Ding et al. 2016. *Int. J. Vehicle Structures & Systems*, 8(3), 126-130
ISSN: 0975-3060 (Print), 0975-3540 (Online)
doi: 10.4273/ijvss.8.3.02

International Journal of Vehicle Structures & Systems
Available online at www.maftree.org/eja

Simulation and Experimental Research on Vehicle Ride Comfort and Suspension Parameters Optimisation

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ABSTRACT:

With the development of vehicle technology and the improvement of living standards, people's ride comfort requirements for vehicle are also increasing. Especially for commercial passenger vehicles, the ride comfort is related to physical and mental health of passengers. Since the suspension is a major chassis system that affects the ride comfort of vehicle, so for how to make the vehicle to maintain good ride in a variety of driving conditions, the design and improvement of suspension are essential. The research content of this paper is simulating and optimizing the suspension parameters based on kinetic model of vehicle. First, a kinetic model of vehicle for minibus system is established. Then, test verification is carried out for the ride comfort of vehicle model. The results show that the established vehicle model can be used for simulation and optimization of front and rear suspension systems through the application of genetic algorithm.

KEYWORDS:

Kinetic model; Suspension system; Ride comfort; Test optimization; Genetic algorithm

CITATION:

Z. Ding and Z. Lei. 2016. Simulation and Experimental Research on Vehicle Ride Comfort and Suspension Parameters Optimisation, *Int. J. Vehicle Structures & Systems*, 8(3), 126-130. doi:10.4273/ijvss.8.3.02

1. Introduction

Vehicle suspension system is a major system that affects the vehicle ride comfort and handling stability. Its main role is to transmit the force and moment of force between vehicle and road surface. How to make a vehicle remain good for bodywork and small internal vibrations in a variety of road and driving conditions has been a pursuing goal for suspension system engineers. Good suspension system not only makes vehicle with better handling and stability in a variety of driving conditions, but also can effectively reduce different spectrum excitation from the road surface, so as to make the vehicle for a good ride comfort [1]. The optimization of suspension performance also needs the help of a virtual prototype model of the vehicle. At the same time, the accurate acquisition of parameters that the model requires is the basis for the establishment of a correct model, and also is the key to virtually match optimization results [2].

Therefore, the objective of this work is to precede the test of mechanical properties on the tire which has a significant impact on vehicle performance, and obtain tire parameters through parameter identification. Second objective is to test the inertial parameters of the mass of the whole vehicle, then establish a multi-body kinetic model of the vehicle to virtually optimize its front and rear suspension systems. Due to the advantages and disadvantages of suspension performance will simultaneously affect handling stability and ride comfort of the vehicle. Hence, both the performance indicators are the targets for satisfactory optimization results.

2. Kinetic modeling of vehicle system

The kinetic modeling of vehicle system starts with the following steps [3]:

1) Simplify the bodywork to a rigid body, focus the quality to the chassis, and modify the vehicle mass by changing the quality of the bodywork;
2) Simplify the power train to a rigid body, and focus the quality to the chassis;
3) Consider the elastic properties of the tire, and regard the specific components (such as steering knuckle, lower arm) as a rigid body in the suspension system modeling.

After simplification, the vehicle system model includes five parts [4]: chassis model (including vehicle model), power train model, steering model, front and rear suspension model, and tire model. The vehicle models before and after the suspension system, steering system, power system, braking system, tires and bodywork are established with the module of Adams-car software. A multi-body kinetic model assembly of the vehicle is shown in Fig. 1. The accuracy of vehicle kinetic model established by Adams software directly affects the subsequent optimization of suspension system, so a ride comfort test is needed for vehicle kinetic model verification. It includes pulse input test and random input test. The vibration response peak of the vehicle under pulse input is an important indicator to evaluate vehicle ride comfort. According to the provisions for the pulse test in GB/T 4970-2009 “Test Method for Vehicle Ride Comfort”, the triangular bump is used as a pulse input stimulus [5]. After selection, the root of seat bracket above suspension and driver’s seat track are selected as the response measuring points, as shown in Fig. 2. In pulse test, the vehicles go through a triangular bump in constant speed of 20 km/h and 50 km/h respectively, as shown in Fig. 3.

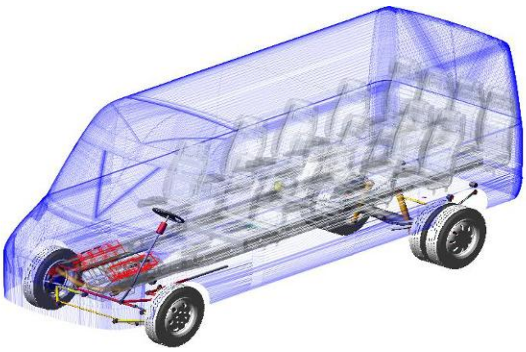

Fig. 1: Kinetic model of the vehicle

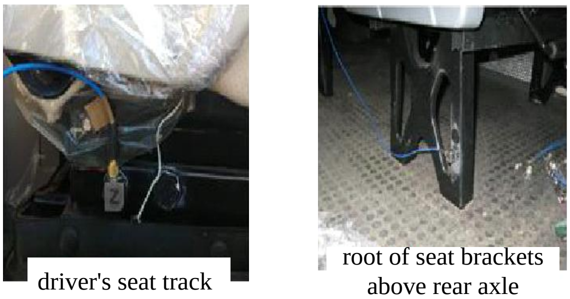

Fig. 2: Pulse test acceleration measuring points

Fig. 3: Pulse test

Comparative results of impulse test data and simulation data for the root of seat brackets of rear suspension are shown in Fig. 4 and Fig. 5 respectively. The simulation data and experimental data match well, besides, the magnitude of vibration damping and period are also coincide with the experimental data. According to the provisions for the random road test in GB/T 4970-2009 "Test Method for Vehicle Ride Comfort", the selection of test measuring point is the same with the impulse test, respectively. Comparisons of the acceleration power spectral density (PSD) at corresponding speed of 50 km/h and 110 km/h between the tested one and simulated one are shown in Fig. 6 and Fig. 7. The simulation data match the experimental data well. The results of pulse and random road tests show that the vehicle model can correctly reflect all the suspension system data in time and frequency domains.

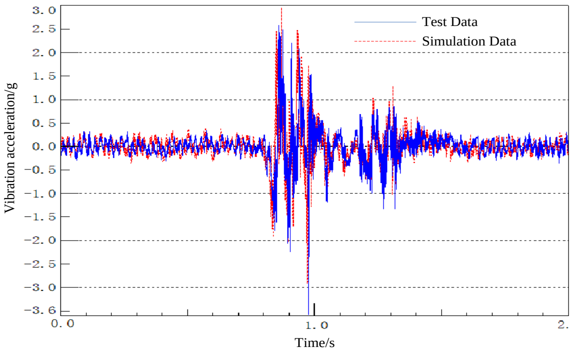

Fig. 4: Vibration acceleration curve comparison at 20 km/h

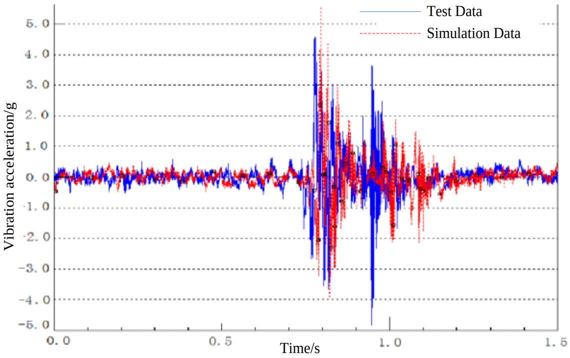

Fig. 5: Vibration acceleration curve comparison at 50 km/h

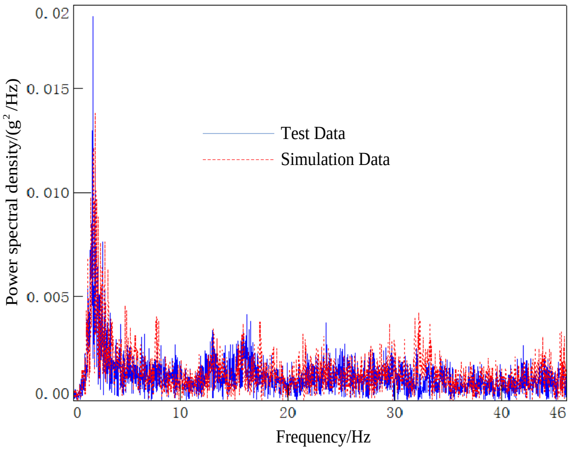

Fig. 6: Vibration acceleration PSD curve comparison at 50 km/h

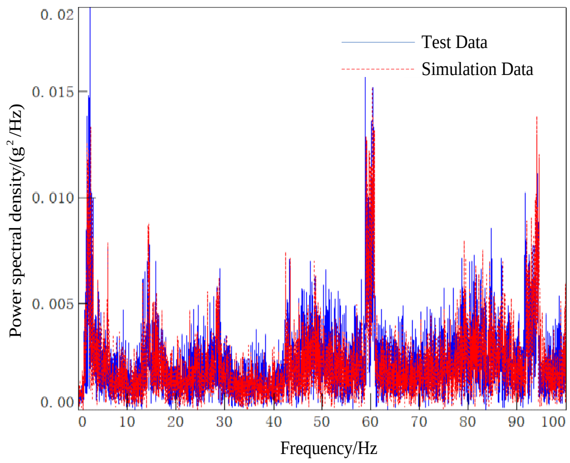

Fig. 7: Vibration acceleration PSD curve comparison at 110 km/h

3. Simulation optimization

For the optimization of suspension system, genetic algorithm is selected herein. Genetic algorithm is a random global search and optimization method developed by natural biological evolution mechanism. It re-composes good adaptive "chromosome" and generate new groups through organized and random information exchanges. It is an efficient, parallel and global search method that can automatically acquire and accumulate information about the search space in the search process, and adaptively control the search process to obtain Pareto sets [6]. In this paper, a classic genetic algorithm NSGA-II is used for multi-objective optimization of front and rear suspension system parameters. This method uses parallel selection so that the entire evolutionary population is evenly distributed in constraint space. Elitist is introduced and Pareto optimal individuals are reserved so that they are forbidden to involve crossover or mutation operation but directly retain to next sub-populations. Sharing function is used to limit the same or similar individuals so as to produce more different optimal solution. Coverage of two sets is used to measure the advantages and disadvantages of Pareto concentrated solutions [7-8]. Fig. 8 is a basic flow of genetic optimization algorithm. Gen represents the evolution generation. Maxgen represents the set maximum generation.

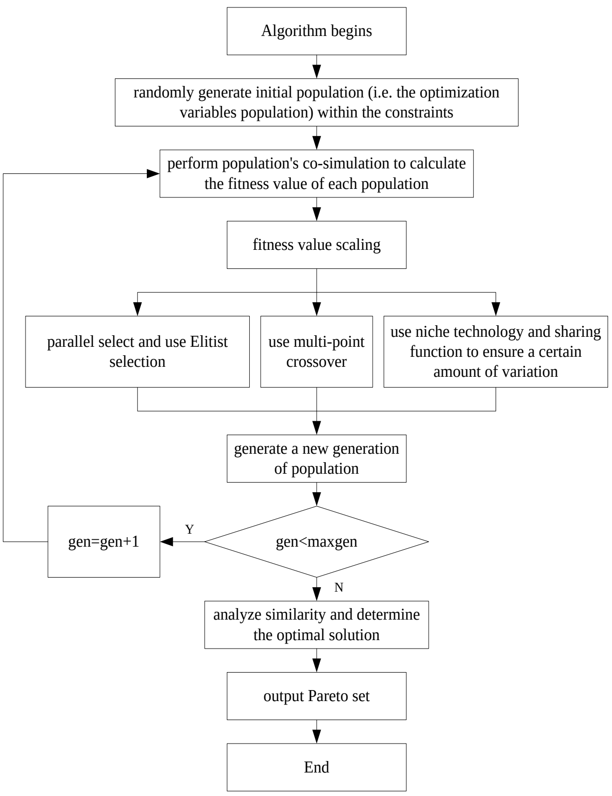

Fig. 8: Optimisation algorithm flowchart

According to the process of genetic algorithm, Matlab is used to optimize the process, combining with vehicle virtual prototype model established by Adams to match and optimize the front and rear suspension [9-10]. In match optimization, it is required to set optimization variables in advance. It mainly considers the variables which have an important impact on the ride comfort and handling stability of the vehicle, and they can be selected according to the actual situation or emphasis. The optimization design variables in this paper include: torsional stiffness of front suspension torsion bar T_1, primary stiffness of rear suspension K_1, composite total stiffness K_2, torsional stiffness of front and rear stabilizer bars T_2 and T_3, damping characteristic curve coefficients of front and rear shock absorbers C_1 and C_2, bush radial stiffness of leaf spring K_3. Thus, optimization variables can be expressed as:

$$X=\{K_1,K_2,K_3,T_1,T_2,T_3,C_1,C_2\} \tag{1}$$

According to the engineering judgement and vehicles configurations, the traditional matching method is used to determine the approximate variation range of design variables K_1, K_2, K_3; T_1, T_2 and T_3. Their proper selection is of great importance for the entire match optimization, and affects the optimization duration and the accuracy of the results.

C_1 and C_2 are the variable coefficients optimized on the basis of existing damping characteristic curve. The variation range of C_1 and C_2 is determined to be 0.6 - 1.8. For the selection of objective function, no-load and full-load conditions should be considered. For the target ride comfort, the z acceleration average root values a_{fz1}, a_{fz2}, a_{rz1} and a_{rz2} of corresponding position above the front and rear suspension that most directly react the suspension are selected. For handling stability target, relevant indicators of two national standard tests are examined here. Because the steady-state rotation has "veto power" of stability control, the maximum lateral acceleration the vehicle under steady-state rotation a_{y1} and a_{y2} are selected. In order to facilitate the optimization programming, A and B are taken as the opposite number of absolute value of the actually measured acceleration values. The second is the average yaw angular velocity r_1, r_2, and the average vehicle roll angle φ_1, φ_2 of hunting test under the reference speed. The objective function is:

$$\min F(X)=\left\{\begin{array}{l} a_{fz1}+w_1 a_{fz2}, a_{rz1}+w_2 a_{rz2}, \\ a_{y1}+w_3 a_{y2}, r_1+w_4 r_2 \\ \varphi_1+w_5\varphi_2 \end{array}\right\} \tag{2}$$

In Eqn. (1), subscripts with 1 are the targets in no-load condition, subscripts with 2 are the targets in full-load condition, and w_1, w_2, w_3, w_4 and w_5 are weight coefficients.

The fitness function uses direct design method, which is negative of objective function that is given by

$$Fit(f(x))=-\min F(X) \tag{3}$$

In order to make the simulation optimization operates reliably and reduces the optimization time, some constraints are added here. The constraints are mainly determined according to the actual situation of the optimized problem so as to appropriately facilitate the constraints of optimization process as:

1) According to the target of vehicle ride comfort, front and rear measuring point weighted acceleration values a_{fz1}, a_{fz2}, a_{rz1} and a_{rz2} when driving on B-class road with constant speed of 80

km/h under the condition of no-load and full-load are required to be less than or equal to 0.6g.

2) In reference to speed hunting test, vehicle body roll angle φ1 and φ2 under the condition of no-load and full-load are required to be less than or equal to 6°.

4. Comparison of optimization results

Under the premise to ensure the optimization effects, in order to reduce the simulation optimization time, the size of the initial population selected in this paper is 500, evolutionary generation is 60, optimal reservation is 10, the proportion of cross offspring is 0.85, optimal individual coefficient of Pareto front-end is 0.3, and weight coefficient $w_1 = w_2 = w_3 = w_4 = w_5 = 1.5$. Optimization variable range is determined according to the actual condition of the vehicles.

The lower limit is:

$$X_{\min} = \begin{Bmatrix} 60,130,1*10^3,7*10^4, \\ 2*10^5,2*10^5,0.6,0.6 \end{Bmatrix} \tag{4}$$

The upper limit is:

$$X_{\max} = \begin{Bmatrix} 85,180,5*10^3,1.5*10^5, \\ 9*10^5,9*10^5,1.8,1.8 \end{Bmatrix} \tag{5}$$

The units of these variables and units of Adams model should be consistent so as to facilitate the calculation of the objective function. Through virtual match optimization, Pareto solution set and its corresponding Pareto front end can be obtained to select a group of better results that focuses on ride comfort as:

$$\begin{aligned} & K_1 = 73.5N/mm,\ K_2 = 151.5N/mm, \\ & K_3 = 3.5*10^3 N/mm \\ & T_1 = 8.8*10^4 N \cdot mm/\deg, \\ & T_2 = 6.8*10^5 N \cdot mm/\deg, \\ & T_3 = 8.2*10^5 N \cdot mm/\deg \\ & C_1 = 1.12,\ C_2 = 0.88 \end{aligned} \tag{6}$$

Therefore, the optimization results can be re-entered to the vehicle model for simulation. Due to the limited space, only comparison of simulation optimization data of handling stability before and after optimization is provided here.

Comparison of lateral acceleration in steady state is shown in Fig. 9. The maximum lateral acceleration of optimized vehicle can be achieved has increased, indicating that its steady circular ability is better. Comparison of yaw rate in hunting test is shown in Fig. 10. The yaw rate of optimized vehicle is significantly reduced, indicating that its controllability and stability are increased. Finally, a routine performance check of vehicle suspension is carried out on optimization results data, such as front and rear suspension offset frequency and ratio, front and rear damping ratio, front and rear roll angle stiffness and ratio, front and rear roll center height, variation of full-load vehicle centroid, variation of full-load wheel route, adjustability and reliability check of full-load wheel alignment parameters, and so on. If these are in line with the design requirements, the optimization program can be implemented.

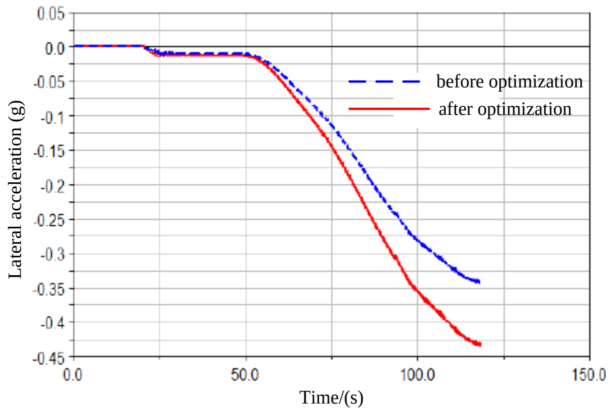

Fig. 9: Comparison of lateral acceleration in steady circular

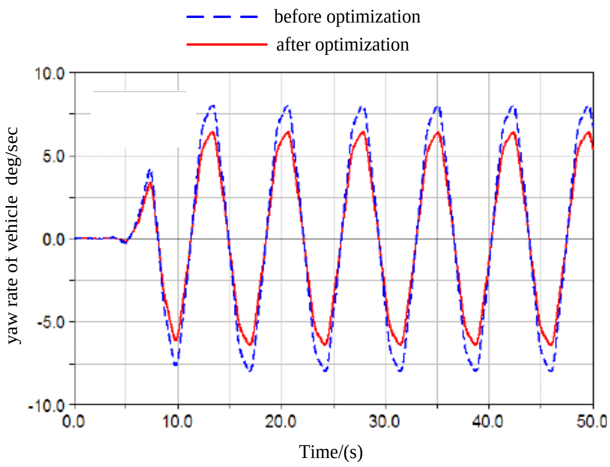

Fig. 10: Comparison of yaw rate in hunting test

5. Conclusion

In order to establish an accurate kinetic model of vehicle and optimisation of vehicle suspension system parameters, this study firstly established kinetic models of each subsystem of minibus, and then established a kinetic model of vehicle. An experimental verification is carried out for the accuracy of the vehicle kinetic model on the perspective of ride comfort. The results show that the established vehicle model can be used for the simulation optimization of front and rear suspension systems. Finally, multi-objective genetic optimization algorithm was used to virtually optimize the front and rear suspension system parameters. The obtained optimization results have shown that the ride comfort of the vehicle has been significantly improved.

ACKNOWLEDGEMENTS:

The National Natural Science Fund: Mechanism and control of removal SO_2/NOx/Hg from oxyfuel combustion flue gas base on CO_2 compression liquefaction process (No. 51566004).

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