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Experimental Investigation of Half Car Vehicle Dynamic System Subjected to Different Road Profiles with Wheel Base Delay and Nonlinear Parameters

S.S. Patole[1] and Dr. S.H. Sawant[2]

[1]P.G.Student,[2]Professor, Department of Mechanical Engineering, Dr. J.J.M.C.OE., Jaysingpur, India

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Abstract

Vibration of a vehicle is directly influenced by road roughness and suspension system. Apart from the road roughness, proper vehicle suspension system can play vital role to provide the driver and passenger comfort. For proper designing of suspension system, nonlinearities in suspension parameters must be considered. In this paper, ride comfort is analyzed by determining vertical displacement i.e bounce (Xs) and pitch (ϑ) of sprung mass from center of gravity for half car vehicle dynamic system. Experimental Set up is developed to determine bounce (Xs) and pitch (ϑ) of sprung mass by using FFT Analyzer. Experimentation is carried out for different road profiles i.e Bumpy (Sinusoidal) input, Rectangular input and Step input for ride comfort analysis.

Keywords: *Half car vehicle dynamic system, Ride Comfort, Road Profile, FFT Analyzer.*

1. Introduction

The vehicle suspension systems differ 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. 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. [1, 6].

Table 1: Experimental Parameters

Parameter	Symbol	Value (Before Scaling)	Value (After Scaling)
Sprung Mass	m_s	515.45 Kg	51.545 Kg
Front Unsprung Mass	m_{uf}	23.61 Kg	2.361 Kg
Rear Unsprung Mass	m_{ur}	28 Kg	2.8 Kg
Front Suspension Stiffness	k_f	12.394 kN/m	1.2394 kN/m
Rear Suspension Stiffness	k_r	14.662 kN/m	1.4662 kN/m
Front Suspension Damping Coefficient	c_f	1.3854 kN-sec/m	0.13854 kN-sec/m
Rear Suspension Damping Coefficient	c_r	1.6352 kN-Sec/m	0.16352 kN-Sec/m
Tyre (195/65R15) Stiffness	k_t	181.81888 kN/m	18.181888 kN/m
Tyre (195/65R15) Damping Coefficient	c_t	0.0138 kN-sec/m	0.00138 kN-sec/m
Radius of Gyration	r	1.55 m	0.155 m
Wheel Base	b	2.5 m	0.25 m

Shock absorption in automobiles is performed by suspension system that carries the weight of the vehicle while attempting to reduce or eliminate vibrations which may be induced by a variety of sources, such as road surface irregularities, aerodynamic forces, vibrations of the engine and driveline, and non-uniformity of the tire/wheel assembly. Usually, road surface irregularities, ranging from potholes to random variations of the surface elevation profile, acts as a major source that excites the vehicle body through the tire/wheel assembly and the suspension system (Wong,1998).

2. Development of Experimental Setup

In order to determine bounce and pitch of sprung mass of half car, passive suspension system is developed. In order to achieve this rack and pinion is provided in which a rack acts as a road. On rack a profile is generated which will act as input. A rack gives linear motion by pinion mounted on a shaft of Hydraulic Motor. When a rack is in motion, an excitation is created by cam profile act as input to the follower (wheel). Thus, the excitation is transmitted to the sprung mass.

Scaling factor of 1/10 is taken for study. For the purpose of analysis the parameters of Hyundai Elantra 1992 Model are used these parameters are shown in Table 1.

2.1 Cam Profile

In order to carry out experimentation, input in the form of cam is provided as per the dimensions. The cam is generated over the rack which acts as road. In order to provide the linear motion to the rack, rack and pinion mechanism is used. The pinion is mounted on the shaft of Hydraulic motor. For the design of the cam average speed of vehicle is taken as 40 Km/hr. The experimental set up is as shown in Fig.1

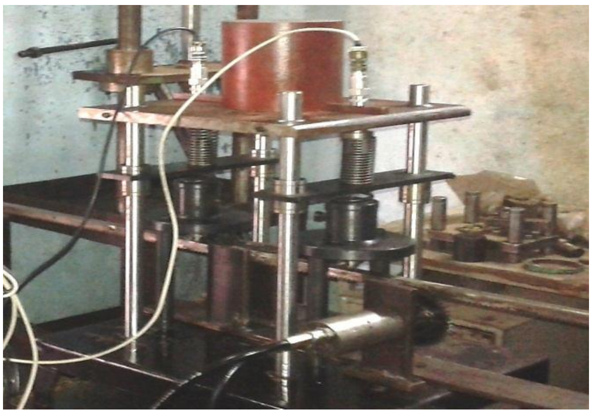

Fig.1: Experimental Set up

3. Experimentation

1. The cam is developed as that of input on the rack which act as road.
2. After assembly of an apparatus in vibration analyzer measurement scheme has to be made, in analyzer proper selection of sensors and their channel is made. Measurement parameters are defined i.e. measurement of bounce and pitch to which accelerometer is connected.
3. All sensors should be attached to vibration analyzer when it is in power off mode. Proper connections of all sensors are made.
4. The accelerometers are attached to sprung mass to measure the bounce and pitch of it.
5. Experimentation is carried out for different road profiles i.e Bumpy (Sinusoidal) Input, Rectangular Input and Step Input.

4. Results

The bounce and pitch of sprung mass for different road profiles of suspension system is determined by FFT Analyzer. The results are shown below.

4.1 Bumpy Input

A single bumpy input, y as described by (Jung-Shan Lin 1997), is used to simulate the road to verify the developed control system. The road input described by Eq. (1) is shown in Fig.2.

$$y = a\,(1 - \cos\omega t) \longrightarrow 0.4 < t < 0.9 \qquad (1)$$

In Eq. (1) of road disturbance, 'a' is set to 0.02 m to achieve a bump height of 4 cm.

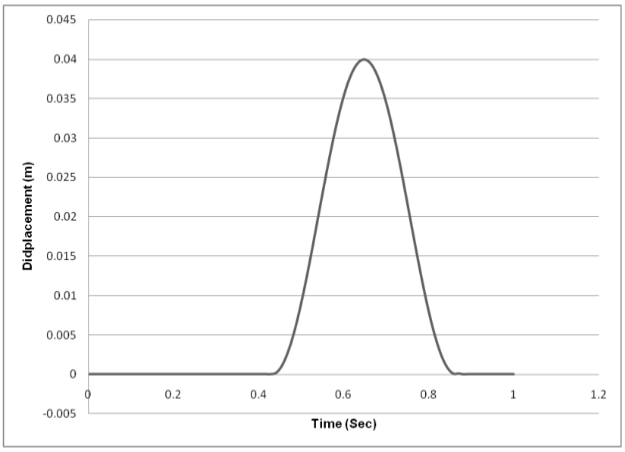

Fig.2: Bumpy Input

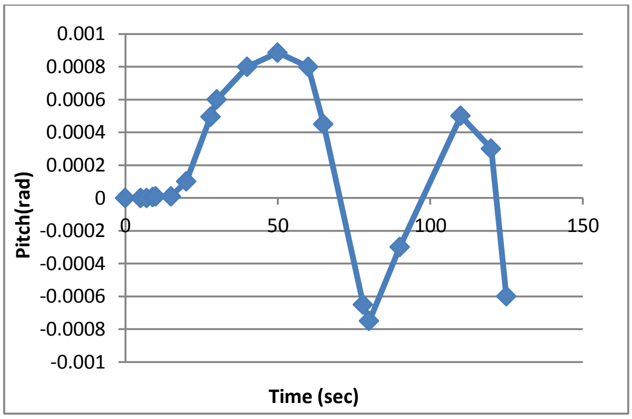

Fig.3: Sprung Mass Pitch (θ)for Half Car System

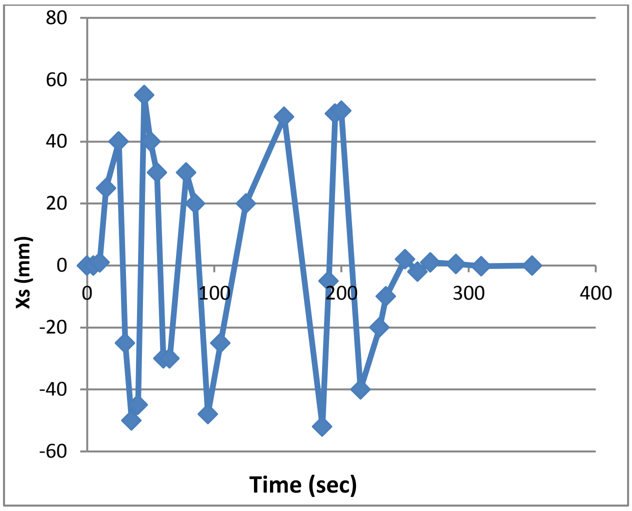

Fig.4: Sprung Mass Displacement (Xs) for Half Car System

4.2 Rectangular Input

The Rectangular Pulse Excitation is represented by the Eq. (2),

$$y = \begin{cases} 0.00 \longrightarrow 0.43 > t > 0.86, \\ 0.04 \longrightarrow 0.43 \leq t \leq 0.86 \end{cases}$$

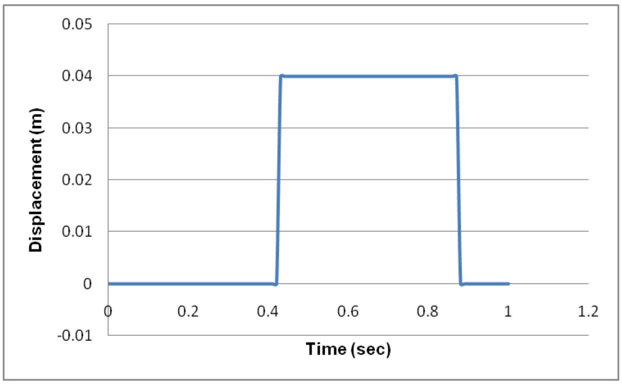

Fig.5: Rectangular Input

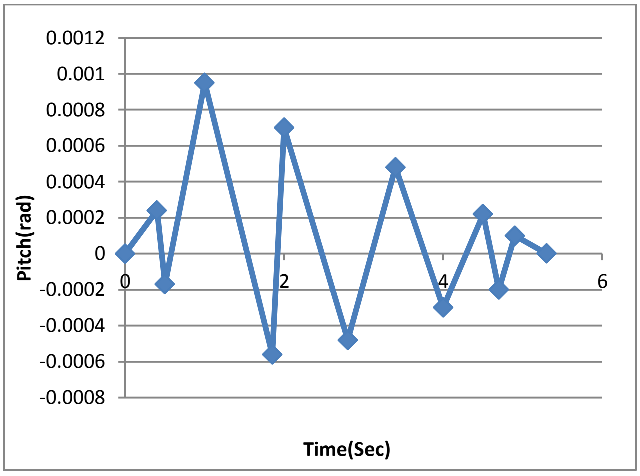

Fig. 6: Sprung Mass Pitch (θ) for Half Car System

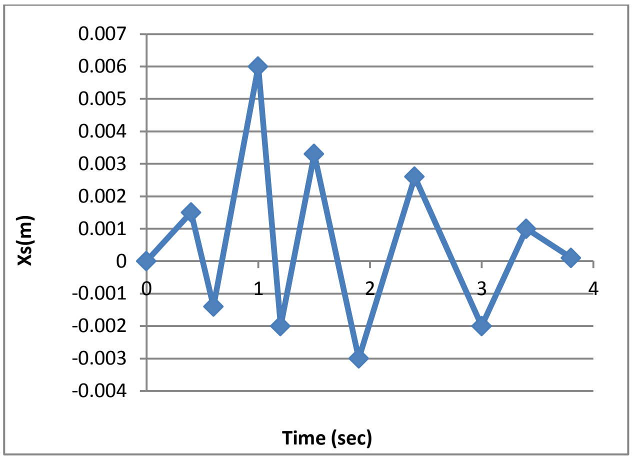

Fig.7: Sprung Mass Displacement (Xs) for Half Car System

4.3. Step Input

The Step in Excitation is represented by the Eq. (3),

$$y = \begin{cases} 0.00 \longrightarrow t < 0.43, \\ 0.04 \longrightarrow t \geq 0.43 \end{cases} \qquad (3)$$

Here the height of the road disturbance is maintained at 4 cm. The road input described by Eq. (3) is shown in Fig.8.

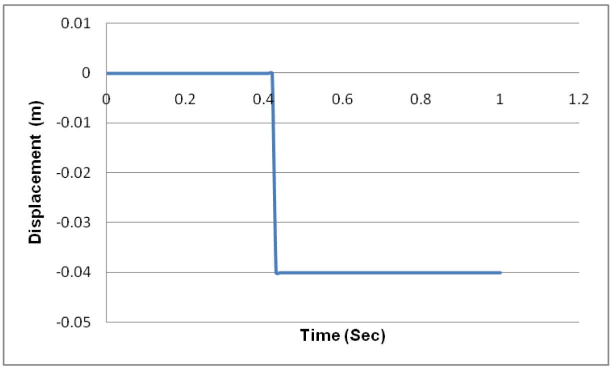

Fig.8: Step Input

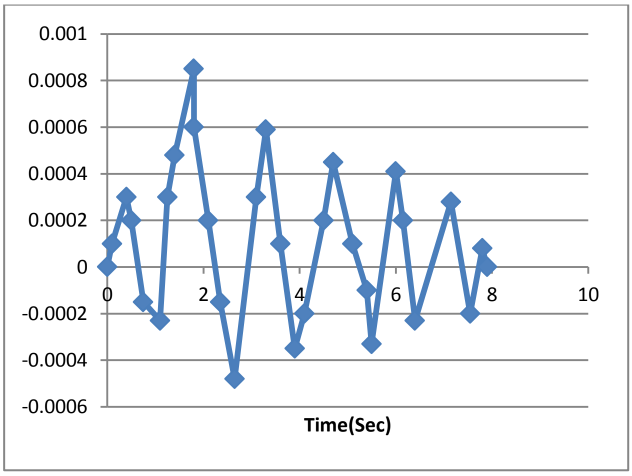

Fig.9: Sprung Mass Pitch (θ) for Half Car System

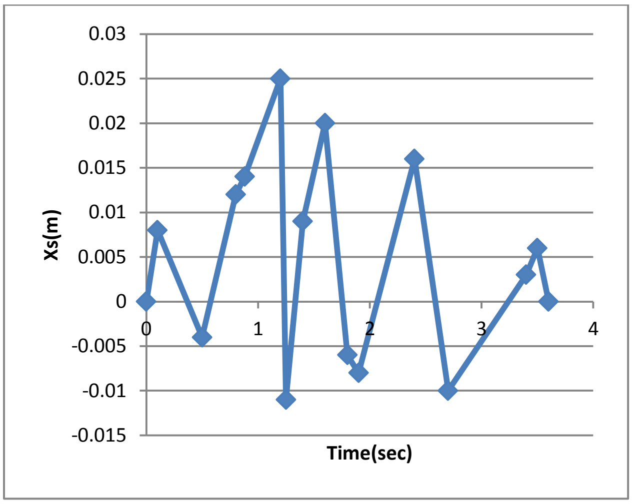

Fig.10: Sprung Mass Displacement (Xs) for Half Car System

Experimental results are shown in Table 2.

Table 2 Experimental Results

Sr. No.	Road Profile	Experimental Result	
		Sprung Mass Displacement (Xs), m	**Sprung Mass Pitch (θ), rad**
1	Bumpy (Sinusoidal) Input	0.055	0.00088
2	Rectangular Input	0.060	0.00095
3	Step Input	0.025	0.00085

Conclusion

Half car models can successfully be used to analyze the suspension system responses to different road profiles i.e Bumpy (Sinusoidal) Input, Rectangular Input, Step Input. Accuracy of the results obtained will depend on how accurately and effectively the system parameters (e.g. Sprung mass, Unsprung mass, Suspension Stiffness, Suspension Damping and Tire Stiffness) have been measured. Sprung mass displacement (Xs) for bumpy input is more than rectangular and step input for half car vehicle dynamic system. Pitch of Sprung mass for bumpy input is more than rectangular and step input for half car vehicle dynamic system. It is seen that behavior of nonlinear passive suspension system tends to move towards the actual behavior of the system.

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