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Optimization of Suspension Performance Parameters using a Quarter Car Model in MATLAB-Simulink

Shorya Sood
B.Tech Automotive Design,
University of Petroleum and Energy Studies, Dehradun
26.shorya@gmail.com

Naveen Gautam
B.Tech Automotive Design,
University of Petroleum and Energy Studies, Dehradun
navgautam25@gmail.com

ABSTRACT

All the on road vehicles in today's world have to face many different kinds of road loads. This demands for the incorporation of a suspension system in them.The fundamental purpose of ground vehicle suspension system is to:

1) Maintain continuous contact between the wheels and road surface

2) Isolate passengers or cargo from the vibrations induced by the road irregularities.

These two purposes are responsible for the handling quality and ride comfort, respectively.

However, these goals are generally contradictory. It is impossible for passive suspensions to achieve simultaneously a best performance of ride comfort and handling quality under all driving conditions.

Keywords

Suspension, Sprung mass, Unsprung mass, optimization, Matlab-Simulink

1. INTRODUCTION

The vehicle suspension systems are categorized into following types, namely:

- Passive suspension system
- Semi active suspension system
- Fully active suspension system

In the case of a Passive Suspension System, the two parameters viz. the spring constant and the damping constant are fixed from the design stage itself, so these cannot be controlled. These Passive suspension systems with no controllable standard characteristics are the most commonly used suspension system in the vehicles being produced presently.The use of this suspension system is so common because of a simple design, concerning high reliability and an absence of the necessity of any power supply contrary to the case of semi-active and active suspension systems.

The main problem of the Passive Suspension System is that if it is designed too soft, the passenger comfort increases but the road holding capability of the vehicle decreases but if it is designed too hard, the road holding capability of the vehicle increases but the passenger comfort is compromised. Thus, its design has to be optimized for obtaining the pest performance.

The Semi active Suspension System works by changing the viscous damping coefficient of the shock absorber. The damping is changed as per the road input and hence the road holding and ride comfort are optimized to some extent according to the condition of the road.

The Electronically controlled active suspension systems can potentially improve the ride comfort as well as the road handling of the vehicle simultaneously. The ride comfort is improved by means of the reduction of the car body acceleration caused by the irregular road disturbances from smooth road. Therefore the active suspension systems are superior in performance than passive and semi-active suspension.

Thus, contrary to case of the Semi-active and the Active suspension systems, the Passive Suspension system has to be optimized before manufacturing in order to obtain the best performance output from it.

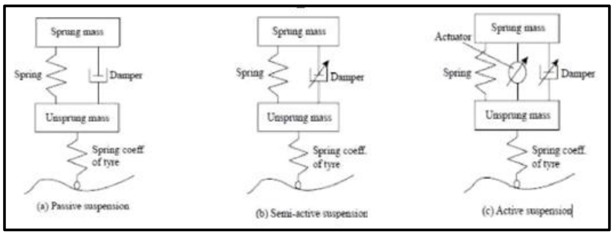

FIG. Quarter vehicle suspension models

1.1 Quarter Car Model

The quarter car model suspension system consists of one-fourth of the body mass, suspension components and one wheel as shown in Figure 1.The quarter car model for passive suspension system is shown in Figure 1(a).The quarter car model of a suspension system is independent of the type of suspensions being used because the mathematical model for any given type is made in the same way. The assumptions of a quarter car modelling are as follows:

- The tire is modelled as a linear spring without damping
- There is no rotational motion in wheel and body
- The behaviour of spring and damper are linear
- The tire is always in contact with the road surface
- The effect of friction is neglected so that the residual structural damping is not considered into vehicle modelling.

Following are some of the definitions related to the quarter car model:

Sprung Mass – it is the portion of the vehicle's total mass that is supported above the suspension, including in most applications approximately half of the weight of the

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suspension itself.

Unsprung Mass – it is the mass of the components directly connected to the suspension, rather than supported by the suspension.

2. MATHEMATICAL MODELLING OF QUARTER CAR

2.1 Mathematical Modelling Basics

A mathematical model of a dynamic system is defined as a set of equations that represents dynamics of the system accurately or at least, fairly well. A mathematical model is not unique to a given mechanical system. A system may be represented in many different ways and therefore may have many mathematical models, depending on one's perspective. The dynamics of a mechanical system can be described in terms of differential equations. Such differential equations may be obtained by using physical laws governing particular system, for example, Newton's laws are used in case of a mechanical system. Mathematical models may assume many different forms. Once a mathematical model of a system is obtained, various analytical and computer tools can be used for analysis and synthesis purposes.

2.2 Mathematical modelling of the Quarter Car Model of a Suspension System

The quarter car model takes into consideration, one-fourth the body mass of an automotive, one set of Suspension spring and damper, the unsprung mass on the tyre under consideration and the stiffness of the concerned tyre as a spring

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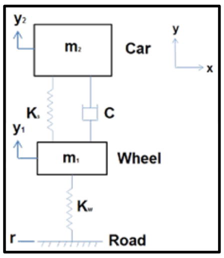

In the above figure:

m_2 – Sprung Mass

m_1 – Unsprung Mass

Ks – Stiffness of Suspension spring

C – Damping coefficient of Suspension Damper

Kw – Stiffness of tyre

y_1 – Vertical displacement of unsprung mass

y_2 - Vertical displacement of sprung mass

r - Vertical displacement of road profile

Further the Free Body Diagrams of the unsprung and the sprung masses are made in order to obtain their equations of motion as per the Newton's 2nd law of motion:

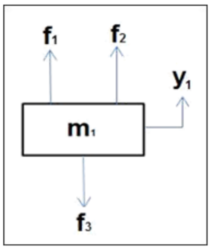

Fig. Unsprung Mass

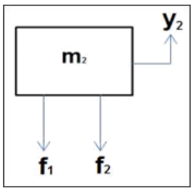

Fig. Sprung Mass

Assuming that $\mathbf{y_2>y_1>r}$, the various equations related to the above diagrams can be written as:

SUSPENSION SPRING FORCE:

$F_1= K_S(y_2-y_1)$

DAMPER FORCE:

$F_2= C(y'_2-y'_1)$

TYRE DEFLECTION FORCE:

$F_3= K_W(y_1-r)$

NEWTON'S EQUATIONS OF MOTION:

1. $M_1y_1''= K_S(y_2-y_1) + C(y'_2-y'_1) - K_W(y_1-r)$
2. $M_2y_2''= -K_S(y_2-y_1) - C(y'_2-y'_1)$

Where,

y_1' – Velocity of unsprung mass

y_2' – Velocity of sprung mass

y_1'' – Acceleration of unsprung mass

y_2'' – Acceleration of sprung mass

3. MODELLING OF THE QUARTER CAR MODEL ON MATLAB-SIMULINK

For the generation of any mathematical model on Simulink, the equations of motion of the system are required. As the 2 defining equations for the quarter car model under consideration have been found out, the various blocks can be used in Simulink library to create the Simulink model required.

The different blocks present in the Simulink library are chosen as per the requirement and connected to form a circuit representing the obtained equations of motion.The various variables present in the circuit are to be defined in the Matlab Editor in order to simulate the circuit.

Following pictures shows the Simulink model that was prepared for the required quarter car model simulation and the Matlab Editor into which the variables are defined:

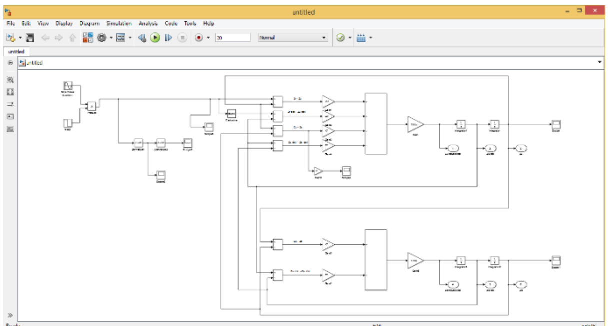

Fig. Simulink Circuit made to simulate the Quarter Car Model

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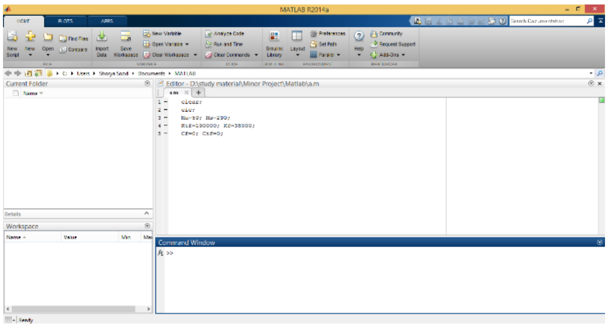


Fig. Variables defined in Matlab Editor

Following Table defines the parameters that were fixed for every simulation in Simulink:

Parameter	Definition/Value
Road Profile	0.02*Sin(ω*Time)
Sprung Mass	290 Kg
Unsprung Mass	59 Kg
Tyre Stiffness	190000 N/m

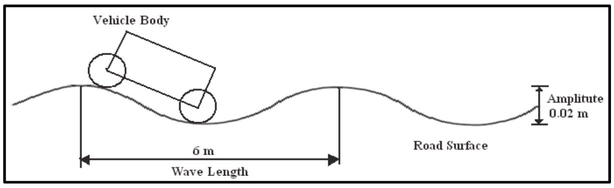
Where,

0.02 – Amplitude of road profile in meters.

ω – Angular frequency of the road.

The values of suspension stiffness, damping and the angular frequency of the road were varied for different simulations.

For understanding the Road profile:



Vehicle is assumed to be traveling over a road with velocity of 5 km/hr during this travel the excitation frequency is calculated as $\boldsymbol{\omega = 2\pi V/\lambda}$.

Thus, ω = (2π * 5 * 1000) /(6* 3600) =1.45rad/sec=0.23 Hz.

4. PERFORMANCE ANALYSIS OF QUARTER CAR MODEL

To understand the performance of a quarter car model , following are some important definitions that have to be known:

1.Ride Rate – The effective stiffness of the suspension spring and the tyre in series is called the ride rate and is defined as:

$$RR = \frac{KsKw}{Ks+Kw}$$

Where,

RR – Ride Rate

Ks – Stiffness of Suspension spring

Kw – Stiffness of tyre

2. Bounce Natural Frequency of the vehicle :-

$$\omega_n = \sqrt{(\frac{RR}{M2})} \qquad \text{(rad/sec)}$$

Where,

ω_n - Bounce Natural Frequency

m_2 – Sprung Mass

3. Damping Ratio :-

$$\zeta s = \frac{C}{\sqrt{4KsM2}}$$

4.1 Study of Response Gain for a given Suspension Model:

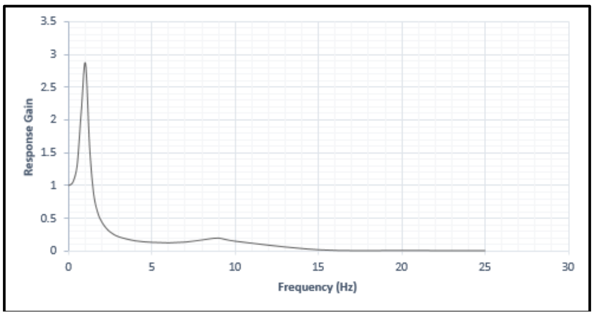
Response Gain in defined as the ratio of the output and input amplitudes for a given system. The term Transmissibility is also used in place of response gain and is defined as the ratio of the response amplitude to the excitation amplitude(of acceleration, velocity or displacement)

Given Suspension Model conditions:

Pre-defined Parameters	
Tyre Stiffness (N/m)	190000
Natural Frequency (Hz)	1
Angular Natural Frequency (rad/sec)	6.283185307
Unsprung Mass (Kg)	59
Sprung Mass (Kg)	290
Ride Rate(N/m)	11448.74111
Suspension Stiffness(N/m)	12182.83659
Damping Ratio	0.2
Suspension Damping(Ns/m)	751.853455

Values were recorded for different road input frequencies ranging from 0Hz to 25Hz and response gain was plotted against them.

The desired plot is given below:



Conclusions from the obtained plot:

The response gain at 0Hz is 1 implying that the sprung mass duplicates the road profile.

- As the frequency increases the response gain also increases upto a certain maximum value.
- This maximum value occurs when the natural frequency of the Suspension system becomes equal to the road input frequency and this is known as the resonance condition.
- Beyond the resonance condition, the response gain starts to attenuate and a slight increase in response gain is seen at the wheel hop resonance condition i.i. the natural frequency of the unsprung mass becomes equal to the road input frequency.

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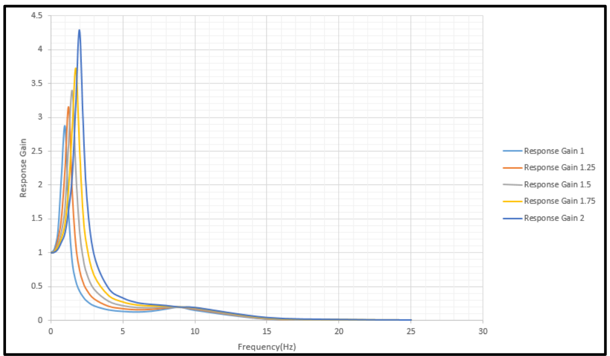
Volume 3 – No.2, April 2016

- Beyond the wheel hop resonance condition, the response gain is attenuated further and reaches 0 approximately.

Similarly the same study was performed for suspension models having natural frequencies 1.25Hz, 1.5Hz, 1.75Hz and 2Hz and the same graph was plotted for all of them together.



From the above graph it is noted that the response gain is higher for a suspension system having a higher natural frequency.

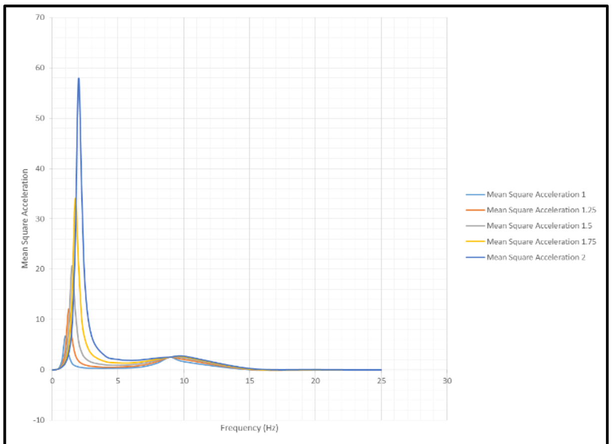
4.2 Study of Mean Square Acceleration for Suspension Systems having different Natural frequencies

The **Mean Square Acceleration** is defined as the product of the square of the response gain and the Peak Road Input Acceleration.

To obtain Suspension models having different natural frequencies, the following table was made:

S.No.	Tyre Stiffness (N/m)	Natural Frequency (Hz)	Angular Natural Frequency (rad/sec)	Sprung Mass (Kg)	Ride Rate (N/m)	Suspension Stiffness (N/m)	Damping Ratio	Suspension Damping (Ns/m)
1	190000	1	6.283185307	290	11448.74	12182.83659	0.2	751.853455
2	190000	1.16	7.288494956	290	15405.43	16764.73031	0.2	881.9770328
3	190000	1.25	7.853981634	290	17888.66	19747.94325	0.2	957.2379887
4	190000	1.5	9.424777961	290	25759.67	29799.84726	0.2	1175.888138
5	190000	1.75	10.99557429	290	35061.77	42996.07795	0.2	1412.451067
6	190000	2	12.56637061	290	45794.96	60337.99864	0.2	1673.225369

The mean square acceleration was recorded for different natural frequencies of the road profile for above given suspension models one by one and the plot of mean square acceleration for different suspension models is given below:



Conclusions from the obtained plot:

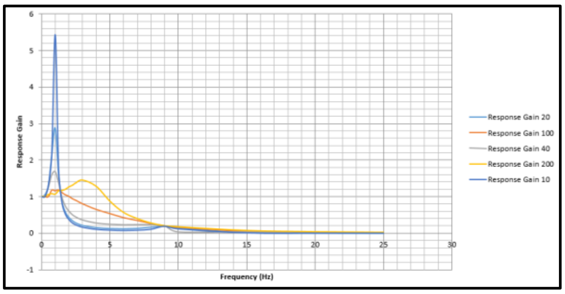
- The Mean Square Acceleration is maximum when the natural frequency of the Suspension Model is 2Hz.
- This implies that as the stiffness of the suspension spring increases, the peak mean square acceleration occurring at the resonance condition increases to a large extent.
- The graph also shows that the mean square acceleration is least for the case when the natural frequency is 1Hz.
- Thus to increase passenger comfort the best suited suspension system natural frequency is 1Hz but in this case the road holding ability of the vehicle will be compromised and vice versa for the 2Hz case.
- The graph also implies that the stiffer the spring becomes, the more the mean square acceleration gets at the wheel hop resonance condition also.

4.3 Study of Response Gain for Suspension Systems having different Damping conditions with fixed Spring stiffness

The following table gives the various Damping Coefficient Values(corresponding to various damping percentages) for a suspension model having fixed spring and tyre stiffness values:

S. No.	Tyre Stiffness (N/m)	Suspension Stiffness (N/m)	Damping Percentage	Damping Coefficient (Ns/m)
1	190000	12183	10	375.92
2	190000	12183	20	751.85
3	190000	12183	40	1503.70
4	190000	12183	100	3759.26
5	190000	12183	200	7518.5

The response gain was recorded for different natural frequencies of the road profile for above given suspension models one by one and the plot of response gain for the different suspension models is given below:



Conclusions from the obtained plot:

- The response gain for 10% damping condition is maximum at the resonance condition but is

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International Journal of Aerospace and Mechanical Engineering

Volume 3 – No.2, April 2016

minimum beyond that.

- The response gain for the 200% damping condition is minimum at the resonance condition but it increase beyond that becoming the highest of all the cases.
- Also for the 100% damping condition, the response gain is very less at the resonance condition but gets high as the frequency increases further.
- The graph shows that the best suited damping condition is of 40% because the response gain is less at the resonance condition as well as beyond it as compared to other cases.

4.4 Study of Sprung Mass Acceleration and Suspension Travel with different Damping conditions

Two cases were studied, one with Stiffness Ratio 5 and second with 20. Following are the pre-defined parameters for the two:

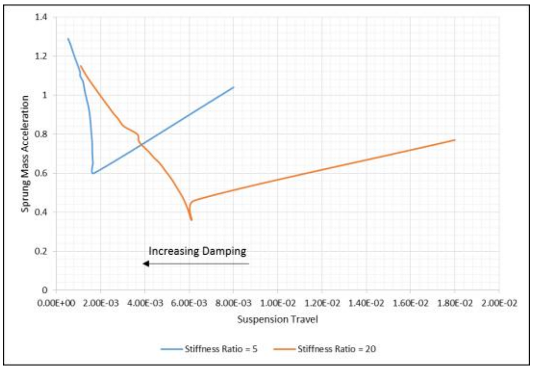
Pre-defined Parameters	
Stiffness Ratio	5
Tyre Stiffness	190000
Suspension Stiffness	38000
Sprung Mass	290
Unsprung Mass	59
Ride Rate	31666.66667
Angular Natural Frequency	10.44966039
Natural Frequency	1.663115105
Road Input	0.02*sin(3.14*time)

CASE 1

Pre-defined Parameters	
Stiffness Ratio	20
Tyre Stiffness	190000
Suspension Stiffness	9500
Sprung Mass	290
Unsprung Mass	59
Ride Rate	9047.619048
Angular Natural Frequency	5.585578428
Natural Frequency	0.888972417
Road Input	0.02*sin(3.14*time)

CASE 2

The sprung mass acceleration vs. suspension travel graphs for the two cases were plotted on a common axes and is as follows:



Conclusions from the obtained plot:

- The maximum value of the Suspension travel is very less in case of Stiffness Ratio = 5 as compared to that when Stiffness Ratio = 20 .
- The Maximum value of Sprung mass acceleration is more in case of stiffness ratio = 5 as compared to the other case.
- Both the cases have an optimum value (lowest point on the curve) for the sprung mass acceleration and suspension travel.
- If the damping value is decreased as compared to the optimum value, the sprung mass acceleration is increased and also the suspension travel increases.
- If the damping value is increases as compared to the optimum value, the sprung mass acceleration is increased but the suspension travel decreases.

5. CONCLUSION

The Passive Suspension System was mathematically modelled and then simulated using MATLAB-Simulink successfully. The research shows that the lower the natural frequency of the system, the lesser is the Transmissibility and the Mean Square Acceleration of the Sprung Mass. Thus a better ride comfort is obtained.

It is also concluded that the damping should be about 40% to obtain the most optimized ride comfort as at this damping ratio the Response Gain of the Sprung Mass is comparatively less at the resonance condition as well as beyond it as compared to other cases.

Hence, optimization of Suspension Performance Parameters was successfully done.

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