Study of the Vehicle Controllability and Stability Based on Multi-body System Dynamics

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Abstract: In this paper, the 135 Degree of Freedom multi-body system dynamics model was built by using the software ADAMS/CAR, according to the requests of Chinese standards, and the simulating research about 6 performances of automotive controllability and stability were carried on. Based on the simulating results, some automobile's performance, such as return-ability, slalom-ability and steering efforts-ability, were excellent, but some other performances, such as steady state cornering ability, steering wheel angle pulse input response ability and steering wheel angle step input response ability, were not satisfied. In order to improve the performance of the automobile, three parameters, i.e. the automotive mass, the load of front axis and the torsion stiffness of rear stabilizer anti-roll bar, were selected as the optimized objects. Within the variety range of the parameters, the multi-body system of the automobile was optimized. Automotive controllability and stability is improved obviously based on the simulating results of the optimized multi-body system.

Keywords: Controllability and Stability, multi-body system dynamics, optimize.

1. INTRODUCTION

The controllability and stability of the automobile refers to ability that the automobile is able to run following the way given by the driver through steering system and steering wheel when the driver doesn't feel excessively tension and fatigue, and the ability of resisting interference and keeping traveling stably when encountered outside interferences. The controllability and stability of the automobile is very important to the high-speed vehicles because it can not only affects the manipulation of convenience, but is also a major performance that decide the safety of the high-speed vehicles. The controllability and stability of the automobile has received increasing at-tention and became a major indicator to measure the modern vehicles [1].

The most influential factors to the Controllability and Stability of the vehicles include vehicle quality, vehicle location of the center of mass [2, 3] Steering System elastic characteristics [4], wheel alignment parameters [5], the wheel cornering characteristics [6] Suspension roll stiffness [7, 8], etc. This paper build the 135 Degree of Freedom multi-body system dynamics model, and the simulating research about 6 performances of automotive controllability and stability were carried on by the software ADAMS/CAR. According to the requests of Chinese standards, this paper uses the large-scale Comprehensive Assessment scores of the Steady Rotary test, the step input test and the Steering Wheel Angle Pulse Input test as the objective function, evaluating and registering the simulating results of each tests of the Controllability and Stability, and finally, optimization the parameters of vehicle quality, front axle load, the behind horizontal stabilizer torsional stiffness.

2. THE MODELING OF MULTI-VEHICLE SYSTEM DYNAMICS

The vehicle is a system which is very complicated. When we use software to simulation analysis it, we can't take into account each parts, because if we do so, the modeling process would become extremely complicated, and the simulation solution time would be extended greatly, or we would even not able to get the results of simulation analysis. So we should neglect and simplified the parts which has little impact in measurement target when modeling. When establishing the vehicle model, this paper simplified the vehicle system as follows:

1. In addition to the tires, the damping elements, the flexible elements and the rubber components, the other parts are considered as rigid during the simulation analysis process. And the sprung mass is considered as a rigid which has 6 degree of freedom.

2. Simplified the flexible link among the rigid appropriately, using the linear elastic rubber sleeves to simulate the dynamic features in actual operating conditions, and neglected each movement rates.
3. The horizontal stabilizer equivalents to the torsion bar spring in the actual operating conditions, but in this paper, we simplified it as a rigid lever and torsion agency.
4. As the engine module and brake system module are only used to control the vehicle's speed, this paper uses the engine and brake system module within the ADAMS/CAR database, and simplified the power transmission system accordingly, just considering the power transmission after the transmission axis, which means adding the toque in the isokinetic universal joints. And the engine has been simplified to a 6 degree of freedom rigid which has a corresponding quality characteristic parameter, and link it to the body and the under frame of the vehicle through 4 nonlinear rubber bushes.

In the modeling process by ADAMS/CAR, the whole vehicle was separated into 7 subsystems. First, we build the models of each system, then establish a communication port order among theses subsystems, and then assembly it into a vehicle model. As shown in Fig. (**1**).

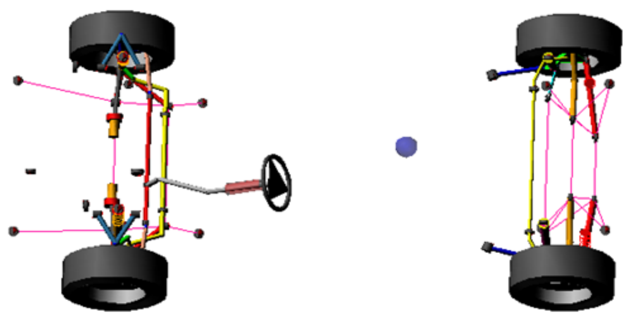


Fig. (1). Figure vehicle model.

As the vehicle is still at the preliminary design stage, there are not any test data to be certificated and contrast. In order to test the feasibility and accuracy of the modeling method, this paper uses the same modeling method to simulating analysis another kind of car in the same platform. The Figs. (**2-5**) are the contrast curves of the shift back to card experiment data, the steady rotary test, the step input test data and the simulating analysis result. From the contrast curves of the experimental data and the simulation result, we can find that the simulating curve and the experimental data curve fit the better. Because there is a great similarity between these two cars, we can note that the modeling method of this paper is feasible and has a high accuracy. It certainly can be used to do the vehicle's performance forecast analysis.

3. CONTROLLABILITY AND STABILITY SIMULATION ANALYSIS

The set of the simulation conditions and simulation methods of this paper have referenced 6 state standards of China when testing the vehicle Controllability and Stability. Just as shown in Table **1**.

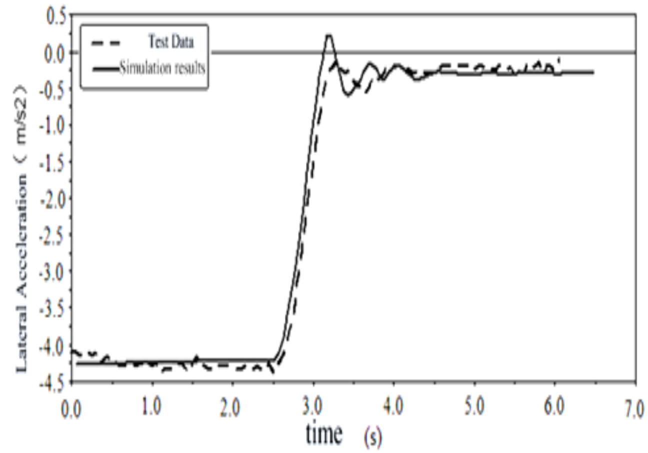


Fig. (2). Shift back to the lateral acceleration time curve.

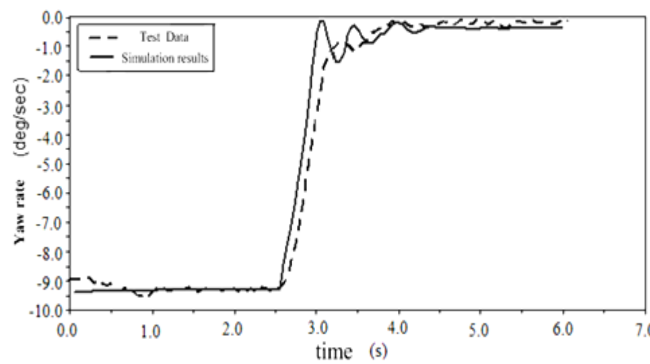


Fig. (3). Shift back to the time yaw rate curve.

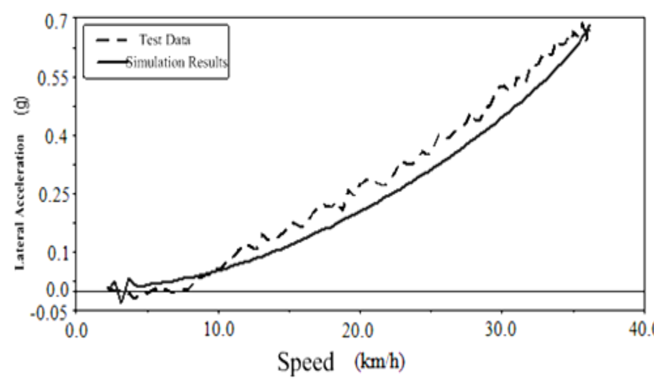


Fig. (4). Rotary steady lateral acceleration and speed of the curve.

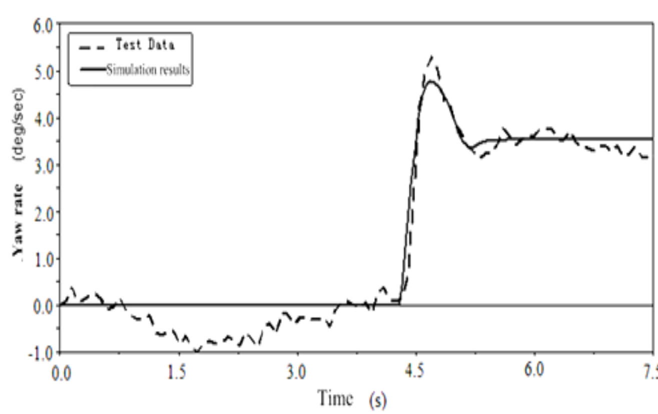


Fig. (5). Step input at the time yaw rate curve.

Table 1. Controllability and Stability Simulation Analysis of six Test and Reference Standards.

Simulation Test	China's Standard Reference
Pylon Course Slalom Test	GB/T6323.1 1994
Steering Wheel Angle Step Input	GB/T6323.2 1994
Steering Wheel Angle Pulse Input	GB/T6323.3 1994
Returnability Test	GB/T6323.4 1994
Steering efforts test	GB/T6323.5 1994
Steering Static circular test	GB/T6323.6 1994

This paper firstly obtained the evaluation value of each performance experiment, then used the state automobile industry standards QC/T-4801999 (the automobile Controllability and Stability indicators limits and evaluation methods) to evaluation and scoreboard the simulation results of each performance test.

From the Table **2**, we can infer that the vehicle's Steering efforts test, shift to the back properties and Pylon Course Slalom Test are excellent, the steady performance rotary is good, but the evaluation score of its neutral turning point's lateral acceleration is only 65.8 points. This result notes that the lateral acceleration of the vehicle's neutral turning point is low, and the evaluation scoreboard points of the Steering wheel angle step input and Steering Wheel Angle Pulse Input characteristics are somewhat low, too. The evaluation scoreboard point of the formant rate is especially low, which is only 48.0 points. This means that the vehicles' transient steering characteristics is normal, which need to be improved further.

4. OPTIMIZATION ANALYSIS OF CONTROLLABILITY AND STABILITY

As this vehicle is still at the preliminary design stage, some targeted parameters of the whole vehicle is decided by the experience, and there is a change scope which can be amended in some extension. The optimization objective of this paper is to chose a parameter which can make the objective function best. This paper chooses the vehicle quality, front axle load and rear horizontal stabilizer torsional stiffness as the optimization parameters, uses the maximum comprehensive evaluation score of the steady rotary test, step input test, Steering Wheel Angle Pulse Input test as the objective function, uses the maximum and minimum score of the corresponding parameters as the binding function, using the method of improving gradually to optimization analysis the vehicle multi-body system.

This paper is the optimization method on three separate optimization of design parameters change, their right to observe the impact of the objective function, each one optimized parameters determine the optimal value, make

Table 2. The Simulation Test and Evaluation.

Simulation Test		Evaluation Index	Total Individual Evaluation Scores	Total Comprehensive Assessment Scores
Pylon Course Slalom Test		The average peak yaw rate γ	92.8	95.2
		Steering Wheel Angle average peak θ	100	
Steering wheel angle step input		Lateral acceleration value of 2 m/s^2 vehicle yaw rate response time T	68.6	68.6
Steering Wheel Angle Pulse Input		Formant frequencies f_γ	48.0	77.1
		Formant level D	88.5	
		$f = 0.6Hz$ when the phase lag angle $\alpha\left(^o\right)$	94.8	
Returnability Test	Back to the low-speed	Residue absolute yaw rate $\Delta\omega_\gamma$	99.5	94.1
		Yaw rate total variance E_{ω_r}	100	
	Back to the high-speed	Residue absolute yaw rate $\Delta\omega_r$	99.6	
		Yaw rate total variance E_{ω_r}	77.3	
Steering efforts test		Steering wheel of the average post F_s	100	100
		Steering wheel of the largest post F_m	100	
Steering Static circular test		Neutral point to the lateral acceleration values a_n	65.8	84.6
		Inadequate degree shift U	88.1	
		Roll degrees compartment K_φ	100	

objective function parameters in the process of change to achieve the best. Optimization after a previous one as a basis for determining the optimal parameters of another value, for a total of three sub-optimization, Vehicle ultimately determine the quality of front axle shaft and a horizontal stabilizer after the torsional stiffness of the optimal value, determine the best optimization programs, as shown in Fig. (**6**).

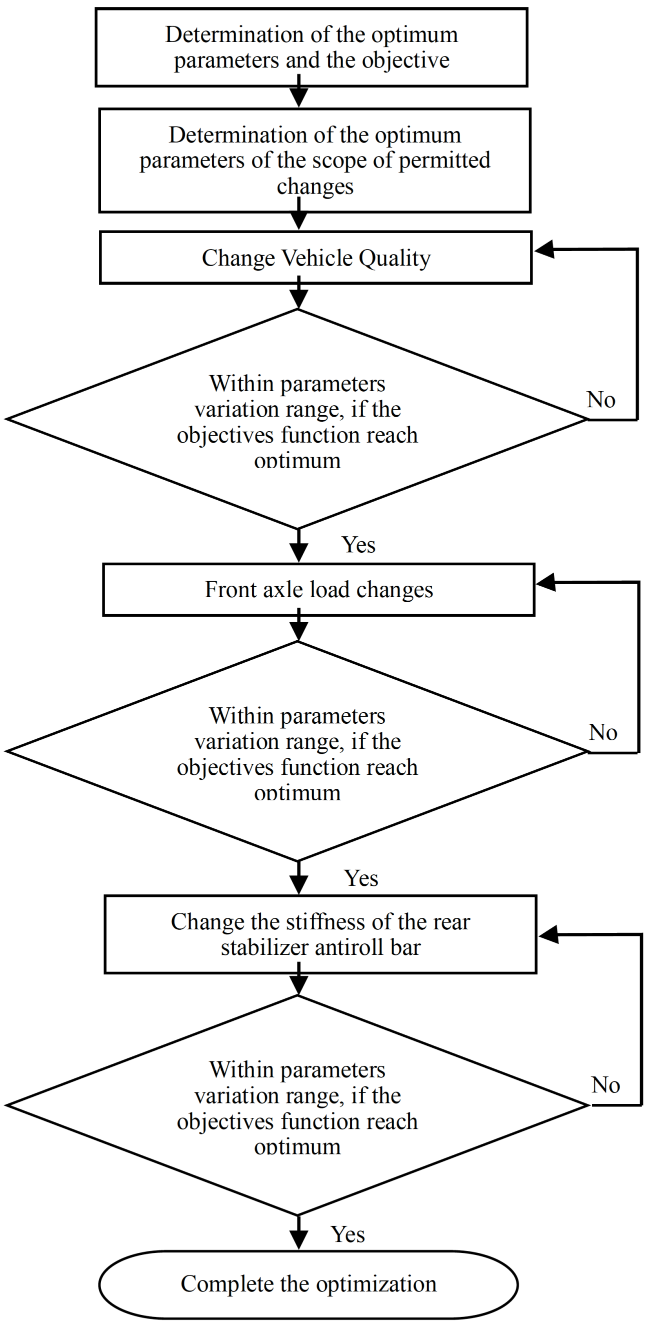
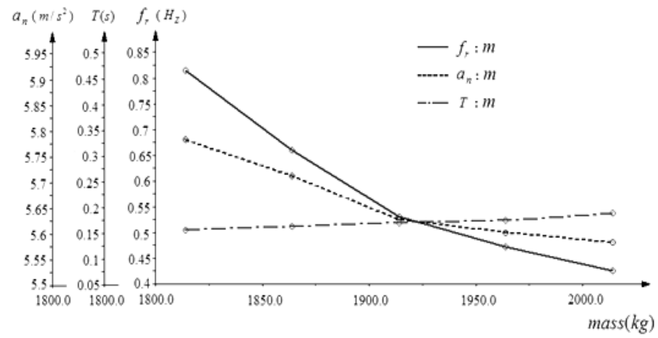


Fig. (6). Optimization step flowchart.

4.1. Vehicle Quality Optimization

Simulation analysis showed that along with the quality and the moment of inertia decreases Steady middle of the rotary at the neutral point to the value of lateral acceleration and angular pulse input frequency of the resonant peak is gradually increasing, and the step input of yaw rate response time has gradually decreased, The driver, a characteristic shift gradually improved (shown in Fig. **7**). When Vehicle Quality reduced to 1814 kg, to the best properties. Figs. (**8-10**) is the key to the quality of m = 1814 kg of calculation results and the original program (m = 1914 kg) The results contrast curve.



In map: a_n——Steady Rotary neutral to the point of lateral acceleration, T——Lateral acceleration values of $2m/s^2$, Vehicle yaw rate response time, f_r——Steering Wheel Angle Pulse Input frequency of the resonant peak

Fig. (7). With the evaluation of vehicle quality curve.

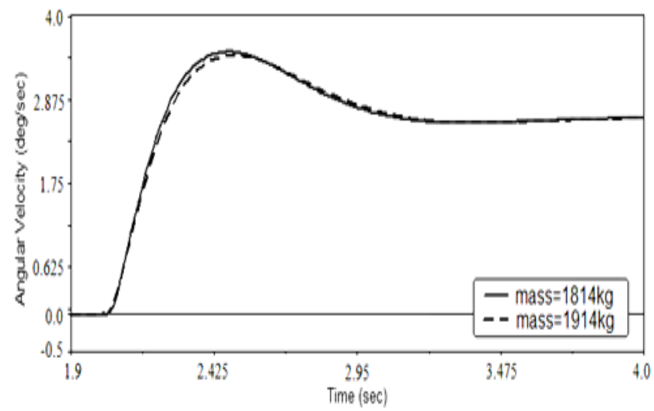


Fig. (8). Step input of yaw rate response curve.

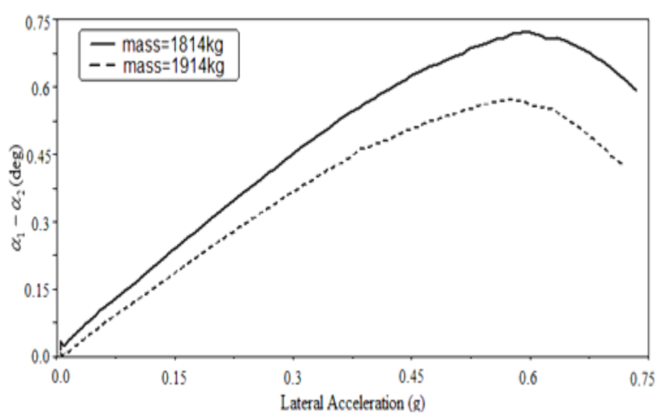


Fig. (9). Steady rotary rear side of the poor angle and lateral acceleration curve.

4.2. Front Axle - Load Optimization

Vehicle axle - load distribution on tire life and vehicle affect the performance [3]. From the tire wear evenly and life expectancy similar to consider various types of load should be little difference; To ensure a good vehicle dynamics and through, the bridge should have driven large enough load, and the output shaft may be appropriately reduced. To ensure a good vehicle Controllability and Stability, steering axle load should not be too small. Precursor to the front car, fully loaded, the front axle shaft in a 47% -60%, empty at the front axle shaft in a 56% -66% between [9, 10].

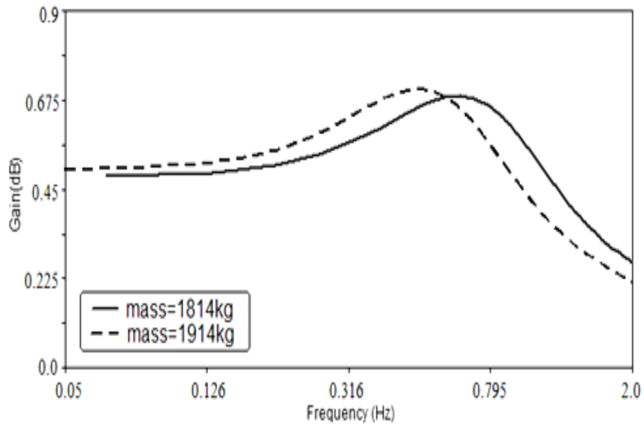


Fig. (10). Yaw rate increase frequency characteristic curve.

The driver, a front axle shaft design of the initial target value for 50.1%, the permissible scope of the changes (49.1% 51.1%) with the front axle to axle - load increases, along with the center will be a corresponding 1216 it helps strengthen inadequate to properties. From the 1st optimization can be seen, along with 100 kg decreased the quality, steering characteristics of the evaluation scores or relative original program has also been enhanced. Therefore, we can in a time optimization is the foundation for change, the gradual increase of front axle load, for the 2nd optimization.

Simulation analysis showed that with the front axle to axle - load increases, Steady middle of the rotary at the neutral point to the value of lateral acceleration and angular pulse input frequency of the resonant peak is gradually increasing, step input of yaw rate gradually decreased response time, the current axis of the axle - load increased 51.1%, Vehicle inadequate to the best properties (shown in Fig. **11**). Chart 12 - 14 shown in figure front axle shaft for a 51.1% and 50.1% when compared to the simulation curve.

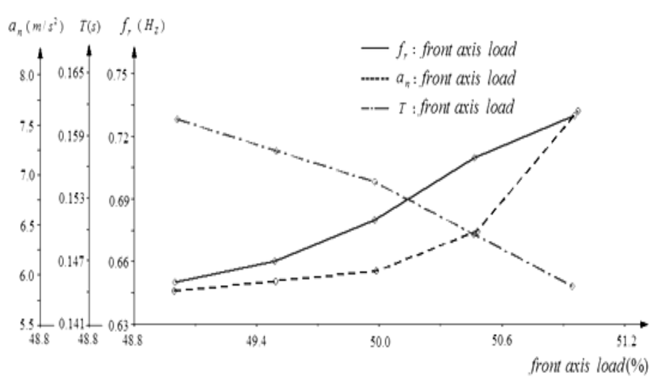


Fig. (11). Evaluation with the front axle shaft of a curve.

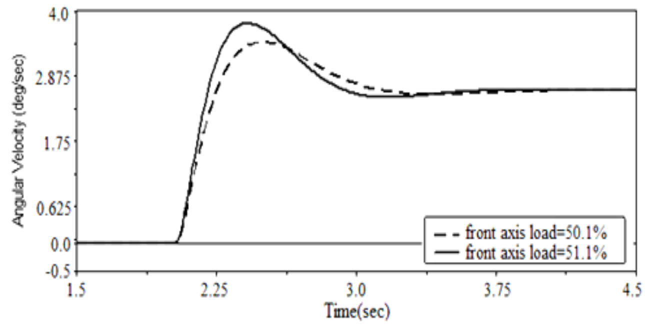


Fig. (12). Step input of yaw rate response curve.

4.3. Rear Horizontal Stabilizer Torsional Stiffness Optimization

Before and after the suspension of the roll stiffness of the vehicle matching the characteristics to a greater impact [11]. The current suspension roll stiffness remained unchanged and only dropped after the suspension roll stiffness, less than conducive to the formation of a steering characteristics, Instead, the car is easy to shift the excessive performance. Adjust before and after the suspension roll stiffness than the one commonly used method is to change before and after the horizontal stabilizer of torsional stiffness value. We can increase before the horizontal stabilizer torsional stiffness (front suspension increased roll stiffness) or decrease after the horizontal stabilizer Torsional rigidity (lower after suspension roll stiffness) to increase automobile steering characteristics of the deficiencies. From the previous steady state cornering performance simulation analysis showed that the car compartments tilt angle and smaller than similar cars, Roll it is not easy to lead, or its evaluation scores of 100 on the front and rear suspension roll stiffness large enough. Thus horizontal stabilizer before increasing the torsional stiffness (front suspension increased roll stiffness) will make cars tilt angle is smaller, Although stability control has improved, but the ride will be lowered accordingly. And appropriate to reduce the horizontal stabilizer after the torsional stiffness not only be better Controllability and Stability. Comfort will have a corresponding enhancement.

The driver, a horizontal stabilizer after the torsional stiffness of the initial design target value 11,500, Allow scope of the changes (9500,13500), Based on the 2nd optimized on the basis of decreased gradually after the horizontal stabilizer of torsional stiffness value, for the 3rd optimization.

Simulation analysis showed that with the horizontal stabilizer after the torsional stiffness decreases, then the lack of increased gradually shift characteristics, When the horizontal stabilizer after the torsional stiffness reduced to 9,500, along with the lack of properties to achieve the best, But step input of yaw rate response time, smaller rate increases, Kok and pulse input frequency of the resonant peak is slightly reduced. After the horizontal stabilizer of the torsional stiffness of 11,500 and 9,500 hours of simulation results contrast curve in Figs. (**13-16**) are shown.

Table 3. The programs the best parameter value.

Optimal Parameters	The Original Proposal	Optimization 1	Optimization 2	Optimization 3
Vehicle loaded with quality (kg)	1914	1814	1814	1814
Loaded with a front axle shaft (%)	50.4	50.4	51.4	51.4
After the suspension horizontal stabilizer torsional stiffness ($10^3\ N \cdot mm / \deg$)	11.5	11.5	11.5	9.5

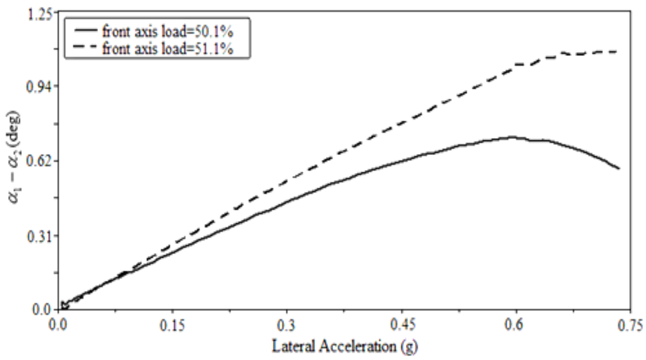


Fig. (13). Steady rotary rear side of the poor angle and lateral acceleration curve.

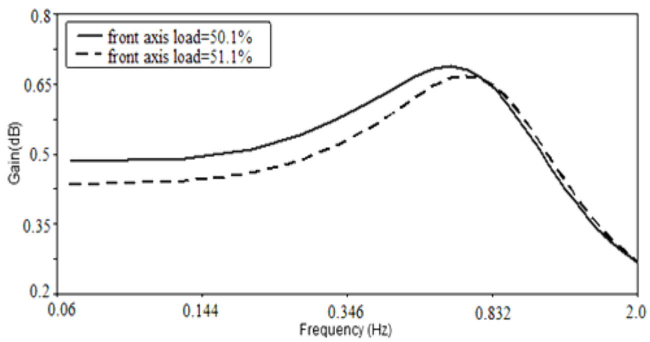


Fig. (14). Yaw rate increase frequency characteristic curve.

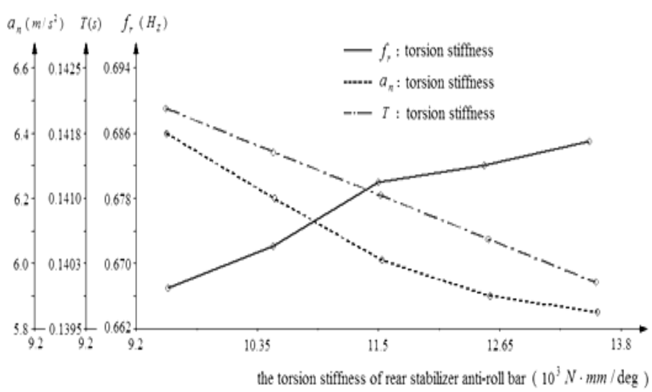


Fig. (15). Evaluation subsequent horizontal stabilizer torsional stiffness of the curve.

K is the horizontal stabilizer after the torsional stiffness value

Table **3** optimization for the above three parameters in three optimization program the best value, Table **4** for the three programs optimization objective function relative to the original volume of the optimization program.

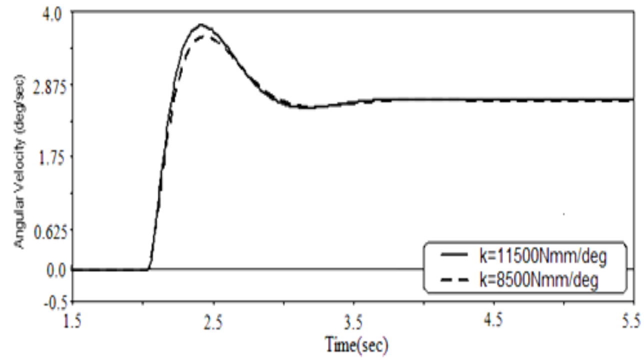


Fig. (16). Step input of yaw rate response curve.

CONFLICT OF INTEREST

The authors confirm that this article content has no conflict of interest.

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