



Research Paper

SIMULINK MODEL OF SUSPENSION SYSTEM AND IT'S VALIDATION ON SUSPENSION TEST RIG

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Suspension system is made to have the ride comfort. The vibrations are not transferred from tire to the passenger if suspensions are good. According to road profile vibrations are transferred through the suspension strut to the chassis (unsprung mass) to the sit and to the passenger. In this case the most important factor is vertical acceleration produced in sprung mass. So, reduction in the vertical acceleration gives the better ride comfort. Thus analysis of suspension is to be done for vertical acceleration. This paper emphasise on the effect of suspension parameters i.e. sprung mass, unsprung mass, damping value, suspension spring stiffness and tire stiffness on vertical acceleration. For analysis purpose, the simple quarter car model is considered. The equation of motions is obtained in the differential form. These equations of motion are basis for the Simulink model in MATLAB. This simulation will help to simulate parameters of suspension. Thus vertical accelerations are obtained at different working conditions and maximum values of accelerations will be obtained. Then same excitations are given to suspension test rig. These values are then actually verified on suspension test rig.

Keywords: Simulink model, Suspension test, RIG

INTRODUCTION

The primary function of the suspension system is in a car to isolate the road excitations experienced by the wheels from being transferred to the passengers. The simplest and the most complicated models are as shown in the following cases. The mathematical models are able to convert the system into mathematical equations so the equations will be solved and some rigid conclusions can

be drawn for proper and optimised performance (Kalyanmoy Deb and Vikas Saxena, 1997).

Using these models the optimisation functions are obtained which contain different parameters to be optimised. The numerical simulation results indicate that the design criteria are significantly improved through optimizing the selected design variables. The effect of vehicle speed and road irregularity

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on design variables for improving vehicle ride quality has been investigated.

Figure 1: Two Degrees of Freedom (DOF) Quarter-Vehicle Model

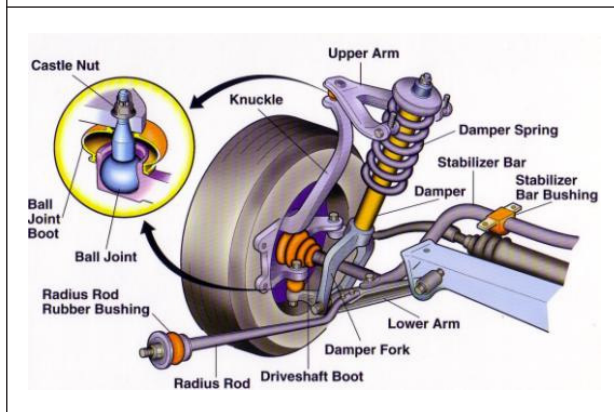


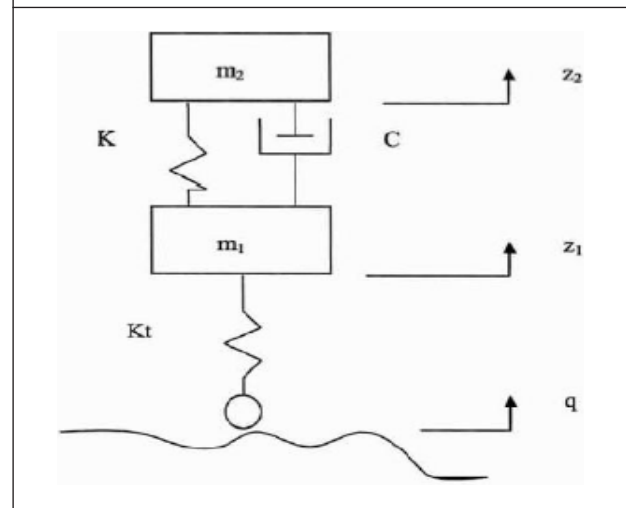
Figure 2 shows a simplified 2 degrees of freedom (DOF) quarter-vehicle model. It consists of a sprung mass (m_2) supported by a primary suspension, which in turn is connected to the unsprung mass (m_1). The tire is represented as a simple spring, although a damper is often included to represent the small amount of damping inherent to the visco-elastic nature of the tire. The road irregularity is represented by q , while m_1 , m_2 , K_t , K and C are the un-sprung mass, sprung mass, suspension stiffness, suspension damping coefficient and tire stiffness, respectively. This is very common model that can be considered while analysing the different parameters of suspension. In this model single suspension is considered for analysis purpose, which is shown in the Figure 1.2. The tire has been replaced with its equivalent stiffness and tire damping is neglected. The suspension, tire, passenger seat are modelled by linear springs with dampers.

q =road irregularity (m), K_t = tire stiffness

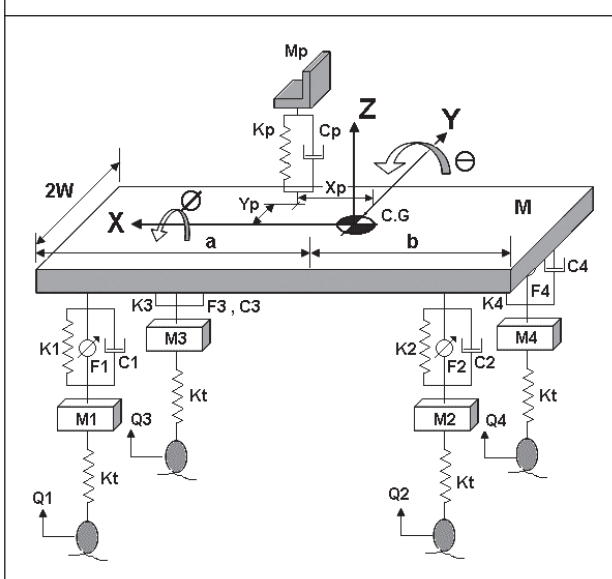
(N/m), m_1 =unsprung mass (kg), m_2 =sprung mass (kg), K =sprung mass stiffness (N/m), C =Damping coefficient (N/m/s), z_1 =displacement of unsprung mass (m), z_2 =displacement of sprung mass (m) (Zhongzhe Chi *et al.*, 2008).

The two degree of freedom is due to excitation of sprung and unsprung masses.

Figure 2: Two-DOF or Quarter Car Model Eight Degrees of Freedom (DOF) Full Car Model



A full car model with eight degrees of freedom is considered for analysis. Figure 1 shows a full car (8DOF) model consisting of passenger seat and sprung mass referring to the part of the car that is supported on springs and unsprung mass which refers to the mass of wheel assembly. The tire has been replaced with its equivalent stiffness and tire damping is neglected. The suspension, tire, passenger seat are modelled by linear springs in parallel with dampers. In the vehicle model sprung mass is considered to have 3DOF i.e. bounce, pitch and roll while passenger seat and four unsprung mass have 1DOF each (Anil Shirahatt *et al.*, 2008).

Figure 3: Eight DOF Model

M_p : Passenger seat mass (kg)

M : Sprung mass (kg)

M_1 & M_3 : Front left and front right side unsprung mass respectively (kg)

M_2 & M_4 : Rear left and rear right side unsprung mass respectively (kg)

K_p : Passenger Seat Stiffness (N/m)

K_1 & K_3 : Front left and front right side spring stiffness respectively (N/m)

K_2 & K_4 : Rear left and rear right side spring stiffness respectively (N/m)

K_t : Tyre stiffness (N/m)

C_p : Passenger seat damping coefficient (Ns/m)

C_1 & C_3 : Front left and front right side suspension damping co-eff. respectively (Ns/m)

C_2 & C_4 : Rear left and rear right side suspension damping co-eff. respectively (Ns/m)

F_1 & F_3 : Front left and front right side actuator force respectively (N)

F_2 & F_4 : Rear left and rear right side actuator force respectively (N)

a & b : C.G location from front and rear axle respectively (m)

$2W$: Wheel track (m)

X_p & Y_p : Distance of seat position from CG of sprung mass (m)

I_x : Mass moment of inertia for roll (kg-m²)

I_y : Mass moment of inertia for roll (kg-m²)

Q_1 & Q_3 : Road input at front left and front right side respectively.

Q_2 & Q_4 : Road input at rear left and rear right side respectively.

Out of above two the simple model is the quarter car model which considers the one suspension from car. For analysis purpose simple quarter car model is considered.

Equations of Motion for 2 DOF System and Simulink Model

From Free body diagram of the system following equations of motions can be derived.

$$m_2 \ddot{z}_2 + C(\dot{z}_2 - \dot{z}_1) + K(z_2 - z_1) = 0$$

$$m_1 \ddot{z}_1 + C(\dot{z}_1 - \dot{z}_2) + K(z_1 - z_2) + K_t(z_1 - q) = 0$$

With Simulink, one can move beyond idealized linear models to explore more realistic nonlinear models, factoring in friction, air resistance, gear slippage, hard stops, and the other things that describe real-world phenomena. Simulink turns your computer into a laboratory for modeling and analyzing systems that would not be possible or practical otherwise.

After you define a model, you can simulate it, using a choice of mathematical integration methods, either from the Simulink menus or

by entering commands in the MATLAB® Command Window. The menus are convenient for interactive work, while the command line is useful for running a batch of simulations. Simulink provides a graphical user interface (GUI) for building models as block diagrams, allowing you to draw models as you would with pencil and paper. Simulink also includes a comprehensive block library of sinks, sources, linear and nonlinear components, and connectors. If these blocks do not meet your needs, however, you can also create your own blocks. The interactive graphical environment simplifies the modeling process, eliminating the need to formulate differential and difference equations in a language or program.

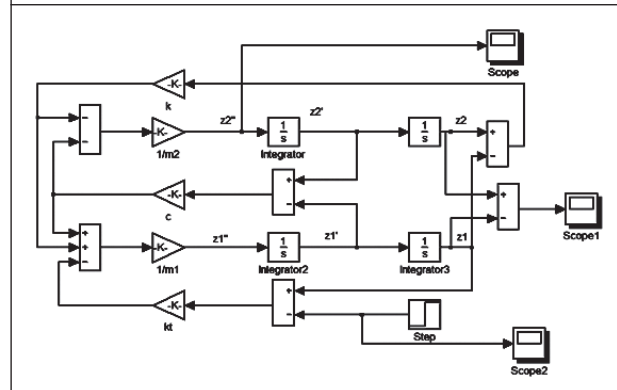
Models are hierarchical, so you can build models using both top-down and bottom-up approaches. You can view the system at a high level, then double-click blocks to see increasing levels of model detail. This approach provides insight into how a model is organized and how its parts interact.

Using scopes and other display blocks, you can see the simulation results while the simulation runs. You can then change many parameters and see what happens for "what if" exploration. The simulation results can be put in the MATLAB workspace for post processing and visualization.

Plots for the Simulink: The simulink results are obtained by varying suspension parameters as follows:

1. Sprung mass from 100 to 250 kg with equal intervals
2. Change in suspension spring stiffness
3. Change in damping coefficient

Figure 4: Simulink Block Diagram of the Suspension System



4. Change in unsprung mass
5. Effect of change in tire stiffness

Plots and respective acceleration and body travels are as follows:

Change in Sprung Mass

Table 1: Acceleration Values at Different Sprung Mass

Mass Kg	Vertical Acceleration (m/s ²)		Body Travel (m)	
	Max	Min	Max	Min
100.00	40.6414	-36.2788	0.0714	-0.0958
137.50	31.1445	-23.3874	0.0576	-0.1060
175.00	25.9298	-16.7602	0.0531	-0.1125
212.50	22.1732	-12.6394	0.0580	-0.1169
250.00	19.3520	-9.8861	0.0587	-0.1201

Figure 5: Change in Suspension Spring Stiffness

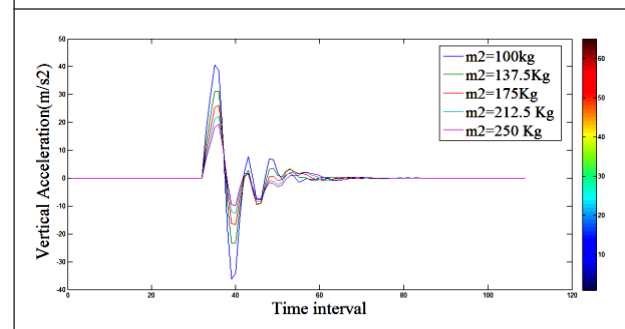
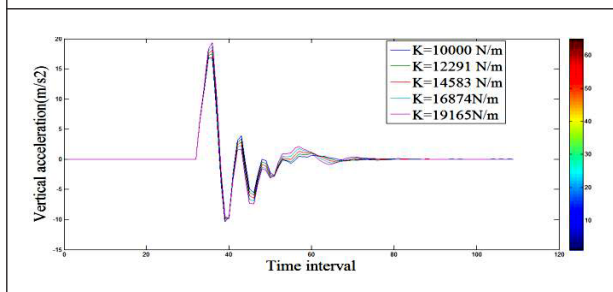


Table 2: Acceleration Values at Different Spring Stiffness

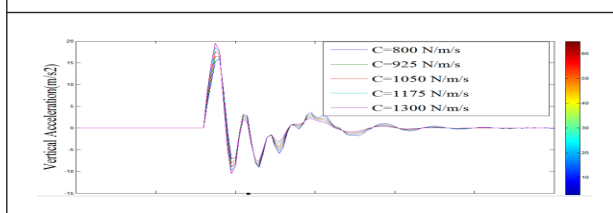
stiffness N/m	Vertical Acceleration (m/s ²)		Body Travel (m)	
	Max	Min	Max	Min
10000	16.8320	-10.3773	0.0411	-0.1262
12291	17.4679	-10.0866	0.0472	-0.1247
14583	18.0963	-9.8423	0.0521	-0.1232
16874	18.7175	-9.7336	0.0559	-0.1217
19165	19.3314	-9.8396	0.0587	-0.1202

Figure 6: Effect of Change in Suspension Stiffness

Change in Damping Coefficient

Table 3: Acceleration Values at Different Damping Coefficient

Damping Coefficient N/m/s	Vertical Acceleration (m/s ²)		Body Travel (m)	
	Max	Min	Max	Min
800	15.9339	-9.0880	0.0807	-0.1355
925	16.4648	-	0.0742	-0.1312
1050	17.5510	-8.9557	0.0685	-0.1271
1175	18.5702	-9.7732	0.0634	-0.1233
1300	19.5265	-	0.0589	-0.1197

Figure 7: Effect of Change in Damping Coefficient

Change in Unsprung Mass

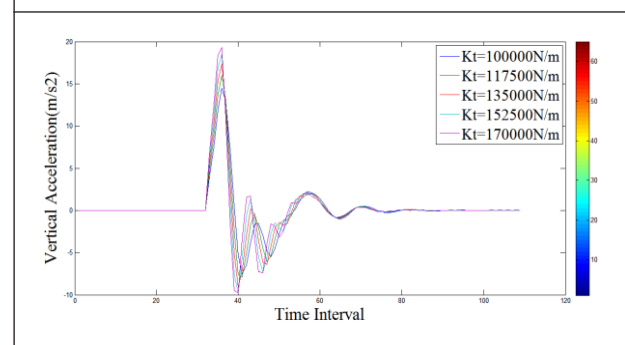
Table 4: Acceleration Values at Different Unsprung Mass

unsprung Mass	Vertical Acceleration		Body Travel	
	Max	Min	Max	Min
50	24.3433	-9.1475	0.0483	-0.1196
62.5	22.0938	-9.3245	0.0483	-0.1189
75	21.6044	-10.0099	0.0485	-0.1192
87.5	20.7262	-10.2262	0.0539	-0.1209
100	19.7185	-10.7132	0.0590	-0.1202

Effect of Change in kt

Table 5: Table Acceleration Values at Different Stiffness Values

kt	Vertical Acceleration m/s ²		Body Travel m	
	Max	Min	Max	Min
100000	14.4766	-7.5293	0.0543	-0.1027
117500	16.0433	-7.9088	0.0557	-0.1064
135000	17.3594	-9.1366	0.0581	-0.1092
152500	18.4383	-9.8698	0.0577	-0.1154
170000	19.2931	-9.7895	0.0587	-0.1203

Figure 8: Effect of Change in Tire Stiffness

RESULTS AND VALIDATION

Result Analysis

The results obtained were analytical and experimental in nature and were to be

compared simultaneously for accuracy. The results are given in subsequent topics.

Result Comparison

After simulating the McPherson strut in Matlab, the results were obtained in graphical format. The results obtained were readable and self explanatory in nature.

As shown in below figures, the results on Matlab simulation are given for various vehicle suspension parameters, which are self explanatory in nature.

The input data for suspension for two strut as follows as follows.

$$m_2 = 250 \text{ kg}$$

$$m_1 = 100 \text{ kg}$$

$$C = 1300$$

$$K = 16195 \text{ N/m}$$

$$K_t = 170000 \text{ N/m}$$

Table 6: Suspension Properties

Property	Strut1	Strut2
C (N/m/s)	1050	1300
K (N/m)	12283	19165

Where,

m_2 = sprung mass or body mass

m_1 = unsprung mass

C = suspension damper damping coefficient coefficient;

K = suspension spring stiffness;

And K_t = Tyre stiffness.

The simulation was carried at different values by changing the different parameters of the suspensions. The results of the same are shown in the previous section.

Following plots shows the response to the step input of 0.1m and speed of 30 kmph for two Struts are available with us.

For Change in Sprung Mass

Table 7: Comparison of Simulink and Experimental Result for Change in Sprung Mass

Sl. No.	Mass in Kg	Simulink Acceleration(m/s ²)		Experimental Acceleration(m/s ²)	
		Max.	Min.	Max.	Min.
1.	100	40.64	-36.27	30.25	-29.3782
2.	137.5	31.14	-23.38	27.38	-20.1872
3.	175	25.92	-16.76	21.54	-12.5684

Change in Spring Stiffness

Table 8: Comparison of Simulink and Experimental Result for Change in Spring Stiffness

Sl. No.	Spring stiffness N/m	Simulink Acceleration(m/s ²)		Experimental Acceleration(m/s ²)	
		Max.	Min.	Max.	Min.
1.	12283	17.4679	-10.0866	15.8312	-11.2349
2.	19165	19.3314	-9.8396	17.8562	-8.3248

Change in Damping

Table 9: Comparison of Simulink and Experimental Result for Change in Damping

Sl. No.	Damping Coeff. N/m/s	Simulink Acceleration(m/s ²)		Experimental Acceleration(m/s ²)	
		Max.	Min.	Max.	Min.
1.	1050	17.5510	-8.9557	13.7840	-7.4523
2.	1300	19.5265	-10.5107	17.4529	-8.2317

Experimental Results

Out of the four parameters viz. road distance, body travel, body acceleration and suspension deflection the body acceleration

is the most important and considerable parameter to be observed and kept within the permissible limits.

Hence in the present experimental tests as per the availability of instrumentation and the economy of project is concerned the concern has been given to body acceleration.

The experimental setup was capable of measuring only the body acceleration. The body acceleration was given in the form of tabulated data on HMI as an xls file reading the acceleration values at particular time interval. The acceleration values were recorded at regular interval of 10 second of simulation time. The tabulated data in the excel file was taken on the portable memory drive as the HMI facilitates the port for USB pen drive for data interaction. The Simulink results and actual results are varying by 15%-20% due to following conditions.

- The analytical analysis uses the standard equations, methods which may or may not consider the actual working conditions.
- The analysis occurs on the basis of many assumptions and considerations, such as there is no material defect in the specimens that practically may have e.g. voids, nonlinear material properties, anisotropic nature, manufacturing defects, defects induced while handling.
- The simulation method also assumes that the specimen for which the model is developed has the perfect geometry and lacks no errors in their alignments, positioning and working.

- There is no human error involved while the results are being recorded unlike in experimental tests.

CONCLUSION

Effect of Change in Sprung Mass

From the result of simulation it is clear that initially the less amount of sprung mass causes the large vertical acceleration. But as sprung mass goes on increasing then vertical acceleration sharply decreases. But sprung mass can't be increased beyond 175Kg otherwise the high dynamic forces strikes the shaker. Thus high sprung mass gives comfort but striking of sprung and unsprung mass should not takes place. The simulation result shows that we have extended the mass upto the 250 kg .But actually we take it upto 175 kg for safety purpose.

Change in Suspension Spring Stiffness

Simulation shows that as suspension spring stiffness increases the vertical acceleration increases. From results it can be proved that 18.63% change in the stiffness changes vertical acceleration only by 3.64%.It does not mean that we should use less stiffness spring. That causes increase in flexibility. Thus it should be such that it will reduce vertical acceleration as well as should give the stability. In suspension spring we take five different values but we are having the two struts thus experimental and simulation values are compared for those two values only.

Change in the Damping Coefficient

The damping coefficient damps the amplitude suddenly and brings system to the mean position. The vertical acceleration increases

by 3.2% while an increase in the damping coefficient is increased by 13.5%.

Change in the Unsprung Mass

The unsprung mass of quarter car model is the $\frac{1}{4}$ th of the total car. This mass is not being changed practically. But still we can get its effect on the vertical acceleration in simulink. As unsprung mass increases the vertical acceleration goes on decreasing.

Tire Stiffness

Tire stiffness is dependent of the pressure in the tire. The minimum pressure in the tire should be maintained. At that air pressure the particular stiffness should be considered. As tire air pressure increases the rigidity of tire and transfers the vibrations. Thus, tire pressure increases the stiffness increases

and vertical acceleration also goes on increasing.

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