Numerical and Experimental Ride Comfort Analysis of Quarter Car Model Active Suspension System Subjected to Different Road Excitations with Nonlinear Parameters

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ABSTRACT

The primary objective of ride comfort analysis is to study response behavior of sprung mass systems for different road excitation forces. This study is done by analyzing the active suspension system quarter car model subjected to three different road excitations considering nonlinear parameters. In this research work ride comfort analysis is carried out by both numerical and experimental method. For numerical method MATLAB/Simulink is used. The PID controller is tuned in MATLAB/Simulink for considerable actuator delay. In experimentation road excitations are generated on the rack. PID controller is used to control the pneumatic force actuator. With the help of FFT analyzer sprung mass displacements are observed. Both the numerical and experimental results are then compared for validation purpose.

Keywords: *Active Suspension, FFT Analyzer, Nonlinear Suspension Parameters, PID Controller, Ride Comfort, Quarter Car Model,*

INTRODUCTION

Vehicle dynamic analysis has been a hot research topic due to its important role in ride comfort, vehicle safety and overall vehicle performance. 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 [1].

The health and safety risks associated with prolonged exposure to high vehicle vibrations have prompted a demand for enhancement of ride quality performance of vehicle. Vehicle drivers are exposed to ride vibration for 8 to 10 hrs a day. In view of driver's health and safety risk associated with prolonged exposure to high levels of ride vibrations and significant dynamic tire forces resulting in accelerated suspensions needs to be further investigated for use in vehicles [2].

So, the ride comfort is a key issue in design and manufacture of modern automobiles. Design of advanced suspension systems is one of the requirements, which provide a comfortable ride by absorbing the road disturbances as well as maintain the vehicle stability.

It is well known that the ride comfort characteristics of passenger vehicles can be characterized by considering the so-called "quarter-car" model [3]. So for this research work quarter car model is selected.

Active suspension system is most advanced suspension system present now a days. Unlike passive systems, which can only store or dissipate energy, active suspensions can continuously change the energy flow to or from the system when required. Therefore, active suspension systems may have the potential to improve the ride quality, road holding ability, and safety of road vehicles through the control of suspension forces to satisfactory levels [4]. Active suspension are having force actuator between sprung mass and unsprung mass which is used to add or eliminate energy.

In this paper both the numerical and experimental work has been carried out for the ride comfort analysis of active suspension system.

MODELING OF ACTIVE SUSPENSION SYSTEM

Quarter car model is used for suspension system analysis and design for its simplicity and yet ability to capture many important parameters. In Fig. 1, Quarter Car Model Active Suspension System is shown.

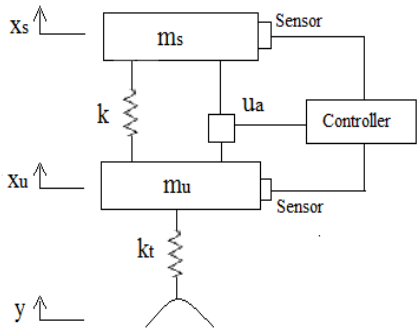

Fig.1. Quarter Car Model Active Suspension System

The sprung mass is denoted by 'm_s' and unsprung mass is by 'm_u'. Instead of damper, the force actuator is used in active suspension system. The tyre is assumed to have only the spring feature and is in contact with the road terrain at the other end.

For this analysis the Hyundai Elantra Model is selected. The Suspension Parameters for Hyundai Elantra Model are, [4]

Sprung Mass: m_s: 236.12 kg

Unsprung Mass: m_u: 23.61 kg

Suspension Stiffness: k: 12394 N/m

Tyre Stiffness: k_t: 181818.88 N/m

By applying Newton's Second Law of Motion for given system,

The equations of motion for the linear active suspension system are,

$$m_s\ddot{x}_s = -k(x_s - x_u) + u_a \quad (1)$$

$$m_u\ddot{x}_u = k(x_s - x_u) - k_t(x_u - y) - u_a \quad (2)$$

Where, y – Road displacement

u_a – Actuator force

x_s – Sprung mass displacement

x_u - Unsprung mass displacement

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 nonlinear effect can be included in spring force f_s with nonlinear characteristic versus suspension rattle space (x_s-x_u) from the measured data on SPMD (Suspension Parameter Measurement Device) is as shown in Fig.2

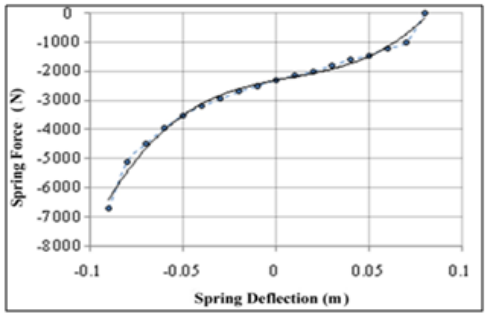

Fig.2. Nonlinear Spring Force Property of Hyundai Elantra Model Suspension Spring

The spring force f_s is modelled as third order polynomial function as,

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

Where the co-efficient are obtained by fitting the experimental data, which resulted in,

k_3 = 3170400 N/m^3, k_2= -73696N/m^2,

k_1= 12394 N/m, k_0= -2316.4 N

(The SPMD data from the model Hyundai Elantra front suspension were used) [5]

The equations of motion for active suspension system with nonlinear parameters are,

$$m_s\ddot{x}_s = [-k_0 - k_1(x_s - x_u) - k_2(x_s - x_u)^2 - k_3(x_s - x_u)^3] + u_a \quad (3)$$

$$m_u\ddot{x}_u = [k_0 + k_1(x_s - x_u) + k_2(x_s - x_u)^2 + k_3(x_s - x_u)^3] - k_t(x_u - y) - u_a \quad (4)$$

ROAD EXCITATION

Vehicle is assumed to be travelling over a road with velocity of 50 km/hr, during this travel the excitation frequency is calculated as,

$$\omega = \frac{2\pi V}{\lambda}$$

$$\omega = \frac{2\pi \times 50 \times 1000}{6 \times 3600} = 14.55\ \frac{rad}{sec} = 2.31\ \text{Hz}$$

For the ride comfort analysis of quarter car model active suspension system, three different road excitations are considered.

Bump Excitation

A single bump road 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. (5) is shown in Fig.3.

$$y = a\,(1 - \cos \omega t) \longrightarrow 0.4 < t < 0.9 \qquad (5)$$

In Eq. (5) of road disturbance, 'a' is set to 0.02 m to achieve a bump height of 4 cm.

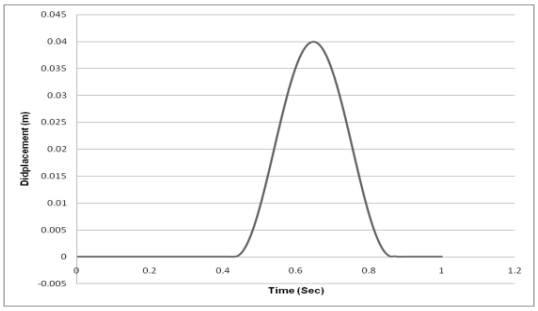

Fig.3. Road Profile for Bump Excitation

Step In Excitation

The Step In Excitation is represented by the Eq. (6),

$$y = \begin{cases} 0.00 & \longrightarrow \quad t < 0.43 \\ -0.04 & \longrightarrow \quad t \geq 0.43 \end{cases} \qquad (6)$$

The height of the road disturbance is maintained at 4 cm. The road input described by Eq. (6) is shown in Fig. 4

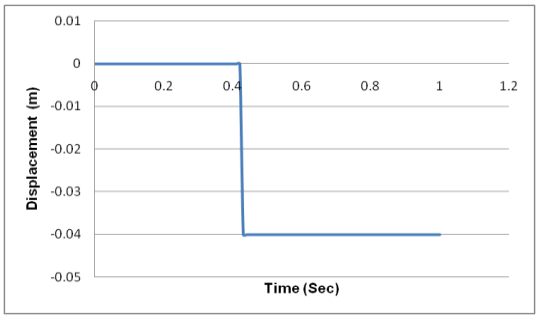

Fig.4. Road Profile for Step In Excitation

Rectangular Pulse Excitation

The Rectangular Pulse Excitation is represented by Eq. (7),

$$y = \begin{cases} 0.00 & \longrightarrow \quad 0.43 > t > 0.86 \\ 0.04 & \longrightarrow \quad 0.43 \leq t \leq 0.86 \end{cases} \qquad (7)$$

The height of the road disturbance is maintained at 4 cm. The road input described by Eq. (7) is shown in Fig. 5.

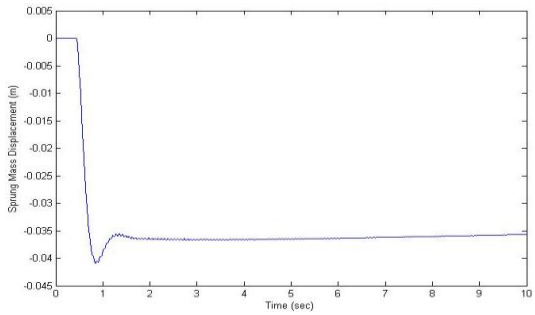

Fig.5. Road Profile for Rectangular Pulse Excitation

SIMULATION AND RESULTS

Mathematical model is transformed to computer simulation model and MATLAB/Simulink is used for the simulation. PID controller is used for controlling the force. For every system, PID controller should be set its value with respect to the system. For this reason, PID controller had to be tuned with system. In this research, Ziegler-Nicols Method is used to tune PID controller. According to the continuous cycle method (Ziegler-Nichols Method) first the PID block was attached to the system by setting $K_P = 1$, $K_I = 0$, $K_D = 0$. And then PID is tuned so that the response for the disturbance is good [6]. The PID controller is tuned until the desired response will be achieved.

Ride comfort analysis of quarter car model active suspension system for linear and nonlinear model is carried out in MATLAB/Simulink. For simulation the model variable-step continuous solver ODE45 (Dormand-Prince) is used.

For Bump Excitation

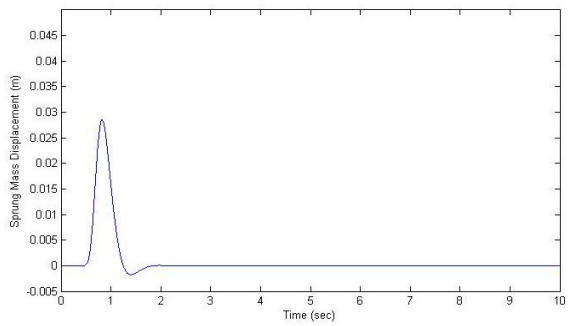

Fig.6. Sprung Mass Displacement of Nonlinear Active Suspension System

For Step In Excitation

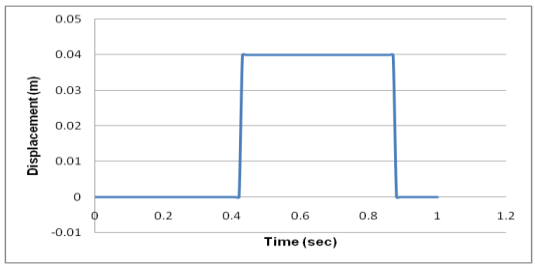

Fig.7. Sprung Mass Displacement of Nonlinear Active Suspension System

For Rectangular Pulse Excitation

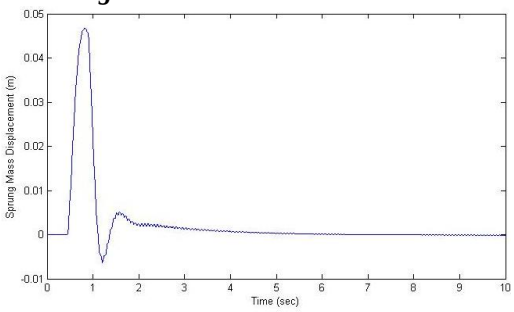

Fig.8. Sprung Mass Displacement of Nonlinear Active Suspension System

Table 1: Results for Nonlinear Active Suspension System

Sr. No.	Road Profile	Sprung Mass Displacement (m)
1	Bump Excitation	0.0285
2	Step In Excitation	-0.0408
3	Rectangular Pulse Excitation	0.0467

EXPERIMENTAL SETUP

The experimental setup is developed for quarter car model active suspension system. The quarter car model is developed for the analysis of active suspension system. Scaling factor of 1:100 is taken. The parameters of quarter car model active suspension system for experimentation are as in Table 2.

Table 2: Quarter Car Model Parameters

Sr. No.	Parameters	Value
1	Sprung Mass (m_s)	23.61 kg
2	Unsprung Mass (m_u)	0.23 kg
3	Suspension Stiffness (k)	123.94 N/m
4	Tire Stiffness (k_t)	1818.188 N/m

In order to determine sprung mass displacement a rack and pinion is provided in which a rack act as a road. As ride comfort analysis is to be carried out for different road profiles; on the rack, bump, step-in and rectangular pulse profiles are generated. A rack is given linear motion by pinion mounted on a shaft of DC motor. When a rack is in motion, an excitation is created by road profiles present on that and same will be transmitted to the follower which acts as a wheel. Thus, the excitation is transmitted to the unsprung mass and then to the sprung mass.

Between sprung and unsprung mass double actuating pneumatic cylinder is situated which works like a force actuator in active suspension system. The suspension spring is situated on the cylindrical rod of a pneumatic cylinder whose one end is supported to the sprung mass of active suspension system. The sprung mass of active suspension system carries LVDT sensor. This LVDT sensor is connected to the PID controller of the same system. Considering the nonlinearities in the system the following P, I and D parameters are set to the PID controller, which are calculated in the MATLAB/Simulink.

P = 846.0987; I = 65.9043; D = 2404.1887

Experimental model for ride comfort analysis active suspension system quarter car model is shown in Fig.9.

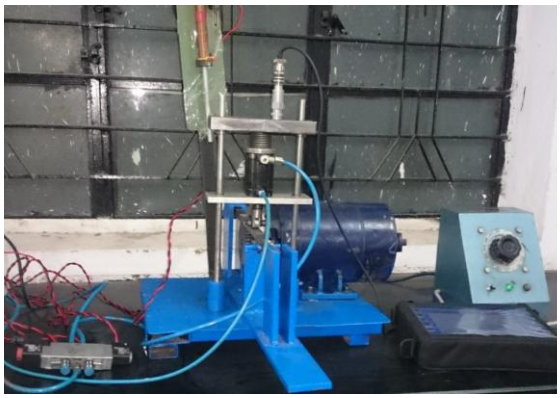

Fig.9. Experimental Setup

LVDT senses the vertical displacement of sprung mass and gives input signals to the PID controller in the form of voltage. With respective to the displacement of the sprung mass, PID controller controls the output voltage. This output voltage is provided to direction control valve, which controls and directs the inlet and outlet pressure of pneumatic cylinder. The air compressor is used to generate the pressure and pressure control valve is used to maintain the air pressure. Accelerometer and FFT Analyzer are used to find out ride comfort.

In order to carry out the experimentation, road profiles of same height are generated on the rack as used in MATLAB/Simulink.

EXPERIMENTAL RESULTS

FFT analyzer is used to measure the sprung mass displacement. The results obtained from FFT analyzer are as follows:

For Bump Excitation

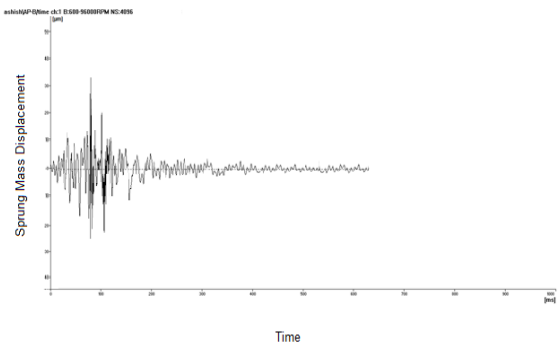

Fig.10. Sprung Mass Displacement for Active Suspension System Quarter Car Model

For Step In Excitation

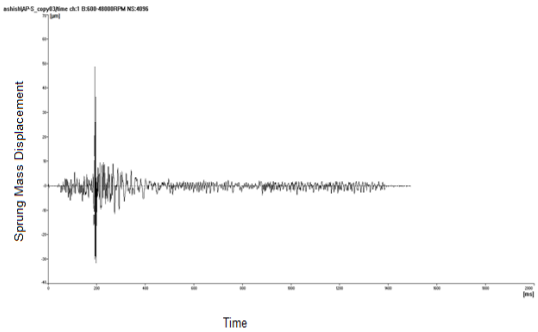

Fig.11. Sprung Mass Displacement for Active Suspension System Quarter Car Model

For Rectangular Pulse Excitation

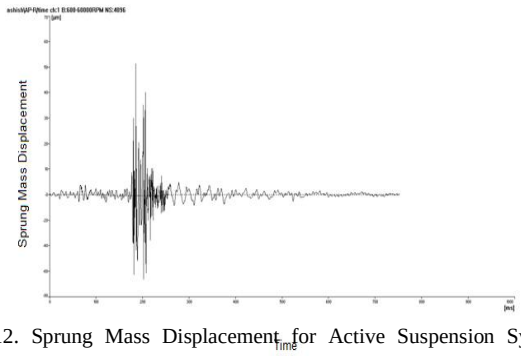

Fig.12. Sprung Mass Displacement for Active Suspension System Quarter Car Model

Table 3: Sprung Mass Displacement of Active Suspension System

Sr. No.	Road Excitation	Sprung Mass Displacement (mm)
1	Bump Excitation	0.03269
2	Step In Excitation	0.04863
3	Rectangular Pulse Excitation	0.05160

RESULTS

Table 4: Results for Sprung Mass Displacement of Active Suspension System

Road Profile	Numerical Results (m)	Experimental Results (m)	Variation (%)
Bump Excitation	0.0285	0.03269	14.70
Step In Excitation	0.0408	0.04863	19.19
Rectangular Pulse Excitation	0.0467	0.05160	10.49

When the results obtained from MATLAB/Simulink are validated with experimental results; it is found that the percentage variation for bump, stepin and rectangular pulse excitations are 14.70%, 19.19% and 10.49% respectively.

CONCLUSION

In this research work the vertical displacement of sprung mass of quarter car model active suspension system for different road excitation with actuator delay and nonlinear parameters is studied. It is observed to be less. The settling time of sprung mass is also found to be less. It seems that vertical displacement of sprung mass is very well controlled with the use active system. This indicates that active suspension system can provide good ride comfort.

The ride comfort is analysed by both MATLAB/Simulink and Experimental methods. The ride comfort of quarter car model active suspension systems, which analyzed in MATLAB/Simulink is validated with experimental result. It is found that the variations in the results are not so large; this ultimately validates the MATLAB/Simulink models and their ride comfort analysis results.

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