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

Shital.P. Chavan and S.H. Sawant

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 half 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 an eccentric cam which gives input motion to shock absorber acting as follower of the cam. The displacements obtained by FFT analyser at C.G. of half car model were compared with the results obtained by linear and nonlinear MATLAB Simulink models.*

Keywords--- *Suspension System, Deterministic, Half 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 half car vehicle model with linear and nonlinear parameters with wheel base time delay is developed. For analysis Hyundai Elantra 1992, suspension half car model with rear suspension and front suspension is experimented for sinusoidal road profile. For this half car experimental model, road surface excitations are provided with an eccentric cam placed at a distance equal to wheel base of the vehicle and the displacement results are obtained with FFT analyzer. These experimental results are compared with the MATLAB Simulink Simulation results.

II. Road Profile and Wheel Travel

It is considered that vehicle is travelling on the road surface which applies harmonic excitations to its wheels. Therefore road is considered as a cam which provides harmonic excitations whereas the wheels of the half car vehicle dynamic system are considered as the follower. The road profile is approximated by a sine wave represented by $q=Y \sin\omega t$.

Where,

q = Road surface excitation at time t in m.
Y= Amplitude of sine wave = 0.01 m, and
l = Wavelength of road surface = 6 m.

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For the above data and velocity (v) of the vehicle in Km/h, the excitation frequency (ω) is given by,

$$\omega = v\left(\frac{1}{0.006}\right)\left(\frac{1}{3600}\right)(2\pi) = 0.2909\, v \text{ rad/sec.}$$

III. 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.1

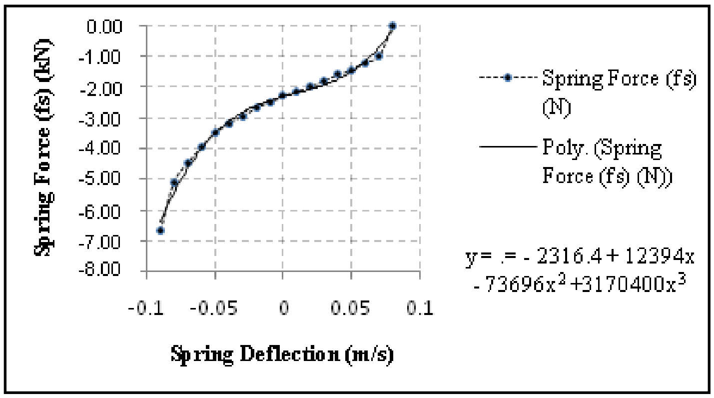

Figure 1: Modeling of Nonlinear Spring Force Wheel Stroke (m) Vs Suspension Force (N)

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

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

Where the co-efficients are obtained from fitting the experimental data, which resulted in k_{f3} = 3170400 N/m^3, k_{f2} = -73696 N/m^2, k_{f1} = 12394 N/m , k_{f0} = -2316.4 N (The SPMD data from the 1992 model Hyundai Elantra front suspension were used) and k_{r3}=3750472 N/m^3 , k_{r2}= -87180 N/m^2, k_{r1}=14662 N/m, k_{r0}=2740.22 N (The SPMD data from the 1992 model Hyundai Elantra rear suspension obtained by reaction method 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. 2 shows the measured data for the damping force versus relative velocity of upper and lower struts, which shows the asymmetric property.

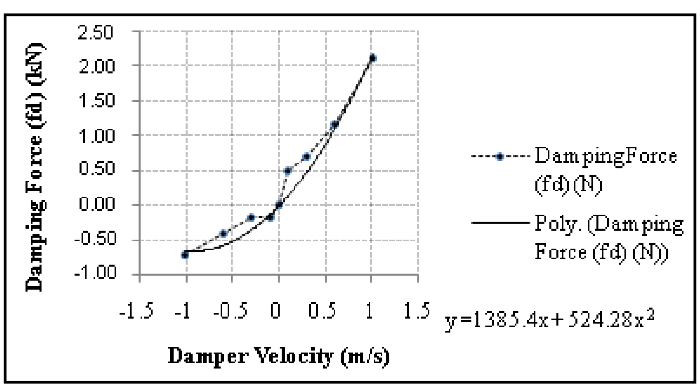

Figure 2: Asymmetric Damping Property of Actual Suspension System [Damper Speed (m/s) Vs Damping Force (N)]

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

$$f_d = c_1 \Delta\dot{x} + c_2 \Delta\dot{x}^2$$

Where the co-efficients are obtained from fitting the experimental data, which resulted in c_{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) and c_{r2}=618.80 N-s^2/m^2 and c_{r1}=1635.18 N-s/m (The SPMD data from the 1992 model Hyundai Elantra rear suspension obtained by reaction method were used).

IV. Modeling of System

For Hyundai Elantra suspension system, half car model is as shown in the Fig. 3.

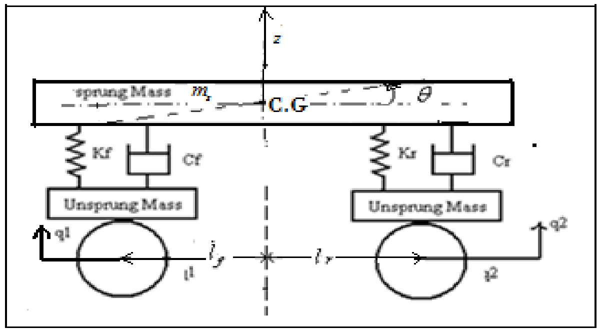

Figure 3: Half Car Model

The equations of motion for this linear model is

$$m_s \ddot{z} + k_f(z - l_f\theta - q) + k_r(z + l_r - q) + c_f(\dot{z} - l_f\dot{\theta} - q) + c_r(\dot{z} + l_r\dot{\theta} - \dot{q}) = 0$$

$$m_s r_s^2 \ddot{\theta} + k_f l_f(z - l_f\theta - q) + k_r l_r(z + l_r\theta - q) + c_f l_f(\dot{z} - l_f\dot{\theta} - \dot{q}) + c_r l_r(\dot{z} + l_r\dot{\theta} - \dot{q}) = 0$$

The equations of motion for this nonlinear model is

$$m_s\ddot{z} + k_{f0} + k_{f1}(z - l_f\theta - q) + k_{f2}(z - l_f\theta - q)^2 + k_{f3}(z - l_f\theta - q)^3 + k_{r0} + k_{r1}(z + l_r\theta - q) +$$

$$k_{r2}(z + l_r\theta - q)^2 + k_{r3}(z + l_r\theta - q)^3 + c_{f1}(\dot{z} - l_f\dot{\theta} - \dot{q}) + c_{f2}(\dot{z} - l_f\dot{\theta} - \dot{q})^2 + c_{r1}(\dot{z} + l_r\dot{\theta} - \dot{q}) +$$

$$c_{r2}(\dot{z} + l_r\dot{\theta} - \dot{q})^2 = 0$$

$$I_y\ddot{\theta} + l_f[k_{f0} + k_{f1}(z - l_f\theta - q) + k_{f2}(z - l_f\theta - q)^2 + k_{f3}(z - l_f\theta - q)^3] + l_r[k_{r1}(z + l_r\theta - q) +$$

$$k_{r2}(z + l_r\theta - q)^2 + k_{r3}(z + l_r\theta - q)^3] + l_f[c_{f1}(\dot{z} - l_f\dot{\theta} - \dot{q}) + c_{f2}(\dot{z} - l_f\dot{\theta} - \dot{q})^2] + l_r[c_{r1}(\dot{z} + l_r\dot{\theta} - \dot{q}) +$$

$$c_{r2}(\dot{z} + l_r\dot{\theta} - \dot{q})^2] = 0$$

Where,

z = Vertical displacement of C.G.
θ = Angular displacement of longitudinal axis about C.G.
l_f= Distance of front wheel from C.G.
l_r = Distance of rear wheel from C.G.
q = Road surface excitation.

The MATLAB simulink model for the same is prepared as shown in Fig.4 and the sprung mass displacement for different excitation frequencies were obtained in time domain and from these results values of amplitude response (X) were obtained at different excitation frequencies.

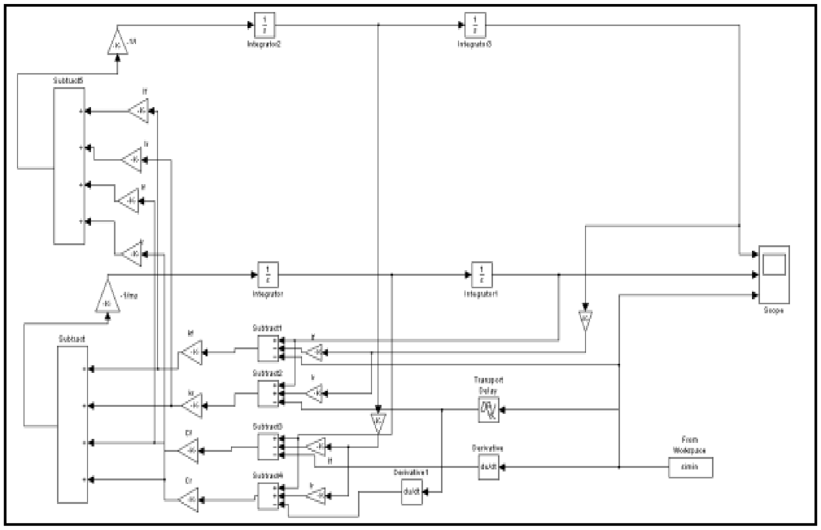

Figure 4: MATLAB Simulink Model for Linear Half Car Two DOF Vehicle Dynamic System

V. MATLAB ANALYSIS

The graphs for theoretical method and MATLAB Simulink model were plotted as shown in Fig.5 and Fig 6.

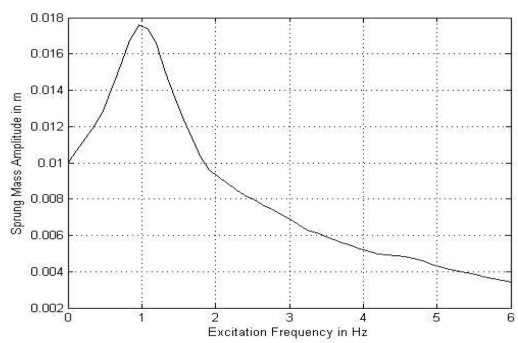

Fig 5: Frequency Response (Bounce) for Different Excitation Frequencies by MATLAB Simulink Model with Linear Parameters

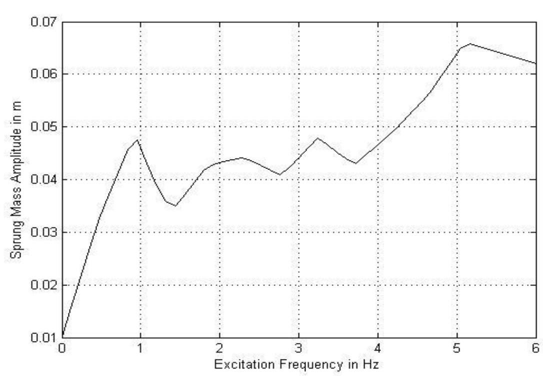

Fig 6: Frequency Response (Bounce) for Different Excitation Frequencies by MATLAB Simulink Model with Nonlinear Parameters

VI. EXPERIMENTAL SETUP

Fig. 7 Shows Experimental Setup.

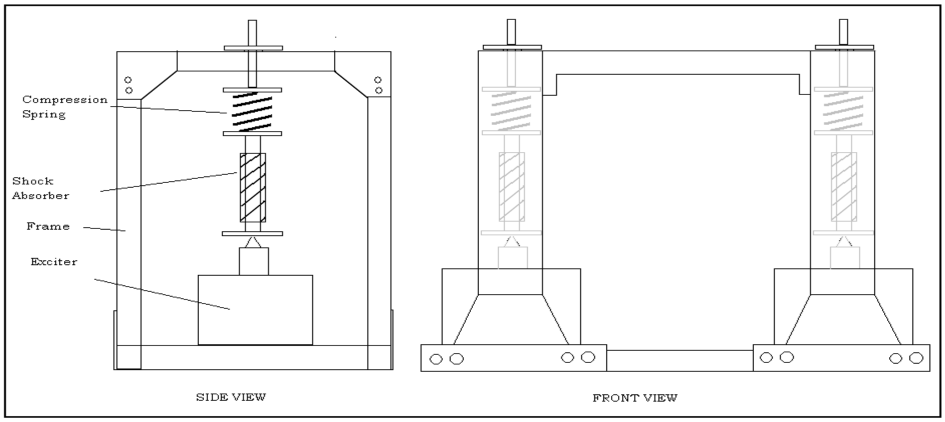

Figure 7: Experimental Setup

VII. EXPERIMENTAL ANALYSIS

The excitation to shock absorbers is given with eccentric cams and the results were obtained at C.G. of Half car model by FFT analyzer in frequency domain. The result obtained is as shown in Fig.8

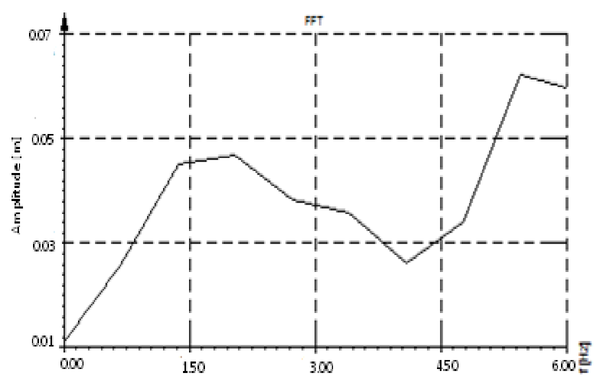

Figure 8: Frequency Response (Sprung Mass Displacement) for Different Excitation Frequencies obtained by Experiment

VIII. RESULTS

The experimental and MATLAB Simulink amplitude response values for different excitation frequencies were tabulated as shown in Table 1.

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Table 1: The Experimental and MATLAB Simulink Amplitude Response Values for Different Excitation Frequencies

Excitation Frequency (f) Hz	Amplitude (X) m [By MATLAB Simulink for linear model]	Amplitude (X) m [By MATLAB Simulink for nonlinear model]	Amplitude (X) m [By Experimental Method]
0	0.01	0.01	0.01
0.4629	0.0127	0.0321	0.024
0.9258	0.0177	0.0489	0.029
1.1528	0.0172	0.0402	0.038
1.3887	0.0141	0.034	0.041
1.8516	0.0098	0.0427	0.043
2.3146	0.0084	0.0443	0.045
2.7775	0.0074	0.0407	0.039
3.2407	0.0063	0.0479	0.035
3.7033	0.0056	0.0428	0.033
4.1662	0.005	0.0488	0.029
4.6291	0.0048	0.056	0.032
5.0920	0.0042	0.066	0.058
5.5550	0.0038	0.064	0.062

IX. Conclusion

The theory of nonlinear dynamics is applied to study the nonlinear model. The simulation result shows considerable difference in linear and non-linear passive controller. It is found that the chaotic response exists in nonlinear suspension. Theoretical nonlinear results and experimentally obtained results show the similar behavior 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.

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