile.

Vehicle is assumed to be traveling over a road with velocity of 10 km/hr to 120 km/hr, during this travel the excitation frequency is calculated as

$$\boldsymbol{f = \frac{2\pi V}{\lambda}}$$

For 10 km/hr f is calculated as

$$f = \frac{2\pi x\ 10\ x\ 1000}{6\ \ x\ 3600} = 2.90\frac{\text{rad}}{\text{sec}} = 0.46\ \text{Hz}$$

Similarly other values of excitation frequencies are calculated upto 120 km/hr i.e. 5.55 Hz

III. MODELING OF SYSTEM

To analyze the effect of these non-linearities on vehicle dynamic system in this passive quarter car model shown in Fig. 2 is taken for study.

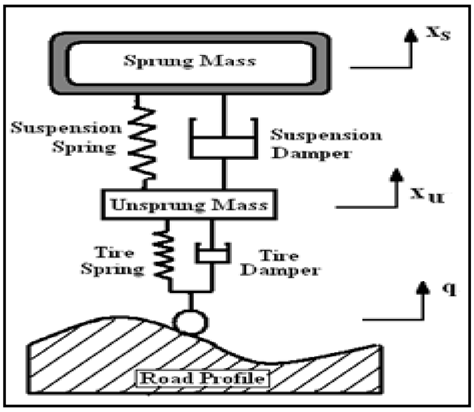

Fig. 2 :TDOF Passive Quarter Car Model.

The parameters of Hyundai Elantra 1992 model considered for the analyses are given in Table1.

Table1: Suspension Parameters for Hyundai Elantra 1992 Model

Sprung Mass (m_s)	236.12 Kg
Unsprung Mass (m_u)	23.61 Kg
Suspension Stiffness (k_s)	12394 N/m
Passive Suspension Damping Coefficient (c_s)	1385.4 N-sec/m
Tire Stiffness (195/65R15 Tire) (k_t)	181818.88 N/m

Tire Damping Coefficient (195/65R15 Tire) (c_t)	13.854 N-sec/m

The equations of motion for this linear model is

$$\ddot{x}_S = -\frac{1}{m_S}\left[\left[k_S(x_S - x_u) + c_S(\dot{x}_S - \dot{x}_u)\right]\right] \quad (1)$$

$$\ddot{x}_u = -\frac{1}{(m_u)}\left\{-\left[k_S(x_S - x_u) + c_S(\dot{x}_S - \dot{x}_u)\right] - \left[k_t(q - x_u) + c_t(\dot{q} - \dot{x}_u)\right]\right\} \quad (2)$$

IV. SUSPENSION NONLINEARITIES

The non-linear effects included in the spring force f_s are 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_s with non-linear characteristic versus suspension rattle space $(x_s - x_u)$ from the measured data (SPMD : Suspension Parameter Measurement Device) shown in Fig.3.

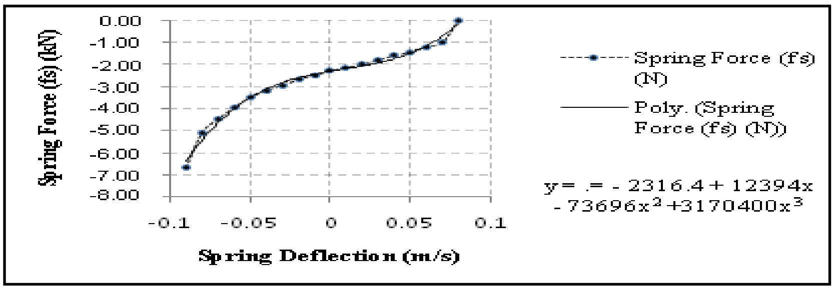

Fig.3: Non-linear Spring Force Property of Hyundai Elantra 1992 Model Suspension Spring

The spring force f_s is modeled as third order polynomial function [6]

$$f_s = k_0 + k_1\Delta x + k_2\Delta x^2 + k_3\Delta x^3 \quad (3)$$

Where the co-efficients are obtained from fitting the experimental data, which resulted in k_{f3} = 3170400 N/m^3, k_{f2} = -73696N/m^2, k_{f1} = 12394N/m , k_{f0} = -2316.4 N (The SPMD data from the 1992 model Hyundai Elantra front suspension were used) Generally, the damping force is asymmetric with respect to the 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. Fig.4. shows the measured data for the damping force versus relative velocity of upper and lower struts, which shows the asymmetric property.

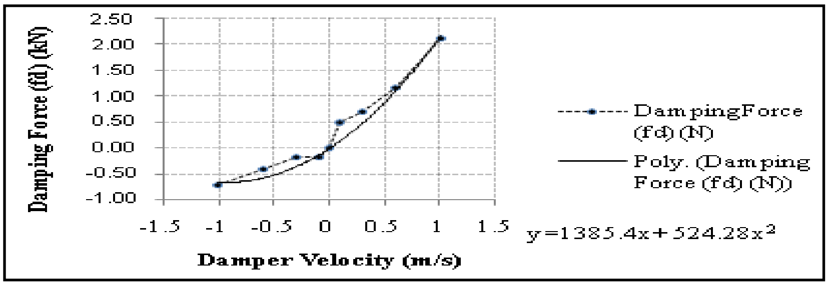

Fig.4: Non-linear Damping Force Property of Hyundai Elantra 1992 Model Suspension Damper

From the measured data the damping force f_d is modeled as second order polynomial function [6] as,

$$f_d = c_1 \Delta \dot{x} + c_2 \Delta \dot{x}^2 \quad (4)$$

Where the co-efficients are obtained from fitting the experimental data, which resulted inc_{f2} = 524.28 N-s^2/m^2 and c_{f1} = 1385.4 N-s/m (The SPMD data from the 1992 model Hyundai Elantra front suspension were used).

For combined non-linearity in suspension spring and damper the equations of motion are

$$\ddot{x}_s = -\frac{1}{(m_s)}\left[k_0 + k_1(x_s - x_u) + k_2(x_s - x_u)^2 + k_3(x_s - x_u)^3 + c_1(\dot{x}_s - \dot{x}_u) + c_2(\dot{x}_s - \dot{x}_u)^2\right] \quad (5)$$

$$\ddot{x}_u = -\frac{1}{(m_u)}\{-\left[k_0 + k_1(x_s - x_u) + k_2(x_s - x_u)^2 + k_3(x_s - x_u)^3 + c_1(\dot{x}_s - \dot{x}_u) + c_2(\dot{x}_s - \dot{x}_u)^2\right]$$
$$- k_t(q - x_u) - c_t(\dot{q} - \dot{x}_u)\} \quad (6)$$

V. MATLAB ANALYSIS

The MATLAB Simulink model is prepared and the sprung mass acceleration for different excitation frequencies were obtained in time domain and from these results values of sprung mass acceleration responses were obtained at different excitation frequencies.

The sprung mass acceleration for linear passive and non-linear suspension spring and damper passive suspension system for Harmonic road excitation (q) are as shown in Fig. 5.

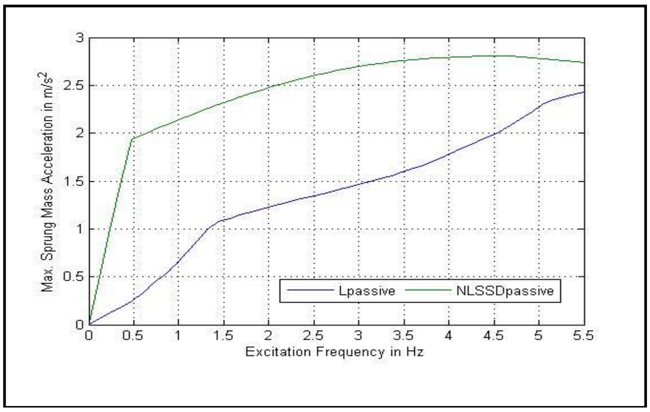

Fig. 5: Sprung Mass Acceleration for Linear and Combined Nonlinear Suspension Spring and Damper Quarter Car Model.

The results are tabulated in Table 2.

Table 2: Max. Sprung Mass Acceleration for Linear Passive and Non-linear Spring and Damper Passive Controller for Different Excitation Frequencies

Excitation Frequency in Hz	Max Sprung Mass Acceleration for Linear Passive Controller in m/s^2	Max Sprung Mass Acceleration for Nonlinear Passive Controller in m/s^2
0	0	0
0.46	0.23	1.92
0.92	0.58	2.11

1.38	1.06	2.28
1.85	1.19	2.43
2.31	1.30	2.56
2.77	1.41	2.66
3.23	1.52	2.73
3.70	1.66	2.78
4.16	1.85	2.80
4.62	2.04	2.81
5.09	2.23	2.77
5.55	2.44	2.73

VI. EXPERIMENTAL ANALYSIS

Fig. 6 Shows Schematic Diagram of Experimental Setup, shown in Fig.7.

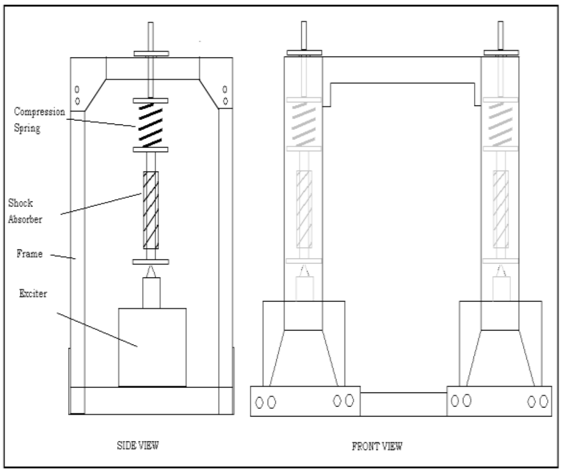

Fig 6: Schematic Diagram of Experimental Setup.

Fig.7. Experimental Setup

The excitation to shock absorber is given with harmonic exciter and the results were obtained at sprung mass of Quarter car model by FFT analyzer in frequency domain. The results obtained are as shown in Fig.8.

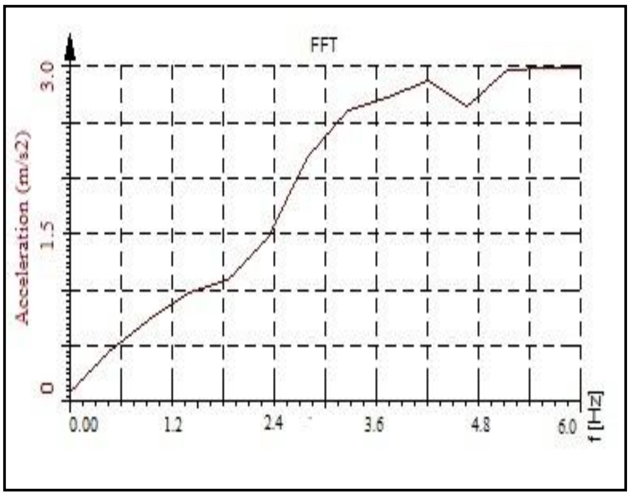

Fig.8: Experimental Results by FFT Analyzer at Different Excitation Frequencies

The results are tabulated in Table 3

Table 3: Max. Sprung Mass Acceleration for Quarter Car Passive Controller with Experimental Analysis and MATLAB Analysis for Different Excitation Frequencies

Excitation Frequency in Hz	Max Sprung Mass Acceleration for Nonlinear Passive Controller in m/s^2(MATLAB Analysis)	Max Sprung Mass Acceleration for Nonlinear Passive Controller in m/s^2(Experimental Analysis)
0	0	0
2.77	2.66	2.25
3.23	2.73	2.42
3.70	2.78	2.70
4.16	2.80	2.82
4.62	2.81	2.76
5.09	2.77	2.90
5.55	2.73	3.00

VII. DISCUSSION OF RESULTS

From the simulation results it is observed that there is considerable difference in sprung mass acceleration of nonlinear suspension spring and damper quarter car model compare with linear suspension spring and damper. And for Experimental results obtained it is observed that the results obtained by theoretical analysis for quarter car passive suspension system are approximately same as that of experimental results.

VIII. CONCLUSION

The simulation result shows considerable difference in linear and non-linear passive sprung mass. It is found that the chaotic response exists in nonlinear suspension. As the results of theoretical (combined nonlinear spring and damper) and experimental analysis of quarter car passive are quite similar because experimental model contains inherent nonlinear properties of suspension parameters, so it is necessary to consider the nonlinearities in suspension system for analysis of dynamic vehicle system. Hence form above we conclude that the analysis of suspension system we can use only theoretical (MATLAB Simulink) models instead of difficult experimental setup.

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