Optimization of suspension damping parameters of a crossover SUV for ride comfort using iterative algorithms

Krishnakumar Pandit, Kirthan L J*

Dept. of Mechanical Engineering, RV College of Engineering, Bengaluru

Abstract

The main role of a damper in a suspension system is to dissipate the oscillation energy and stabilize the system. This paper reports the results of optimization of damping coefficient of a suspension system with fixed spring stiffness and geometry uisng iterative algorithms. A full scale model of the vehicle was developed using Solidworks 2020 and Matlab R2021a. Multibody simulation with an impulse test and two-time domain dynamic tests were modelled using Simscape Multibody option. The algorithms were used to simulate the suspension system and iteratively optimize the root mean square vertical acceleration of the vehicle for ride comfort. The algorithms minimised the RMS acceleration values for the dynamic tests from 7.67 to 5.56 m/s^2 (27.48%) for the sinusoidal track and from 10.34 to 6.076 m/s^2 (41.2%) for the twist track. However, for the impulse test, the acceleration slightly increased from 5.866 to 6.1372 m/s^2 (4.6%)

Keywords: *SUV, passive damping, suspension system, optimization, iterative algorithm*

1.0 Introduction

With growing demand for crossover SUV style vehicles throughout the world, the automobile industry is shifting towards comfort vehicles. Active damping set-up can prove expensive to implement due to the rising cost of manufacturing, Thus, passive damped suspension set-up has to be optimized with emphasis on ride comfort of the vehicle. Effect of vibration on the human body and ride comfort of a vehicle has been extensively researched and ISO 2631 standards were established, which specify that ride comfort of a vehicle is closely linked with vertical acceleration of the vehicle [1]. G A Hassaan [2] built a quarter car model and simulated it with road humps for the sake of maintaining ride comfort for the passengers using Matlab. The results showed that the vehicle comfort depends directly on the speed of the vehicle and damping coefficient values of the suspension set-up. A R Bhise et al. (2016) simulated a quarter

**Mail address: Kirthan L.J, Associate Professor, Department Mechanical Engineering, RV College of Engineering, Bengaluru – 59*
Email: kirthanlj@rvce.edu.in, Ph: 9632340386

car model to optimize an active suspension system using road profiles like half sine waves and step inputs [3]. Balasaheb B.A et al reported the best performance of suspension set-up when designed for slight underdamping [4]. Marzbanrad et al. [5] reported that the suspension of the vehicle to be as soft as possible with the front being slightly stiffer to avoid excessive pitching motion of the vehicle. They also stated that the ride comfort when optimized for one type of road profile may not provide the optimum result for other road profiles. Research works related to using optimization algorithms for setting up suspension systems have been carried out extensively. Shirahatti et al presented an optimal design of a suspension system which utilized a genetic algorithm and simulated annealing techniques to optimize multiple suspension parameters [6]. Sancibrian et al improved the kinematic characteristics of a double wishbone suspension system using a multi-objective optimization algorithm [7]. K Hemanth et al. simulated a quarter car model and using a skyhook control algorithm minimized the root mean square acceleration of the vertical motion of the vehicle fitted with a magneto-rheological damper [8].

From the review of literature [1-8], it was found that the full-car model for analysis and optimization of suspension parameters is not fully researched. The full-car model has more parameters of pitch and roll that affect the response of the suspension system giving it an edge over a quarter car model in teRMS of realism. The stiffness parameters in the studies were varied independently and its effect on the load capacity of the vehicle, ground clearance and other factors were not considered in most studies which affects the real-world operation of vehicles. Passive suspension studies using modern simulation technologies were also found to be lacking in research. This research was aimed at optimizing the passive damping properties without affecting the suspension stiffness using a full-car model of the vehicle using Multibody simulation approach.

2.0 Modelling the vehicle suspension system

Crossover SUVs being one of the highest selling types of vehicles in the world, a model of a typical crossover SUV inspired from the weight and overall dimensions of Mahindra’s XUV500 was considered. MacPherson type front suspension and multi-link rear suspension was modelled on a skateboard type chassis as seen in Fig. 1. Solidworks was used to model the suspension system of the vehicle. Each component was assigned inertial properties by specifying the material as shown in Table 1. The model was transferred to Matlab Simscape Multibody for multibody simulations and further optimization.

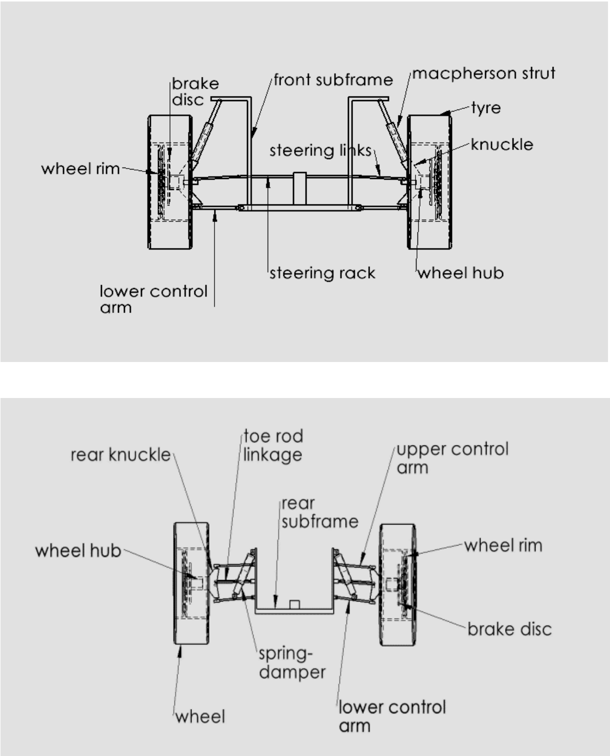

Fig. 1. a) MacPherson front suspension and b) Multilink rear suspension

Table 1. Material selection for Components

Material	Components
Plain carbon steel	front subframe, front knuckle, steering rack, wheel rims, macpherson strut, brake discs, wheel hubs, rear sub frame, rear knuckle, rear struts
Aluminium 6063-T6	front lower control arm, steering linkages, rear lower control arm, rear upper control arm, rear toe links

2.1 Initializing suspension variables

The vehicle was given a translatory degree of freedom in y-axis (perpendicular to the ground), and two rotational degrees of freedom in x and z directions (pitch and rolling motions respectively). A test bed for each wheel of the vehicle was modelled in Simscape Multibody and contact was established using the sphere to plane contact block found in Matlab contact forces library [9]. The weight distribution of the vehicle was set up using point loads on the front and rear subframes of the vehicle leading to a final mass of 1840 kg and a weight distribution of 60:40 front to rear respectively. The suspension parameters were initialized by dropping the vehicle on the stationary test platfoRMS, observing the equilibrium pitch and roll values and manually trying out stiffness values until the equilibrium pitch and roll values were zero. The final values of the suspension parameters are given in table 2. The vehicle model after initialization is as shown in Fig. 2.

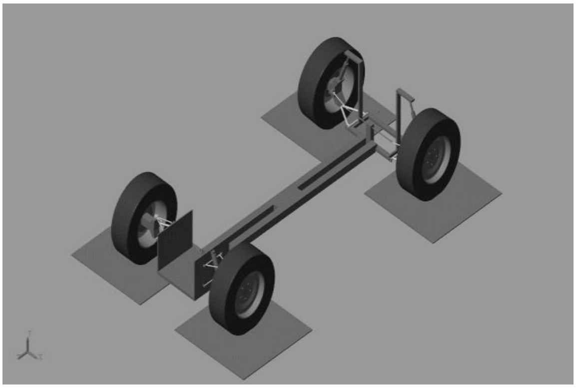

Fig. 2. complete model of the vehicle in Simscape Multibody of Matlab

Table 2. Suspension parameters

Parameters	Value
Front shock stiffness (N/m)	1.59×10^5
Front shock damping coefficient (Ns/m)	1×10^4
Rear shock stiffness (N/m)	6×10^5
Rear shock damping coefficient (Ns/m)	1×10^4
Tyre-plane contact stiffness (N/m)	1×10^6
Tyre-plane contact damping (Ns/m)	1200

2.2 Test simulation parameters and road profiles

Three test cases were modelled to simulate the suspension system out of which two were dynamic tests that involved an input of road profile. A soft suspension with minimal damping perfoRMS better in dynamic tests but fails in impulse tests and vice versa for a hard suspension. The goal of these tests will be to find the most comfortable suspension that perfoRMS optimally in all three tests. The inputs of road profiles were given to the test platfoRMS on which the wheels of the vehicle were supported. First was the impulse test where the vehicle was dropped from a height of 500mm onto the test platfoRMS. The second test was a constant amplitude sinusoidal input to the test platfoRMS. The test simulates a test track with four bumps per metre of test track modelled using absolute values of sine wave as the input. The frequency and phase were calculated based on the velocity of the vehicle and the wheelbase of the vehicle respectively as shown in Table 3. For test purposes, a velocity of 10 km/hr was selected. The third test included a twist track where each of the wheels experience a different input at a particular point of time, commonly used in the industry for durability analysis of vehicle components. The test is named twist track as the vehicle observes a twisting force as it traverses the test track. This track was simulated using square wave inputs to each wheel taking into consideration a vehicle velocity of 10 km/hr. The details of the inputs are as shown in Table 4.

The tracks were simulated for a period of 5s in this study. Vehicle data such as vertical motion of the body of the vehicle, pitch, roll, vertical acceleration were recorded. The impulse test took the least amount of time and the twist track took the largest amount of tie to simulate.

Table 3. Constant amplitude sinusoid parameters

	Front wheels	**Rear wheels**
Input function	$f = A*\sin(\omega t)$	$f = A*\sin(\omega t+\Phi)$
Amplitude (A)	10 cm	10cm
frequency (ω)	1.745278*v rad/s	1.745278*v rad/s
phase (Φ)	0 rad	16.9641 rad

Table 4. Twist track- square wave input parameters

	Wheel 1	**Wheel 2**	**Wheel 3**	**Wheel4**
Amplitude (m)	1	1	1	1
Time period (s)	1.8/v	1.8/v	1.8/v	1.8/v
Pulse width (% of period)	50	50	50	50
Phase delay (s)	0.1	0.15	4.86/v+0.1	4.86/v+0.15

2.3 Optimization Algorithms

The first iterative algorithm is an algorithm to find the critical damping coefficient for a given suspension set-up using the impulse test case. The impulse test was chosen for simulation as it was the fastest among the three tests to execute and as the algorithm is iterative, the simulation times can add up to a very slow algorithm for slow executing tests. The algorithm follows the ideology of the binary search algorithm where in the algorithm tries to find an element (optimum damping coefficient) in the given range of values. The values are set such that the lower limit corresponds to an underdamped oscillation scenario while the upper limit corresponds to an overdamped oscillation scenario, the logic being that the critical damping coefficient lies in the given range. During the execution of the algorithm, it also optimizes for damping coefficient values for which minimum RMS acceleration and minimum peak acceleration were obtained during the simulation.

A counter variable ‘count’ is declared, set to zero and is updated in every loop to reflect the loop number. Cmax, Cmin, min_peak, min_RMS, C_minpeak and C_minRMS variables are declared and set to their respective values. When the condition of maximum possible value of ‘count’ is met, execution stops. Cm variable (mean damping coefficient) is defined as the average of Cmax and Cmin variables. Front and rear damping coefficient values are set as Cm and simulations are run. The simulation results are stored in the form of a row matrix ‘Simout’. Simout variable has outputs from the simulation like displacement, acceleration, rate of change of pitch and roll etc. RMS_acc variable is used to store the root mean squared acceleration. In each loop a min_RMS variable is checked. If min_RMS is greater than RMS_acc value, then min_RMS variable is updated with the value of RMS_acc variable. Along with it, C_minRMS variable is also updated with the value of the Cm variable. Similarly, minimum peak acceleration is checked and compared in each loop. Variable M is defined as the 1st peak of the curve. Variable eq is defined as the last data in the array Simout.d. If first peak(M) is greater than equilibrium value (eq), then Cmin value is set to Cm (mean value). If first peak(M) is lesser than equilibrium value (eq), then Cmax value is set to Cm (mean value).

From the execution of the critical damping coefficient algorithm, it was observed that the value of minimum peak acceleration of the impulse test case resulted in minimum RMS acceleration for the other two dynamic tests, thereby providing the most comfortable ride. To further optimize the damping coefficient for minimum peak acceleration during impulse test, another algorithm, referred to as refinement algorithm, was developed. The algorithm chose the damping coefficient value corresponding to the minimum peak acceleration value obtained from the first algorithm and searched for better values within a small range of values in the vicinity of it. This was because it was observed that as the values increased or decreased from the damping coefficient corresponding to minimum peak acceleration value, the peak acceleration increased implying that there must be one value for which the peak is minimum.

3.0 Results and Discussion

3.1 Responses from the critical damping algorithm

The critical damping algorithm was run for ten iterations for the impulse test for 5 sec simulation time. The results of the execution are as shown in Table 5. Equilibrium value is independent of the value of damping coefficient. The algorithm very quickly narrows down the search for the critical damping coefficient. By loop number 6, it was observed that the 1st peak in displacement (M) has almost reached the value of equilibrium displacement (eq). This means that the value Cm has almost reached the value of critical damping coefficient. The algorithm, after loop 6, is not as effective in narrowing down the critical damping coefficient as it has done till loop 6. The value keeps fluctuating between 16800±300 Ns/m. The critical damping coefficient was approximately 16900 Ns/m. The minimum value of RMS for the impulse test takes place in loop number 3 at damping coefficient at 21250 Ns/m with the RMS acceleration being 5.8664 m/s^2 and the corresponding peak acceleration being 5.1853 m/s^2. This value corresponds to a slightly overdamped case. The minimum value of peak acceleration for the impulse test takes place in loop number 4 at a damping coefficient of 15625 Ns/m with the peak acceleration being 1.6601 m/s^2 and the corresponding RMS acceleration being 6.1372 m/s^2. This value corresponds to a slightly underdamped condition. The minimum RMS test result has a higher peak value than the minimum peak result and yet maintains a lower RMS value. This is because the oscillations are overdamped in this case and the vibrations die down very quickly.

Table 5. Results of the critical damping coefficient algorithm for 10 iterations

Loop No.	1st peak displacement (m)	Equilibrium value (m)	Damping coefficient (Ns/m)	RMS acceleration (m/s^2)	Peak acceleration (m/s^2)
1	0.0461	0.0559	55000	6.0180	10.3811
2	0.0493	0.0559	32500	5.9948	9.8658
3	0.0532	0.0559	21250	5.8664	5.1853
4	0.0571	0.0560	15625	6.1372	1.6601
5	0.0548	0.0560	18438	6.3411	3.5425
6	0.0559	0.0560	17031	6.2964	2.6276
7	0.0564	0.0560	16328	6.0276	2.1533
8	0.0561	0.0560	16680	6.0250	2.3963
9	0.0560	0.0560	16855	6.3499	2.4922
10	0.0559	0.0560	16943	6.0346	2.5703

The values of damping coefficient corresponding to minimum peak acceleration and minimum RMS acceleration were used to simulate the vehicle with the other two dynamic loading tracks. The results of the test are shown in Table 6. The results show that damping corresponding to minimum peak acceleration values provide a lower value of RMS acceleration for the dynamic test tracks thus proving the necessity of a refinement algorithm to look for better values of damping coefficients with even lesser peak accelerations.

Table 6. Results for dynamic tracks with optimum values of damping coefficient

		min. peak acceleration damping (from algorithm), (m/s^2)	min. RMS acceleration damping (from algorithm), (m/s^2)
Constant Amplitude Sinusoid track	peak acceleration	5.93	5.7466
	RMS acceleration	7.67	6.3439
Twist track	peak acceleration	9.84	6.0950
	RMS acceleration	10.34	6.3017

3.2 Responses from the Refinement Algorithm

The refinement algorithm was run for 4 iterations for the impulse test for 5 sec simulation time. The results of the execution are plotted in Table 7. Even though further refinement might be possible in the loops after loop 4, it doesn't make practical sense to continue execution of the program as the divisions in the next loop will be less than 1 Ns/m leading to fractional differences in the damping coefficient values. Some of the peak acceleration readings have absurd values greater than 10 m/s^2. This is due to an error in which there is a very small spike in value during freefall of the vehicle which is recorded as the 2nd peak. In such cases the third peak becomes the proper value and is thus written in brackets. Even though the refinement from the 1st loop to 2nd loop is substantial, the later loops further refine the value to only -0.03 m/s^2 and 29 Ns/m. Thus the refinement can be stopped earlier if so desired. Using the refinement algorithm, minimum peak acceleration of the vehicle is decreased from 1.6601 to 0.5479 m/s^2. Thus an improvement of 67.03% is observed in the minimum acceleration values. Damping coefficient value is decreased from 15625 m to 14156 Ns/m which could only be possible due to this algorithm.

Table 7. Results for refinement algorithm

		Lower limit (Ns/m)	**Upper limit** (Ns/m)	**C_mean** (Ns/m)	**peak acceleration** (m/s^2)	**min_peak** (m/s^2)
Loop 1 (div=600)	div 1	14125	14725	14425	0.7700	0.7700
	div 2	14725	15325	15025	1.2158	
	div 3	15325	15925	15625	1.6601	
	div 4	15925	16525	16225	2.0780	
	div 5	16525	17125	16825	2.4890	
Loop 2 (div=120)	div 1	14125	14245	14185	0.5767	0.5767
	div 2	14245	14365	14305	10.06 (1.29)	
	div 3	14365	14485	14425	0.77	
	div 4	14485	14605	14545	0.8633	
	div 5	14605	14725	14665	0.9531	
Loop 3 (div=24)	div 1	14125	14149	14137	10.06(1.15)	
	div 2	14149	14173	14161	0.5559	0.5559
	div 3	14173	14197	14185	0.5767	
	div 4	14197	14221	14209	0.6048	
	div 5	14221	14245	14233	0.6136	
Loop 4 (div=4.8)	div 1	14149	14154	14151	10.06(1.17)	
	div 2	14154	14159	14156	0.5479	0.5479
	div 3	14159	14163	14161	0.5559	
	div 4	14163	14168	14166	0.5662	
	div 5	14168	14173	14171	10.06(1.18)	

Front and rear damping coefficients are assigned the values 10000 Ns/m (initial value) and 14156 Ns/m (optimized value) and tested on both constant amplitude sinusoid tracks and twist tracks. The results prior and post optimization from the refinement algorithm are as shown in Table 8.

Table 8. The results prior and post optimization from the refinement algorithm

Variables		C = 21250 Ns/m (initial value)	C = 14156 Ns/m (optimized value)
Sinusoid track	peak acceleration	5.93 m/s^2	5.66 m/s^2
	RMS acceleration	7.67 m/s^2	5.5621 m/s^2
Twist track	Peak acceleration	9.84 m/s^2	9.8 m/s^2
	RMS acceleration	10.34 m/s^2	6.0757 m/s^2

4.0 Conclusion

A generic crossover SUV suspension system was modelled with MacPherson front suspension and multilink rear suspension in Simscape Mutlibody of Matlab. Three test cases were considered for the simulation and a suspension parameters were optimized for all the three test cases using two iterative algorithms for a vehicle velocity of 10km/hr. The iterative algorithms used vertical displacements and the acceleration in the vertical direction as variables that were minimized. It was found that for a comfortable suspension with damping suited for most applications, the damping coefficient corresponds to a case of slightly underdamped oscillation scenario with values lesser than the critical damping coefficient. For the sinusoidal track, the algorithms minimised RMS acceleration values from 7.67 to 5.5621 m/s^2 (27.48%) and from 10.34 to 6.0757 m/s^2 (41.2%) for the twist track. However, while for the impulse test, the values increased only from 5.86 m/s^2 to 6.13 m/s^2 (4.6%).

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