Simulation of Quarter Car Model Using Matlab

Kuldeep.K.Jagtap[1], Dhananjay.R.Dolas[2]

PG Scholar, JNEC,Aurangabad[1], Associate Professor, JNEC, Aurangabad[2]

kdjagtap007@gmail.com[1],dhananjay_dolas@rediffmail.com[2]

Abstract-Vehicle suspension design includes a number of compromises to provide good leveling of stability and ride comfort. Suspension system has to perform complexity requirements, which includes road holding and equality, driving pleasure, riding comfort to occupant. Riding pleasure depends on vertical acceleration, with main objective to minimize vertical acceleration. In this work, the geometric parameters of suspension system are optimized using Matlab as an optimization tool. The values of tire deflection reduced from 0.023 to 0.021 m and Seat acceleration reduced from 34.19 to 31.86 m/s^2.

Keywords-Seat Acceleration, Tire Deflection, ride comfort, mathematical model.

I. INTRODUCTION

Traditionally automotive suspension designs have been a compromise between the three conflicting criteria of road holding, load carrying and passenger comfort. The suspension system must support the vehicle, provide directional control during handling maneuvers and provide effective isolation of passengers/payload from road disturbances. The parameters of Passive Suspension system are generally fixed, being chosen to achieve a certain level of compromise between road holding, load carrying and comfort.

The work presented here tries to analyze the effect of spring stiffness and Damping coefficient on the Seat Acceleration and Tire deflection and analysis by equation of motion using mathematical blocks available in Simulink.

S. J. Chikhale et al [1] analyzed that suspension model in Matlab-Simulink and MSc-Adams. Results for the analysis are compared for understanding the capability of the software to deal with the vibration problems.

P. Sathishkumar et al [2] discussed about the mathematical modeling and simulation of 2DOF quarter car model. The state space mathematical model is derived using Newton's second law of motion and free body diagram concept. The performance of the system is determined by Matlab/Simulink.

Nikos E. Mastorakis et al [3] have tested the performance of a suspension damper using a FFT with the simulation environment under MATLAB/Simulink. The Simulink results and experimental results are compared and are similar in nature.

K. S. Patil [4] obtained a mathematical model for the passive and active suspension systems for quarter car model in their research paper. Comparison between Passive and Active Suspension for Quarter Car Model is done. Dynamic model for linear quarter car suspensions systems has been formulated and derived.

S. Segla [5] has optimized the performance of the passive suspension system and compared with the performances of active and semi-active suspensions. Mat lab is used for the optimization of the suspension parameters.

2. Mathematical Model for Quarter Car

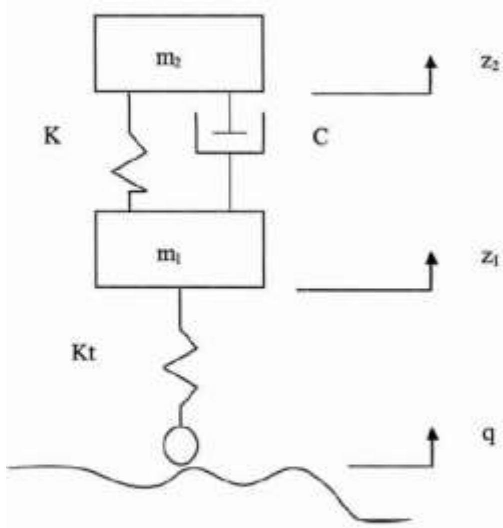

Fig 1.Quarter Car Model

The equation of motion for 2DOF system is given as:

$$M_2\ddot{Z}_2 + C(\dot{Z}_2 - \dot{Z}_1) + K(Z_2 - Z_1) = 0$$

$$M_1\ddot{Z}_1 + C(\dot{Z}_1 - \dot{Z}_2) + K(Z_1 - Z_2) + K_t(Z1 - q) = 0$$ Taking the Fourier transfer on both sides

$$Z_2(-\omega^2 m_2 + jwc + K) = Z_1(jwc + K)\ Z_1(-\omega^2 m_1 + jwc + K + K_t) = Z_2(jwc + K) + qK_t$$

For the purposes of design optimization, according to James' principle, the root mean square (RMS) of the sprung mass acceleration $\ddot{Z}_2$ can be expressed as follows,

Taking the amplitude ratio for amplitudes of sprung mass and road excitation

$$\frac{|Z_1|}{|q|} = \left[\frac{(1-\lambda^2)^2 + 4\xi^2\lambda^2}{\Delta}\right]^{1/2}$$

Where,

$$\Delta = \left[(1-(\omega/\omega_0)^2)(1+\gamma-\frac{1}{\mu}(\omega/\omega_0)^2 - 1\right]^2 + 4\xi^2(\omega/\omega_0)^2\left[\gamma-(\frac{1}{\mu}+1)(\omega/\omega_0)^2\right]$$

$$\gamma = \frac{K_t}{k}, \mu = \frac{m_2}{m_1}, \omega_0 = \sqrt{K/m_2}, \xi = \frac{C}{2\sqrt{m_2 K}}$$

The amplitude ratio between sprung mass displacement,Z_2, and the road excitation,q,is

$$\left|\frac{z_2}{q}\right| = \gamma\left[\frac{1+4\xi^2\lambda^2}{\Delta}\right]^{1/2}$$

The amplitude ratio between sprung mass acceleration, $\ddot{Z}_2$, and the road excitation, $\ddot{q}$, is

$$\left|\frac{\ddot{z}_2}{\dot{q}}\right| = \omega\gamma\left[\frac{1+4\xi^2\lambda^2}{\Delta}\right]^{1/2}$$

The suspension working space is the allowable maximum suspension displacement. The suspension working space in response to the road displacement input is:

$$\frac{f_d}{\dot{q}} = \frac{\gamma}{\omega}\lambda^2\left[\frac{1}{\Delta}\right]^{1/2}$$

The dynamic tire load is defined as $F_d = K_t(z_1 - q)$, and also the static tire load is $G = (m_1 + m_2)g = m_1(\mu + 1)g$ where g is gravitational acceleration. Thus the amplitude ratio between the relative dynamic tyre load, $\left|\frac{F_d}{G}\right|$ and the road input, q, becomes

$$\frac{F_d}{G_q} = \frac{\gamma\omega}{g}\left[\frac{\left(\frac{\lambda^2}{1+\mu} - 1\right)^2 + 4\xi^2\lambda^2}{\Delta}\right]$$

3. Simulink Model of Quarter Car

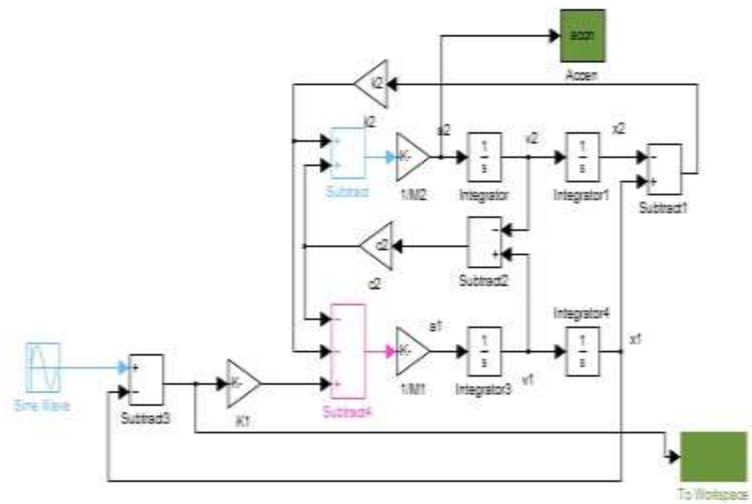

Fig no 2.Simulink Model of Quarter Car

The simulink results are obtained by suspension parameters as follows:

Table no1. Fix Parameters

Parameter	Value
M_1	40 Kg
M_2	60 Kg
K_t	100000 N/m

Table no 2. Variable Parameters

Parameter	Value
K_2	18760(N/m)
C2	900N-s/m)

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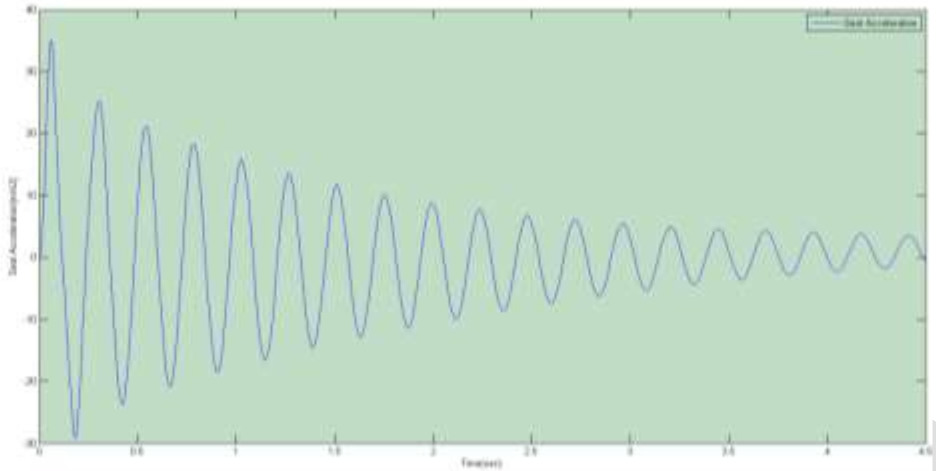
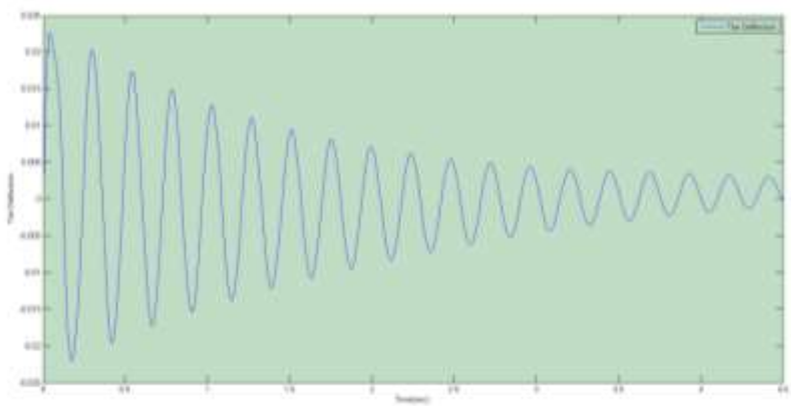
a. Plot for Seat Acceleration



Graph no 1. Seat Acceleration

b. Plot for Tire Deflection



Graph no 2. Tire Deflection

From both the graphs, we get the maximum and minimum values of the seat acceleration and TD as follows:

Table no 3. Observed Values

Parameter	Maximum Value	Minimum Value
Seat Acceleration(M/s^2)	34.9176	-29.1136
Tire Deflection(m)	0.023	-0.0219

Now, optimization program was run using for the objective function:-

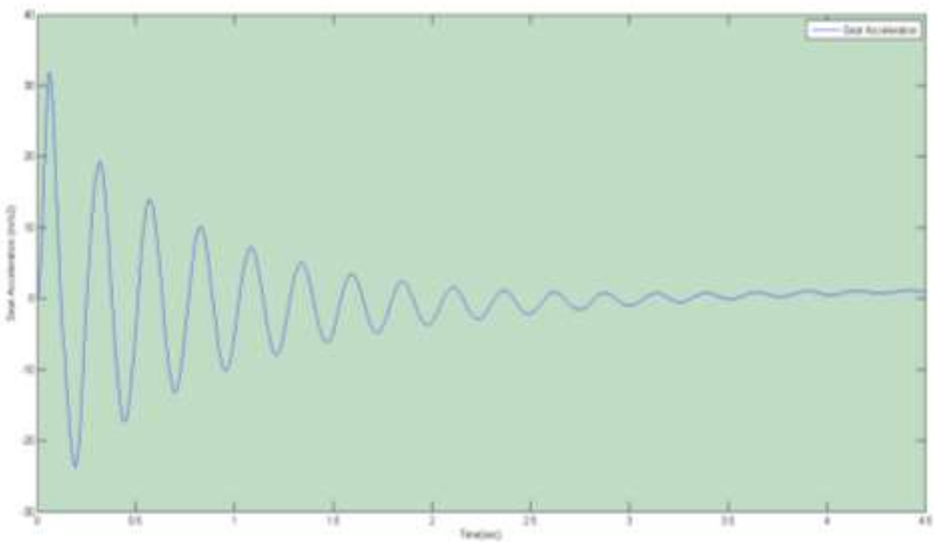
Min (TD) and min (Seat Acceleration)

The obtained values for the design variables K & C are:

Table no 4. Optimized Parameters

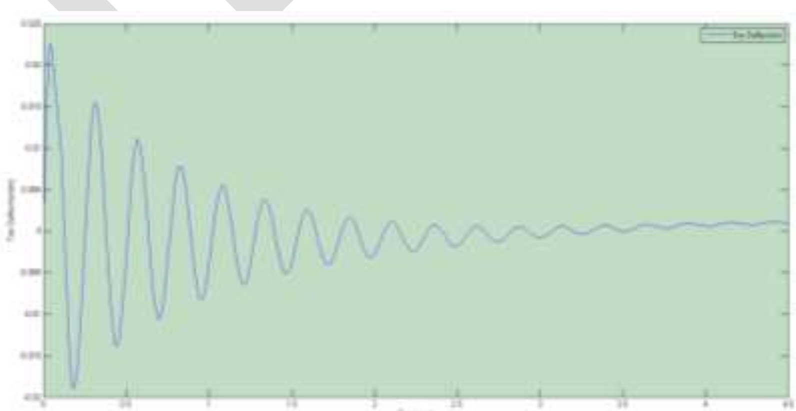
Parameter	Value
K_2	12265 (N/m)
C2	536 N-s/m)

c . Plot for Seat Acceleration



Graph no 4. Seat Acceleration vs. Time

d. Plot for Tire Deflection



Graph no 5. Tire Deflection vs. Time

From both the graphs, we get the maximum and minimum values of the seat acceleration and TD as follows:

Table no 5. Maximum and Minimum values of the Seat Accl^n & TD

Parameter	Maximum Value	Minimum Value
Seat Acceleration(M/s^2)	31.8604	-23.7734
Tire Deflection(m)	0.022	-0.0189

4. Results and Discussion

Table no 6. Results comparison of Optimized Values

Parameter	**Classical Value**	**Optimized Value (Simulink Model)**
Seat Acceleration(m/s^2)	34.9176	31.8604
Tire Deflection(m)	0.023	0.022

From Table No 6 we can compare the classical and simulink values and can observe that the Seat Acceleration has been decreased by 3.0572 m/s^2 and Tire Deflection by 0.001 m.

Thus we can see that the seat acceleration values are reduced by optimization. The ride comfort will be increased with the minimization of the Seat acceleration and tire deflection**.**

Conclusion

The simulation result shows considerable difference in linear passive sprung mass. As the results of analysis of quarter car passive suspension are quite similar because experimental model contains inherent nonlinear properties of suspension parameters, so it is necessary to consider the nonlinearities in suspension system for analysis of tire dynamic force. Hence form above we conclude that the analysis of suspension system we can use only theoretical (MATLAB Simulink) models instead of difficult experimental setup.

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