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Optimization of Suspension System for Minimization of Tire Dynamic Force

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ABSTRACT

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. The force transmitted by the deflection of Tires to the unsprung mass is known as the Tire-Dynamic Force (TDF). The force should be considered while designing the suspension system. The TDF can cause vehicle instability and increase in the sprung mass acceleration.

In this dissertation, a 2-Degree Of Freedom (2-DOF) quarter car model is considered. The objective is to minimize the Tire Dynamic Force (TDF) by optimum suspension design so that minimum Vertical Accelerations would be experienced by the passengers. Spring Stiffness and Damping Coefficient are used as design variables during optimization.

The GA is used as an optimization tool with above mentioned objective function. The optimization results obtained were simulated and compared with classical values. It is observed that the Seat acceleration, using optimized values, is reduced by 8.76% compared to the classical values.

From experimental validation main objective function of Tire Dynamic Force value is reduced from 2300 N to 2100 N, it is decreased by 8.69 % and Seat acceleration reduced by 8.30% after replacing the conventional by optimized strut. Thus provides improved ride comfort to occupant/driver.

Up till now, seat acceleration is given prime importance for ride comfort. But

TDF plays an important role in designing the Suspension system as the large value of TDF can cause discomfort to the occupant. So, the TDF should be considered by the designers to improve the ride comfort and road handling properties of a vehicle.

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INTRODUCTION

The purpose of the vehicle suspension system is to maintain contact between the wheels and road surface and to isolate the vibrations from passing directly to the passengers. In the area of ride comfort Work has been continued. Big magnitude of vertical vibration generated by the transportation modes like buses, cars, trains may cause diseases and health problems to the human. Researchers are continuously working to improve the ride comfort of the occupant. Many optimization techniques have been used to minimize the vertical accelerations. Literature survey has been carried out regarding the same.

Koch et al.[1] has designed a quarter car test rig for the active suspension system to study the nonlinear dynamic behavior of suspension system and have experimentally validate the controllers for active suspension systems. A quadricycle suspension system is integrated in the test rig in order to reduce the test rig's mass and to facilitate the realization of a high bandwidth active suspension system.The active suspension is realized by an electrical linear motor which is incorporated in parallel to the passive suspension strut. A nonlinear as well as a linear test rig model is derived and validated. The results from the test rig are as per the requirements. Shirahatt et al. [2] gives the suggestions for full passenger car model Optimal Design of Passenger. Optimization is achieved in ride comfort and road griping and the model is with eight DOF. A number of objectives such as maximum bouncing acceleration of seat and sprung mass, jerk, suspension travel, root mean square (RMS) weighted acceleration of seat and sprung mass as per ISO2631 standards, road holding and tire deflection are minimized subjected to a number of constraints. The constraints arise from the practical kinetic and comfort ability considerations, such as limits of the maximum vertical acceleration of the passenger seat, tire displacement and the suspension working space. The Genetic Algorithm (GA) is used to solution of the problem and results were compared to those obtained by simulated annealing (SA) method and found to yields similar performance measures. Gundogdu [3] gives an Optimization of a four-degrees-of-freedom quarter car seat and suspension system using genetic algorithms to determine a set of parameters to achieve the best performance of the driver. Alkhatib et al. [4] used genetic algorithm (GA) method, and applied to the optimization problem of a linear one degree-of-freedom (1-DOF) vibration isolator mount and the method is keep going to the optimization of a linear quarter car suspension model. He found an most advantageous relationship between the root mean square (RMS) of the absolute acceleration and the RMS of the relative displacement. Baumal et al. [5] with the use of Genetic Algorithm have to find out the active control and passive mechanical parameters of a suspension system. The seat accelerations have been minimized using the GA and within the constraints of the system.

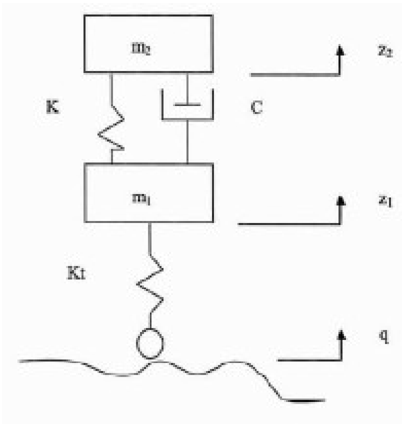

Fig. 1 Quarter Car Model

The mathematical modeling is based on transform basis which contains Fourier transform.

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

Where,

$$M_1\ddot{Z}_1 + C(\dot{Z}_1 - \dot{Z}_2) + K(Z_1 - Z_2) + K_t(Z1 - q) = 0$$

Taking the Fourier transfer on both sides

$$Z_2(-\omega^2 m_2$$

Where, ω = Natural Frequency, $j = \sqrt{-1}$

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

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

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, λ = Eigen Value

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

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

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:

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

The dynamic Tire load is defined as

$G = (m_1 + m_2)g = m_1(\mu + 1)g$ where g is gravitational acceleration. Thus the amplitude ratio between the relative dynamic Tire load, $\left|\frac{F_d}{G}\right|$ and the road input, q, becomes

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

Road irregularity or unevenness represents the main disturbing source for either the rider or vehicle structure itself.

As a function of time, the road conditions are given by

$$Q_1 = \begin{cases} \frac{h}{2}(1 - \sin\omega t) \\ \end{cases}$$

Where t= Time la

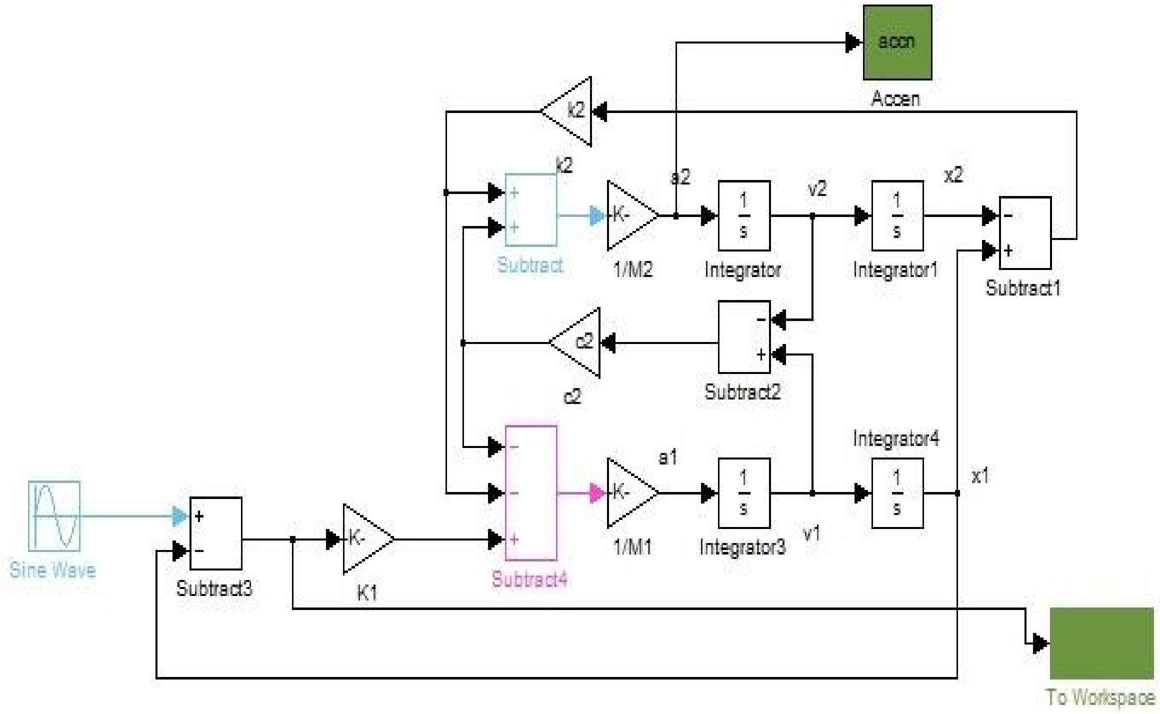

Fig. 2: Simulink Model

The simulink results are obtained by suspension parameters as follows:

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

Table 1: Fix Parameters

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

Table 2: Variable Parameters

A. Plot for Seat Acceleration

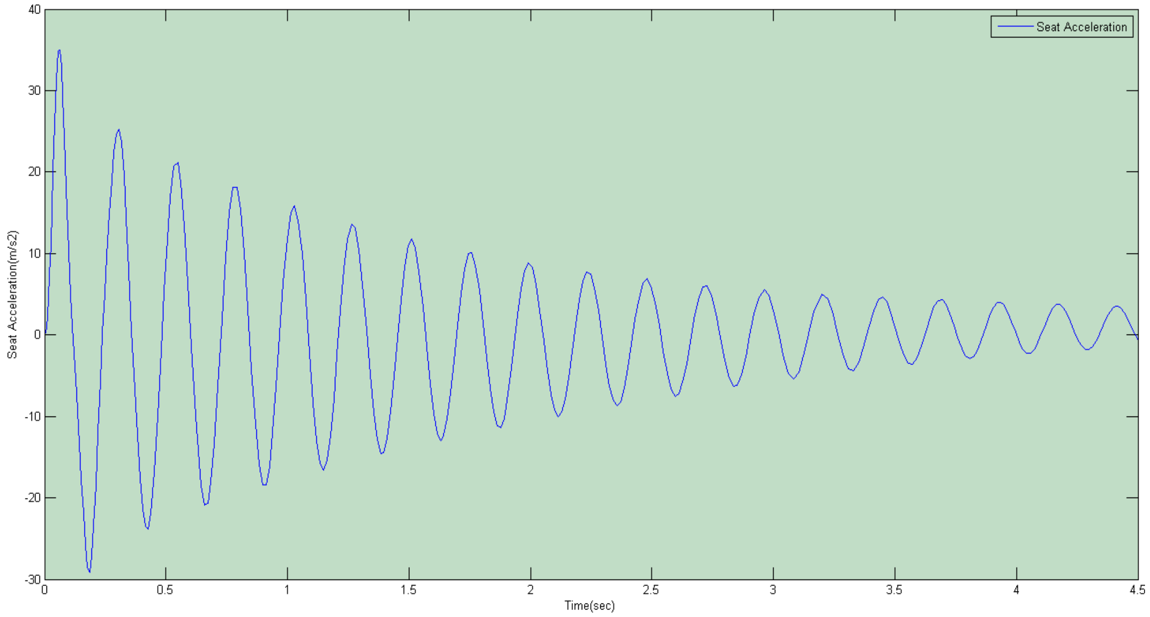

Fig. 3 Graph 1: Seat Acceleration

B. Plot for Tire Deflection

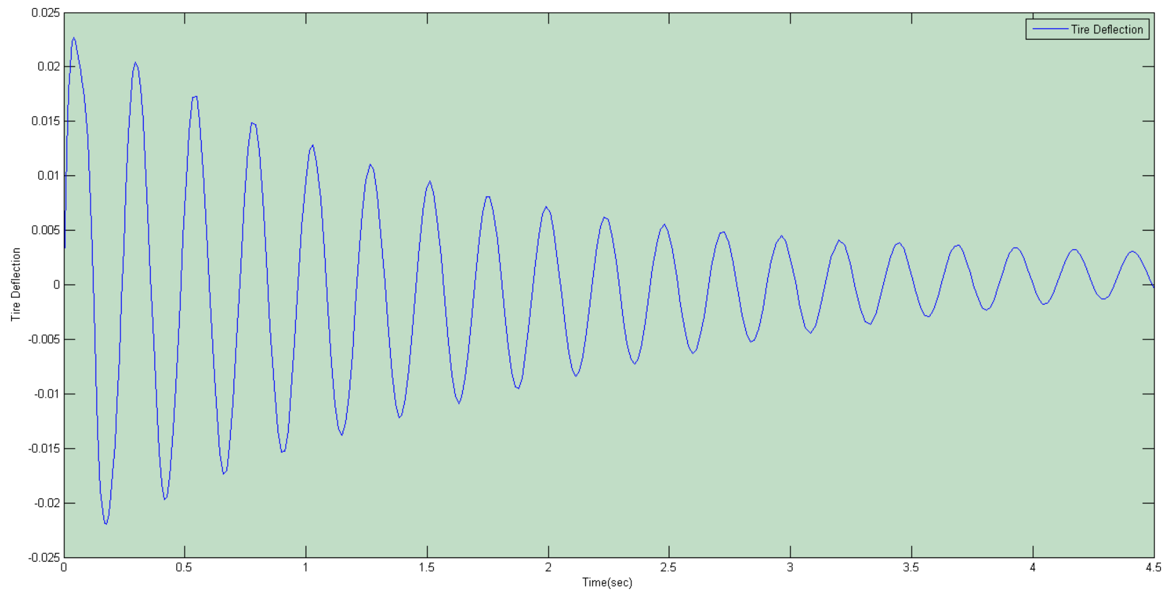

Fig. 4 Graph 2: Tire Deflection

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

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

Table 3: Observed Values

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

1) Min (TD) and min (Seat Acceleration)

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

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

Table 4: Optimized Parameters

C. Plot for Seat Acceleration

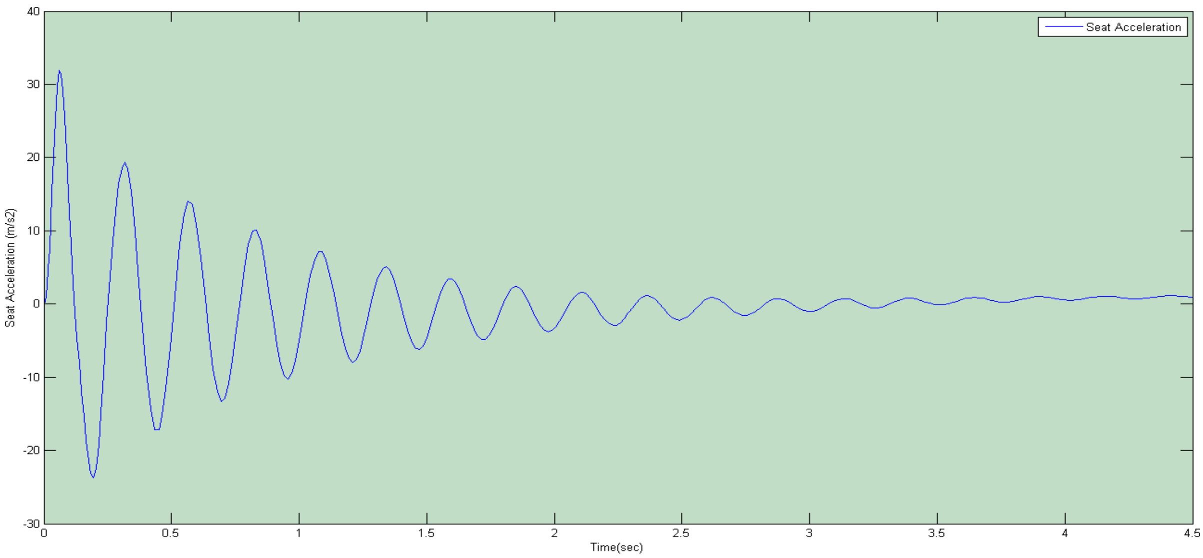

Fig. 5 Graph 3 Seat Acceleration vs. Time

Plot for Tire Deflection

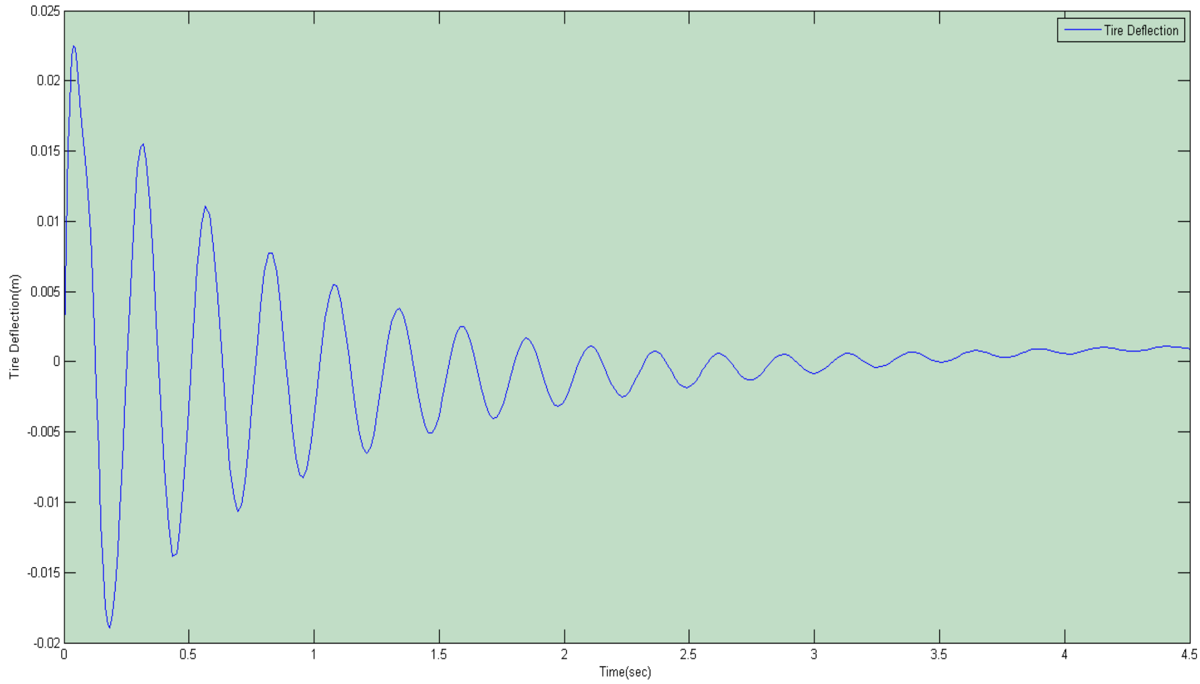

Fig. 6 Graph 4: Tire Deflection vs. Time

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

Table 5: Maximum and Minimum values of the Seat Accl^n & TD

I. EXPERIMENTATION SETUP AND RESULTS

Fig. 7: Quarter Car Test Rig

The Test rig consist of rigid frame at the bottom and which supports reaction frame. The reaction frame mounts the head of strut and one end to upright. The exciter is at bottom will give the excitation with help of compressed air. The flow air is to controlled by digital timer. The data acquisition consisting of the sprung mass

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Acceleration and unsprung mass acceleration were measured by accelerometers. The accelerometer and HMI (Human machine interface) were Delta Electronics make.

A. Comparison for optimized and Non-Optimized Values

1. Plot for Tire Deflection

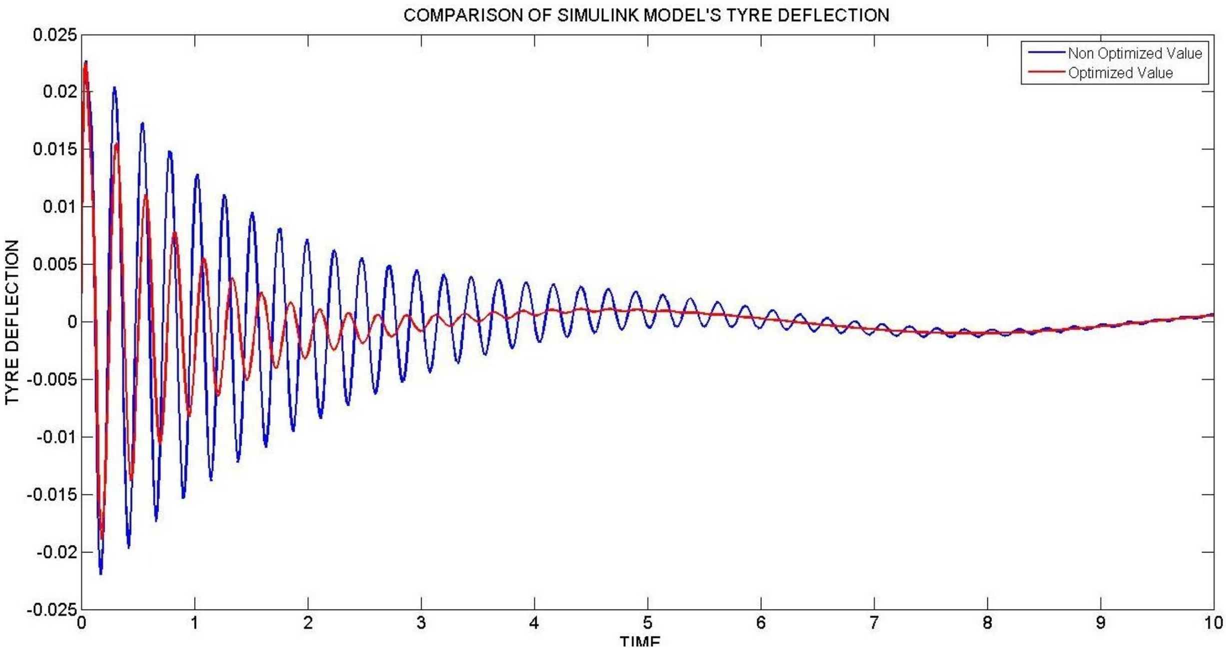

Fig. 8: Graph 5: Comparison of Tire Deflection

2. Plot for Seat Acceleration

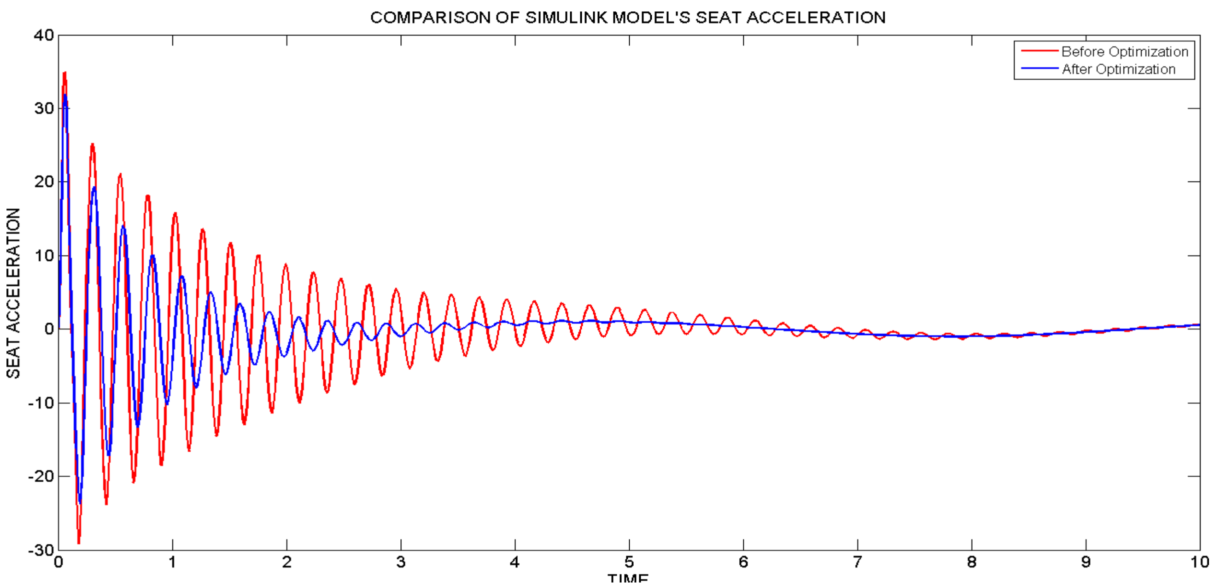

Fig. 9 Graph 7: Comparison of Seat Acceleration

The above graphs shows the pattern showed tire dynamic force considerably decreased in 8.3 % and Seat acceleration by 8.9 %.The result showing the greater sign of improvement in segment.

Parameter	Classical Value	Optimized Value	Experimental Value
Seat Acceleration (m/s^2)	34.9176	31.8604	32.0121
Tire Deflection (m)	0.023	0.022	0.021

Table.6: Results comparison of Optimized Values (Experimentation with Classical Values)

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.

Parameter	Initial Value	Optimized Value
Spring Stiffness(N/m)	18760	12236
Damping Coefficient (N-s/m)	900	536

Table 7: Comparison of Initial and Final Values of Strut There is a decrease in the spring stiffness which would ultimately reduce the production cost of the strut. The damping coefficient has been increased by some extent but it would be easy to control it.

1) Variation in Suspension Spring Stiffness:

Simulation shows that as suspension spring stiffness rises, the vertical acceleration rises. From results it can be proved that 34.62% vary in the stiffness changes vertical acceleration by 8.32%.It does not mean that we should use less stiffness spring. That causes increase in flexibility. Thus it should be such that it will minimize vertical acceleration as well as should give the stability.

2) Variation in the Damping Coefficient:

The damping coefficient damps the amplitude suddenly and brings system to the mean position. The damping coefficient is reduced by 40.4%.

3) Tire stiffness:

Tire stiffness is dependent of the pressure in the Tire. The low pressure in the tire should be

maintained. At that air pressure the particular stiffness should be considered. As tire air pressure rises the rigidity of tire and transfers the vibrations. Thus, tire pressure increases the stiffness increases and vertical acceleration also goes on increasing.

The spring stiffness value has been reduced from 18760 (N/m) to 12236 (N/m) It will have direct effect on cost reduction for manufacturing the spring. Thus overall performance can be increased in ride comfort in terms.

IV. CONCLUSION

The Optimization results show considerable difference in Seat Acceleration and Tire Dynamic Force. The Genetic Algorithm tool is more reliable. The results of Simulation and experimental analysis of quarter car passive suspension are quite similar because experimental model contains inherent accuracies, and the optimization tool used was more reliable with the experimental results.

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