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Performance Evaluation of Shock Absorber Acting as a Single Degree of Freedom Spring-Mass-Damper System using MATLAB

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***Abstract*— Shock absorbers are required in an automobile suspension system in order to absorb and dissipate the energy transferred to the vehicle due to the impulse experienced during vehicle motion over uneven (in this case bumpy surfaces i.e.:-speed breaker profile) road profile. The main components of the shock absorber are the spring and the damper (dashpot). The spring acts as a detacher between the driver and chassis on one side and the motion of the wheels due to bump over an uneven surface on the other. It transfers the shock to the dashpot via deflections. It also supports the sprung mass of the vehicle. The damper bears the weight of the sprung mass and brings the vehicle to its equilibrium position of normal motion behavior in the shortest possible time by absorbing the impulse force and dissipating it internally.**

Passenger vehicles are generally designed for maximum user comfort and not for high performance. A prerequisite for this is to have a specific stiffness range and damping ratio for the suspension system. The objective of the paper is to study and analyze shock absorber (for the rear suspension system of a two-wheeler) by determining the motion transmissibility in order to achieve maximum ride comfort. Analytical results based on the equations of absolute motion of single degree of freedom of a spring-mass-damper system under the condition for forced damped vibrations as well as simulated results for the same (on Simulink Platform) in MATLAB were the tools used to achieve this objective.

Keywords*— Motion Transmissibility, Equations of Absolute Motion, Simulink, MATLAB.

1. INTRODUCTION

The modern motorcycle uses suspensions to bring about numerous necessary aspects; it provides a smooth and comfortable ride by absorbing the bumps and faults in the road. It also allows the driver to fine tune the machine to give him/her better control over it when driving.

We have taken a case into consideration the rear end of a two-wheeler of Gross Vehicle Weight (GVW) of approximately 250 kg, which is travelling over a standard speed breaker profile (according to National Road Authority of India). The vehicle is subjected to a bump at the bottom end of the rear wheel. Due to the presence of shock absorber, the complete motion due to the shock is not transferred to the vehicle body, but it is damped due to the presence of a shock absorber. The conditions for this damping will be different for different speeds of the vehicle and accordingly, the motion transmissibility of the system will be different. We have studied and analyzed the change in motion transmissibility with respect to the different speeds and have listed the results observed accordingly.

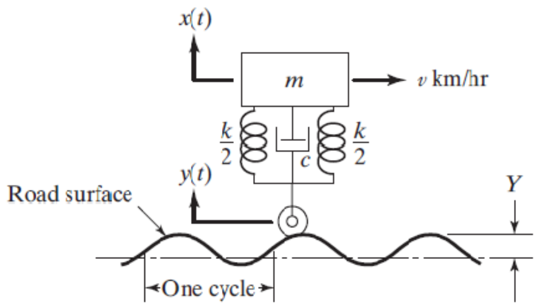


Fig. 1: Equivalent Spring-Mass-Damper System.

2. ROAD PROFILE

A standard speed breaker profile was taken into consideration for the experimentation. Two step input is used to denote wheel travel upwards and download on speed breaker.

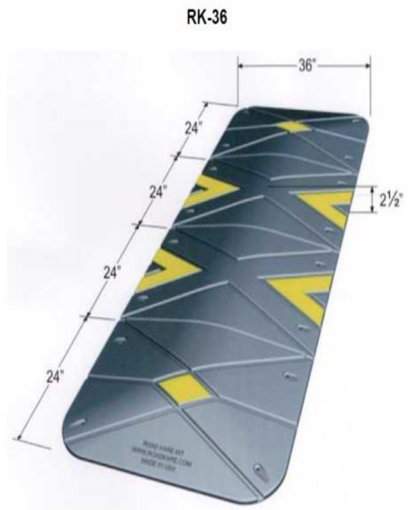
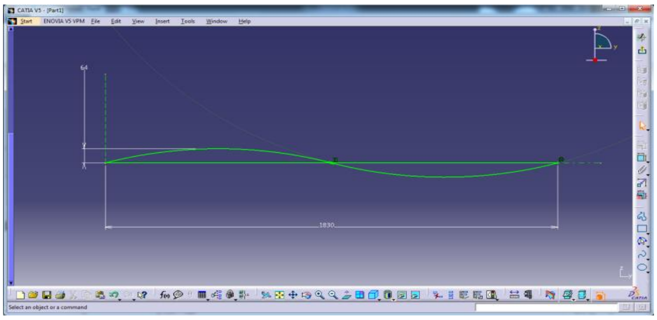


Fig. 2: Standard Speed-breaker Profile According to NHAI Specifications.



Wavelength, λ	1.828 m
Amplitude, Y	0.064 m

Table 2.1 Specifications of Speed Breaker

3. METHODOLOGY

Based on the theory and equations of absolute motion analytical results are obtained and a mathematical model of an equivalent system was developed in Simulink Platform in MATLAB. Results were obtained from both and via comparison, a conclusion was drawn.

4. LITERATURE REVIEW

- K. Kamalakannan , A. ElayaPerumalb, S. Mangalaramananc, K. Arunachalamd, Faculty of Automobile Engineering, Hindustan University, India & Faculty of Mechanical Engineering, Anna University, India. The aim of their paper is to simulate and analyze a simple and low-cost semi active suspension system using ““MATLAB and SIMULINK‟ platform and establish its superiority, and also involves the development and simulation of a virtual quarter car model. Thereafter the graphical results obtained are analyzed. The model was developed using equations of motion involving stiffness, damping ratio and displacement. These equations are translated into a simple data flow circuit in the simulation software to obtain definite results in the form of graphical output. This development of a simpler and cheaper semi active suspension system, which will allow its fitment in comparatively affordable cars.

- Andronic Florin , Manolache-Rusu Ioan-Cozmin , Pătuleanu Liliana, of University of Suceava, 13 Universităţii,720229, Suceava, Romania. In current article is simulated and analyzed the handling and ride performance of a vehicle with passive suspension system, quarter car model with two 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. The schema that was created in Matlab Simulink, were compared with the State space model and the Transfer function. After completing the simulation scheme can be introduced excitation signals, this case a step signal.

- K. S. Patil, Vaibhav Jagtap, Shrikant Jadhav, Amit Bhosale, Bhagwat Kedar, Department of Mechanical Engineering Shree Chhatrapati Shivaji College of Engineering. Pune, India. In this study, the active suspension system is proposed based on the Proportional Integral Derivative (PID) control technique for a quarter car model for the enhancement of its road handling and comfort. Comparison between passive and active suspensions system are performed by using road profile as an input. The performance of the active suspension system is evaluated by comparing it with passive suspension system. The performance of these will be determined by performing computer simulations using the MATLAB and SIMULINK toolbox. The simulation is enhanced with 3-D animation of car going on bump created in VRML.
- M.S.M.Sani, M.M. Rahman, M.M.Noor, K. Kadirgama and M.R.M.Rejab, Faculty of Mechanical Engineering, Universiti Malaysia Pahang, Tun Abdul Razak Highway,26300 Gambang, Kuantan, Pahang Malaysia. This paper was focused on the dynamic characteristics of an automotive shock absorber. The design of interchangeable shock absorber test rig was developed and fabricated for the dynamics measurement system. This test rig integrated with the computer systems to record the signal. It can be seen from the results that there is a good agreement between the experimental and simulated results in terms of stiffness and damping value except few discrepancy. The acquired results show that the range of discrepancy within 10%. The good range of stiffness of the passenger vehicle shock absorber is 20 N/mm to 60 N/mm while the damping of passenger vehicle shock absorber is 1 Ns/mm to 6 Ns/mm.
- Nikhil S. Kothawade, Amol D. Halwar, Ajay I. Chaudhari, Bhushan R. Mahajan Department of Mechanical engineering, Amrutvahini COE, Sangamner, Maharashtra, India. This paper mainly focuses on the measurement of transmissibility of shock absorber and its analysis at various loads and speeds. Transmissibility is a measure of effectiveness of the vibration isolating material. For the measurement of the transmissibility of the shock absorber a test rig is designed and developed. An experiment on the test rig is carried out at different speeds and loads which lead to the output in terms of sinusoidal waveform on strip chart recorder. The waveform is used to find out the transmissibility at different load-speed combination. The results obtained are used to find out the behavior of transmissibility at different speed and loads.

5. THEORETICAL BACKGROUND

5.1 TERMINOLOGY USED

A. Natural frequency (ωn)

When no external force acts on the system after giving it an initial displacement, the body vibrates. These vibrations are called free vibrations and their frequency is known as natural frequency. It is expressed in rad./sec. or Hertz.

B. Critically Damping co-efficient(Cc)

The critical damping co-efficient 'Cc' is that value of damping coefficient 'c' at which the frequency of free damped vibrations is zero and the motion is non-periodic.

C. Damping co-efficient (ξ)

It is defined as the ratio of damping coefficient to critical damping coefficient. Mathematically:-

$$\xi = c/Cc$$

D. Amplitude (X and Y)

It is the maximum displacement of vibrating body from its equilibrium position.

X= Displacement of Body.

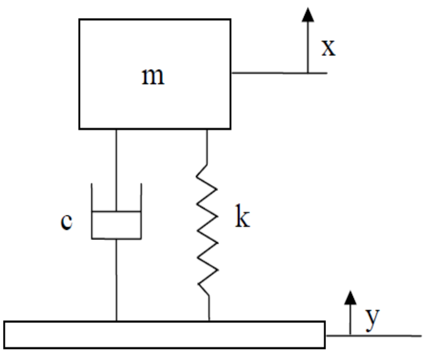
Y= Displacement of support.

E. Time period (tp)

The time required to complete one cycle.

5.2 ABSOLUTE MOTION

Absolute motion of a mass means its motion with respect to the coordinate system attached to the earth. As shown in figure, the absolute displacement of support is y=B sin ωt and the absolute displacement of the mass m from its equilibrium position is x. The displacement of mass m is relative to the support is z. The net elongation of the spring is (x'-y') and the relative motion between the two ends of the damper is (x-y). Then z= x-y and z'= x'-y'.



The equation of motion can be written as:-

$$mx'' + c(x'-y') + k(x-y) = 0$$

OR $$mx'' + cx' + kx = cy' + ky$$

Solving the equations we get,

$$\frac{X}{Y} = \frac{\sqrt{\left\{1 + \left(\frac{2\xi\omega}{\omega n}\right)^2\right\}}}{\sqrt{\left\{\left[1 - \left(\frac{\omega}{\omega n}\right)^2\right]^2 + \left(\frac{2\xi\omega}{\omega n}\right)^2\right\}}}$$

The ratio of **X/Y** is called the **displacement or motion transmissibility** which is the ratio of amplitude of the body to amplitude of the support.

5.3 ANALYTICAL CALCULATIONS

The good range of stiffness of the passenger vehicle shock absorber is 20 N/mm to 60 N/mm. Passenger vehicles are generally design for maximizing ride comfort and not for high performance; hence considering least stiffness value i.e.:- 20 N/mm and the damping ratio is taken as 0.25.

<u>Input:-</u>

Stiffness (k) = 20 N/mm
Damping coefficient (ξ) = 0.25
Mass (m) = 100 kg
Vehicle speed (v) = 25 km/h = 6.944 m/s

- Natural Frequency

  $$\omega n = \sqrt{\frac{k}{m}}$$

  = √(20000/100)
  =**14.14 rad/sec.**

- Critical damping coefficient
  Cc = 2*√(k*m)
  = 2*√(20000*100)
  = 2828.427 Ns/m.

- Damping coefficient
  ξ = c/Cc
  0.25 = c/2828.427
  c = 707.107 Ns/m.

- Time period
  tp = λ/v
  = 1.828/ 6.944
  = 0.2632 sec

- Excitation frequency
  ω = 2π/ tp
  = 2π/ 0.2632
  = 23.87 rad/sec

- Motion Transmissibility

$$\frac{X}{Y} = \frac{\sqrt{\left\{1 + \left(\frac{2\xi\omega}{\omega n}\right)^2\right\}}}{\sqrt{\left\{\left[1 - \left(\frac{\omega}{\omega n}\right)^2\right]^2 + \left(\frac{2\xi\omega}{\omega n}\right)^2\right\}}}$$

(Y = 0.064 m.,
ξ = 0.25,
ω = 23.87 rad./sec. and
ωn = 14.14 rad./sec.**)**
X = 0.042 m

6. MATHEMATICAL MODEL

The MATLAB Simulink model for Hero Splendor rear Suspension quarter Car Model is shown in fig. The sprung mass displacement for different excitation frequencies were obtained in time domain. This displacement for excitation frequency is inversely proportional to natural frequency of vehicle dynamic system. The values of amplitude response(X) are obtained in form simulated results.

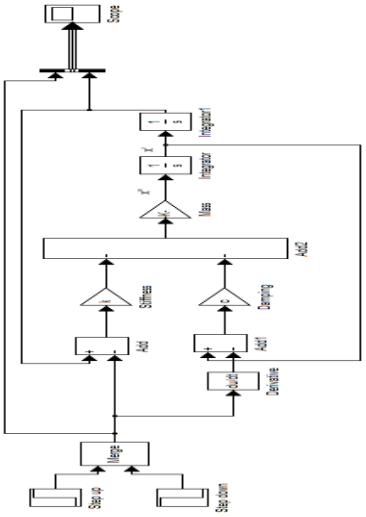
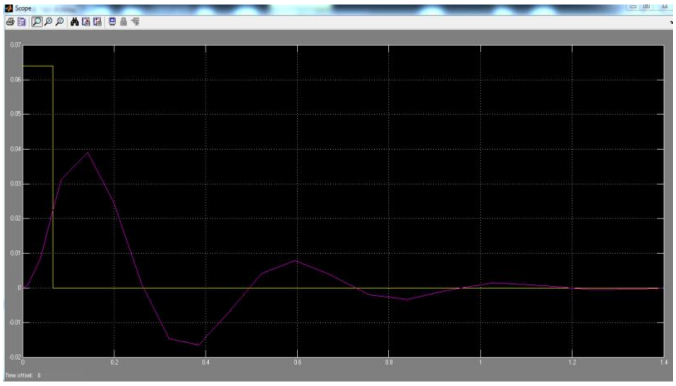
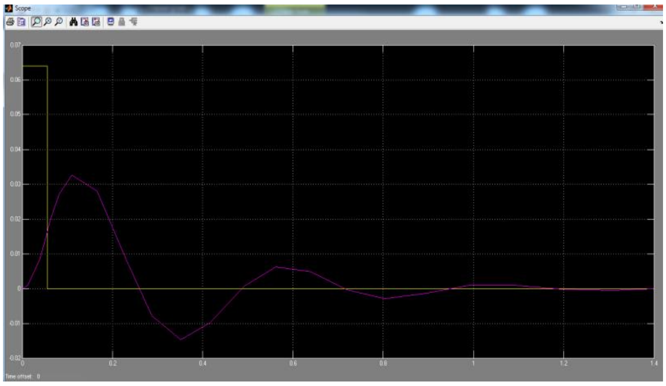


Fig 6.1 Simulink tree of quarter car model in single DOF system.

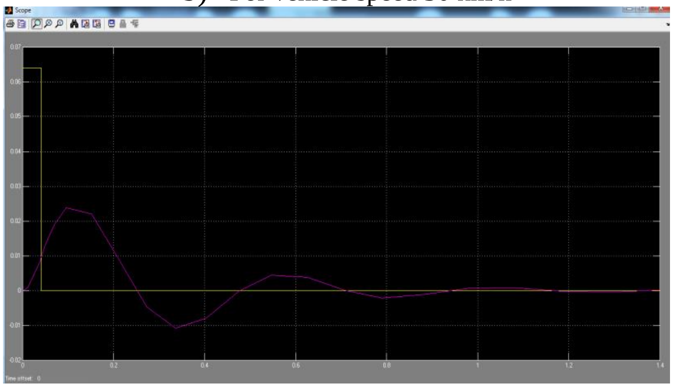
6.1 SIMULATED RESULTS



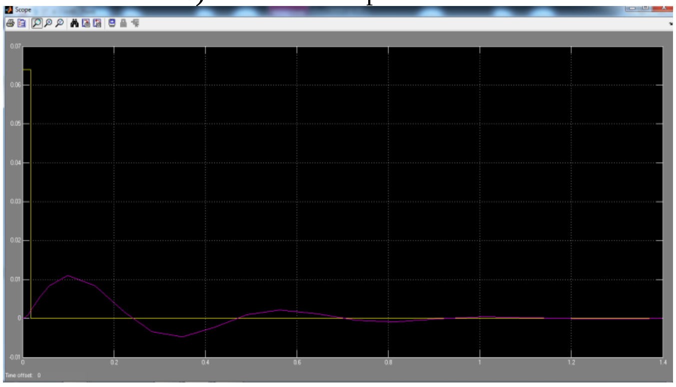
a) For vehicle speed 25 km/h



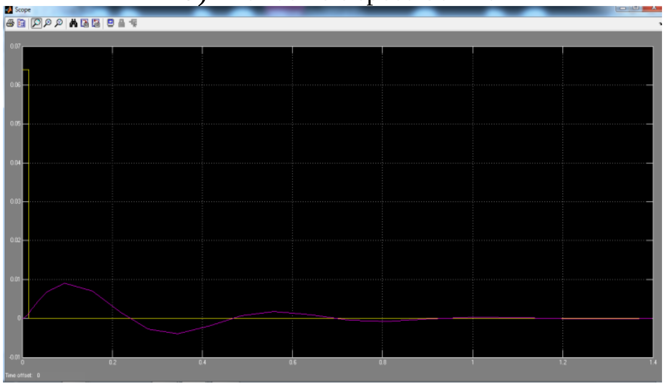
b) For vehicle speed 30 km/h



c) For vehicle speed 40 km/h



d) For vehicle speed 50 km/h



e) For vehicle speed 60 km/h

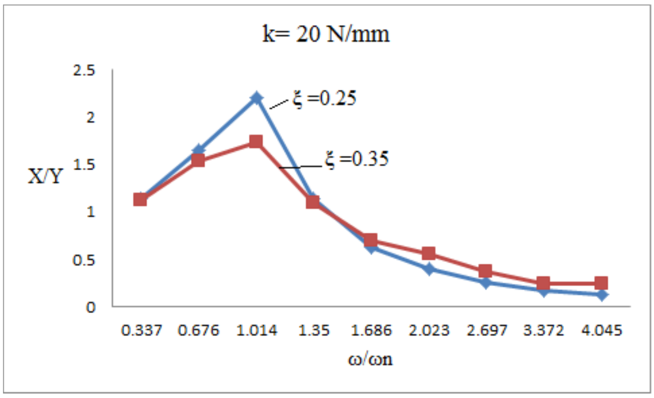
***(Key:-** Yellow ** step input*
Purple ** output)

7. RESULTS AND DISCUSSION

The theoretical and MATLAB Simulated results for different vehicle speed have been listed in below table:-

Velocity (km/h)	ω/ ωn	Amplitude X (m)		% error	Amplitude Ratio (X/Y)
		Analytical	*MATLAB*		
25	1.686	0.042	0.04	4.76	0.625
30	2.023	0.028	0.032	14.2	0.39
40	2.697	0.0167	0.016	4.19	0.25
50	3.372	0.012	0.011	8.33	0.1718
60	4.045	0.0093	0.0088	5.37	0.125

Frequency response curve for the following system is shown in below fig.



Explanation:-

The graph shown is a plot between Amplitude ratio (X/Y) and the ratio of excitation frequency and natural frequency of vehicle.

Analytical as well as Simulink results obtained show that the displacement amplitude decreases as velocity of vehicle increases. The excitation frequency is proportional to speed of vehicle. And hence the increase in speed or indirectly increase in excitation frequency, the amplitude X decreases. However, the amplitude is a complex function of suspension stiffness and damping coefficient and cannot be simply associated with velocity. For values of frequency ratio below 1, the amplitude ratio is always more than 1. As the excitation frequency approaches to the value of natural frequency (i.e. frequency ratio approaches unity), the amplitude ratio increases to much higher values. At resonance condition i.e. frequency ratio becomes unity and amplitude reaches infinity. Thereafter, for frequency ratios more than 1, amplitude ratio starts decreasing rapidly and is always less than 1 for frequency ratios greater than √2.

8. CONCLUSION

Two-wheeler rear suspension model or single wheel model of suspension analysis described in most of theory books provides a simple but approximate means of suspension system analysis. As verified by MATLAB Simulink the error involved lies between 0 – 15%.

The suspension system gives best performance when designed to be slightly under-damped and not when over-damped.

9. FUTURE SCOPE

- Determination of vehicle performance and driver comfort at different conditions of speed, mass, spring stiffness and vehicle speed etc.
- Expanding the concept for dependent rear suspensions of various vehicles.
- Performance of all terrain or SUV type vehicles over harsh terrain, to determine optimum system performance of both the driver and the machine.

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