

Roll Stiffness Optimization for Anti-roll Bar in Interconnected Air Suspension

Zhong-Xing Li¹, Rong-Zhou Xu^{1*} and Hong Jiang²

¹*School of Automobile and Traffic Engineering, Jiangsu University,
Zhenjiang 212013, P.R. China*

²*School of Mechanical Engineering, Jiangsu University,
Zhenjiang 212013, P.R. China*

Abstract

Lateral Interconnected Air Suspension (hereinafter referred to as Interconnected Air Suspension or IAS) tends to deteriorate vehicle roll stability on the condition of steering while travelling in high speed, so interconnection state is generally closed when lateral acceleration of car body exceeds its designed threshold (0.4 g in this paper). In this paper, a roll stiffness optimization strategy of anti-roll bar in IAS based on genetic algorithm is proposed for better roll stiffness as well as better roll angle vibration characteristics both in the state of interconnection and non-interconnection. And the strategy is used to optimize the anti-roll bar of a passenger car equipped with IAS. In the optimization strategy, weighted sum of body roll angle's mean value and standard deviation is originally determined as the objective function based on impact sensitivity analysis, i.e. the analysis of anti-roll bar roll stiffness's influence on body roll angle's mean value and standard deviation. Besides, totally 6 driving conditions are considered in the optimization to make it more realistic. The optimization result shows that optimal roll stiffness of front and rear anti-roll bar is 1998 N · m/deg and 1402 N · m/deg respectively. The proposed optimization strategy helps to resolve the problem of how to balance vehicle roll stability and roll angle vibration characteristics under different working conditions during the process of anti-roll bar roll stiffness matching for IAS vehicles. It can also be guidance or a reference for the matching of other parameters in IAS vehicles.

Key Words: Roll Stability, Roll Angle Vibration Characteristics, Full Car Model, Optimization Conditions Analysis, Genetic Algorithm

1. Introduction

Interconnected air suspension (IAS) is a new type of suspension with good vibration isolation and torsion eliminating performance that has considerable development potential [1]. However, it is inclined to cause big body roll angle on the condition of high speed steering, which to some extent hinders its popularization and application [2]. Anti-roll bar is an important component to enhance vehicle roll stability in suspension system, which is even more crucial for interconnected air suspension where the anti-roll bar's effect of enhancing vehicle

roll stability is more prominent according to the above analysis. Therefore, reasonable matching of anti-roll bar angle stiffness for IAS is particularly important. Currently, there are quite a few research findings on the matching of anti-roll bar roll stiffness at home and abroad. In 2010, Taguchi Methods was used by Dong Junhong of Hunan University to carry out the robust matching optimization design of suspension parameters including anti-roll bar roll stiffness, and the best combination of suspension parameters in local area was obtained through orthogonal test [3]. However, since only a few factor levels are considered and the interaction between optimization variables is neglected, the optimal solution can sometimes be missed in Taguchi Methods. In 2012, the

*Corresponding author. E-mail: joe937809231@yahoo.com

matching of anti-roll bar roll stiffness was performed by Hu Jiuqiang of Southwest Jiaotong University according to the traditional matching process, i.e. the permissible roll angle-roll stiffness of the whole body-roll stiffness of front and rear suspension. Then the principle and correction factor of anti-roll bar roll stiffness matching was discussed [4], but this matching method is only suitable for the traditional suspensions with constant stiffness. In 2014, Pravin Bharane of India Pune Dnyanganga Engineering Research Institute analyzed the influence of anti-roll bar structural parameters on its stiffness and stress distribution based on finite element analysis software Ansys, and then optimized the structure of the anti-roll bar. However, the optimization is merely based on the anti-roll bar's characteristics, regardless of its impact on vehicle dynamic performance [5]. Additionally, Wu Wenguang from Hunan university of China put forward a genetic algorithm based optimization strategy for anti-roll bar roll stiffness in which the minimum of vehicle roll angle's amplitude under sinusoidal steering angle input was chosen as objective function [6]. But it can be seen from the objective function that only roll stability was taken into account in the optimization process, in other words, the influence of anti-roll bar roll stiffness on roll angle vibration characteristics was neglected, resulting in the deterioration of vehicle roll angle vibration characteristics. The new cited paper [7] also shows an optimization strategy for anti-roll bar roll stiffness based on genetic algorithm [7]. But in this optimization strategy, anti-roll bar roll stiffness, spring stiffness and the damping coefficients of shock absorbers were optimized all together, which means the optimization result should

be a compromise of the aforementioned three suspension parameters due to the irreconcilability between them. Therefore, none of these suspension components could be made full use of. Wang Changxin from Jilin university of China utilized multi-objective immune algorithm to optimize the roll stiffness of anti-roll bars which was mounted on a traditional suspension [8]. However, since only one or two driving conditions were taken into consideration during both the optimization processes stated by Wu Wenguang and by Wang Changxin, their optimization results can not fully satisfy vehicle's demand for good rolling characteristics under complex driving conditions.

The traditional matching methods are no longer applicable since the stiffness of air suspension is changing throughout its compression and stretch travel and in the interconnected air springs, gas flow caused by road excitation couples with the gas flow caused by the body roll [9]. Therefore, an optimization matching strategy of anti-roll bar roll stiffness based on genetic algorithm is proposed. After that, anti-roll bar roll stiffness of a certain IAS vehicle is optimized, taking into account vehicle roll stability and lateral angular vibration characteristics under different conditions to make sure the optimal results meet the demand for good rolling performance both in interconnection and non-interconnection state.

2. Mathematical Model of IAS

2.1 Full Car Model with Seven DOFs

As is shown in Figure 1, kinematic and dynamic theoretical analysis of interconnected air suspension vehi-

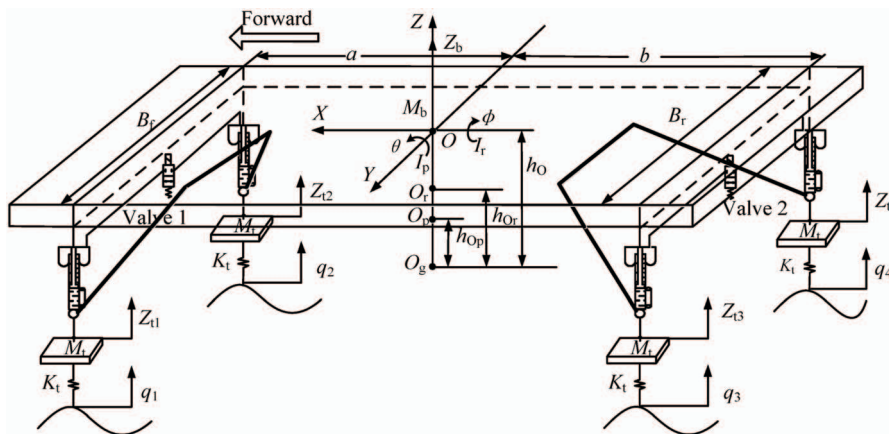


Figure 1. Full car physical model with IAS.

cle physical model was conducted, followed by the establishing of a full car mathematical model with seven DOFs including body vertical, roll, pitch and vertical movement of the four wheels, as shown by equation (1):

$$\begin{cases} M_b \ddot{Z}_b = F_1 + F_2 + F_3 + F_4 - M_b g \\ I_r \ddot{\Phi} = (F_1 - F_2) \cdot \frac{B_f}{2} + (F_3 - F_4) \cdot \frac{B_r}{2} + M_b g \Phi (h_o - h_{o_r}) \\ \quad + M_b a_y (h_o - h_{o_r}) - 4 M_t a_y (h_{o_r} - r) \\ I_p \ddot{\Theta} = (F_3 + F_4) b - (F_1 + F_2) a \\ M_{t1} \ddot{Z}_{i1} = K_t (q_1 - Z_{i1}) - F_1 + M_b g b / (2(a+b)) \\ M_{t2} \ddot{Z}_{i2} = K_t (q_2 - Z_{i2}) - F_2 + M_b g b / (2(a+b)) \\ M_{t3} \ddot{Z}_{i3} = K_t (q_3 - Z_{i3}) - F_3 + M_b g a / (2(a+b)) \\ M_{t4} \ddot{Z}_{i4} = K_t (q_4 - Z_{i4}) - F_4 + M_b g a / (2(a+b)) \end{cases} \quad (1)$$

In which,

$$\begin{cases} F_1 = (P_1 - Pa) \cdot A_{e1} + c \cdot \dot{f}_{d1} + K_{\Phi bf} \cdot (f_{d1} - f_{d2}) / B_f^2 \\ F_2 = (P_2 - Pa) \cdot A_{e2} + c \cdot \dot{f}_{d2} + K_{\Phi bf} \cdot (f_{d1} - f_{d2}) / B_f^2 \\ F_3 = (P_3 - Pa) \cdot A_{e3} + c \cdot \dot{f}_{d3} + K_{\Phi br} \cdot (f_{d3} - f_{d4}) / B_r^2 \\ F_4 = (P_4 - Pa) \cdot A_{e4} + c \cdot \dot{f}_{d4} + K_{\Phi br} \cdot (f_{d3} - f_{d4}) / B_r^2 \\ f_{d1} = Z_{i1} - (Z_b - a\theta + \Phi \cdot B_f / 2) \\ f_{d2} = Z_{i2} - (Z_b - a\theta - \Phi \cdot B_f / 2) \\ f_{d3} = Z_{i3} - (Z_b + b\theta + \Phi \cdot B_r / 2) \\ f_{d4} = Z_{i4} - (Z_b + b\theta - \Phi \cdot B_r / 2) \end{cases} \quad (2)$$

where, M_b is body mass, kg; M_t is tire mass, kg; a_y is lateral acceleration of body centroid, $m \cdot s^{-2}$; I_r and I_p denote body rotational inertia around X axis and Y axis respectively, $kg \cdot m^2$; Φ and θ denote body roll angle (positive when the car body tilts to the right) and body pitch angle (positive when the car body tilts to the front) respectively, rad; q_i ($i = 1, 2, 3, 4$) are road vertical excitations on the wheels, m; Z_{ii} ($i = 1, 2, 3, 4$) are vertical displacements of the wheels, m; Z_b is vertical displacement of body centroid, m; K_t is vertical stiffness of the tires, N/m; F_i ($i = 1, 2, 3, 4$) is suspension force, N; f_{di} ($i = 1, 2, 3, 4$) is suspension travel, m; A_{ei} ($i = 1, 2, 3, 4$) are effective areas of the air springs of front left, front right, rear left, rear right respectively, m^2 ; $K_{\Phi bf}$ and $K_{\Phi br}$ is respectively roll stiffness of front and rear anti-roll bars, $N \cdot m/rad$; c is damping coefficient (equaling c_c and c_s in compression and stretch travel respectively),

$N \cdot s/m$; B_f and B_r are respectively front and rear wheel tracks, m; a, b are respectively the distances from body centroid to the front and rear axles, m; O_g is the projection of body centroid O to the ground; h_o is the height of centroid, m; h_{o_r} is the height of roll center, m; h_{op} is height of pitching center, m; T is pitching moment while vehicle turning, $N \cdot m$; g is gravitational acceleration, m/s^2 ; Pa represents standard atmospheric pressure.

Supposing that the air spring is an adiabatic system, the internal gas motion equation can be described as [10]

$$p_i \left(\frac{V_i}{m_i} \right)^\kappa = const \quad (i = 1, 2, 3, 4) \quad (3)$$

where, V_i ($i = 1, 2, 3, 4$) are the volumes of each air spring; m_i are the masses of gas inside each air spring; κ is the isentropic exponent, which equals 1.4 for air.

Based on the principle of one dimension isentropic flow, mass flow rate through the holes can be described as follows [11]:

$$q_m = \begin{cases} AP_{up} \sqrt{\frac{1}{RT_{up}} \frac{2\kappa}{\kappa-1} \left[\left(\frac{P_{dn}}{P_{up}} \right)^{\frac{2}{\kappa}} - \left(\frac{P_{dn}}{P_{up}} \right)^{\frac{\kappa+1}{\kappa}} \right]} & \frac{P_{dn}}{P_{up}} \geq 0.528 \\ AP_{up} \left(\frac{2}{\kappa+1} \right)^{\frac{1}{\kappa-1}} \sqrt{\frac{1}{RT_{up}} \frac{2\kappa}{\kappa+1}} & \frac{P_{dn}}{P_{up}} < 0.528 \end{cases} \quad (4)$$

where, P_{up} is the gas pressure of upstream; P_{dn} is the gas pressure of downstream; T_{up} is the gas temperature of upstream; A is the effective cross section area of the orifice.

2.2 Simulating Model of Full Car with IAS

The dynamic simulating model of IAS vehicle is established with Matlab/Simulink, as shown in Figure 2.

Parameters of the model car are listed in Table 1.

3. Roll Stiffness Optimization of Anti-roll Bar

3.1 Objective Function

The purpose of roll stiffness optimization is to improve vehicle roll characteristics and alleviate the contradiction between its handling stability and ride com-

fort. Two aspects are included in vehicle roll characteristics, i.e. roll stability and roll angle vibration characteristics, where the former embodies vehicle handling stability and the latter embodies riding comfort.

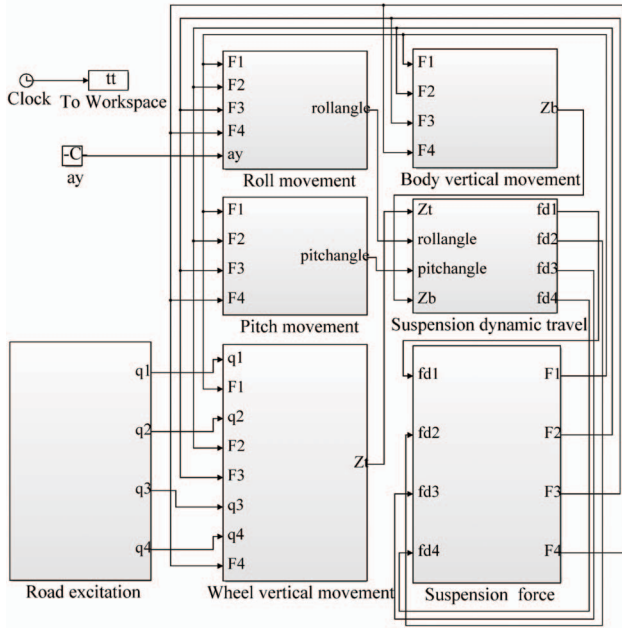


Figure 2. Simulating model of full car with IAS.

Table 1. Parameters of full car model

Parameter	Value
Body mass M_b (kg)	1839
Wheel mass M_t (kg)	40
Body rotational inertia around X axis I_r ($\text{kg} \cdot \text{m}^2$)	606
Body rotational inertia around Y axis I_p ($\text{kg} \cdot \text{m}^2$)	4192
Front wheel track B_f (m)	1.515
Rear wheel track B_r (m)	1.515
Distance from centroid to front axle a (m)	1.3
Distance from centroid to rear axle b (m)	1.4
Tire stiffness K_t (kN/m)	250
Original Roll stiffness of front anti-roll bar $K_{\Phi_{bf}}$ ($\text{N} \cdot \text{m}/\text{deg}$)	600
Original Roll stiffness of rear anti-roll bar $K_{\Phi_{br}}$ ($\text{N} \cdot \text{m}/\text{deg}$)	400
Damping coefficient in compression travel c_c ($\text{N} \cdot \text{s}/\text{m}$)	1800
Damping coefficient in stretch travel c_s ($\text{N} \cdot \text{s}/\text{m}$)	2800
Original volume of air spring V_0 (m^3)	0.001026
Effective area of air spring A_e (m^2)	0.009

As an important parameter to represent the roll characteristics of vehicle, body roll angle can reflect requirements for both handling stability and riding comfort. Specifically, mean value of roll angle reflects roll stability while the standard deviation reflects its vibration characteristics. As for IAS vehicle, anti-roll bar has great influence on both roll stability and roll angle vibration characteristics, which is shown in Figures 3 and 4. It should be pointed out that the mean value and the standard deviation of roll angle are calculated within the time period when roll angle fluctuation is caused purely by road roughness.

It can be seen from Figure 3 that a significant inverse correlation lies between roll angle mean value of IAS vehicle and the sum of front and rear anti-roll bar's roll stiffness. In particular, when roll stiffness of front and rear anti-roll bars are both in the range of $100 \sim 1000 \text{ N} \cdot \text{m}/\text{deg}$, roll angle mean value declines obviously as the sum increases. Furthermore, comparing point (100, 2000, 0.5868) with point (2000, 100, 1.558) in Figure 3, we can easily see that the ratio of front anti-roll bar roll stiffness to that of the rear one will also affect the roll angle mean value, and the larger the ratio, the greater the roll angle mean.

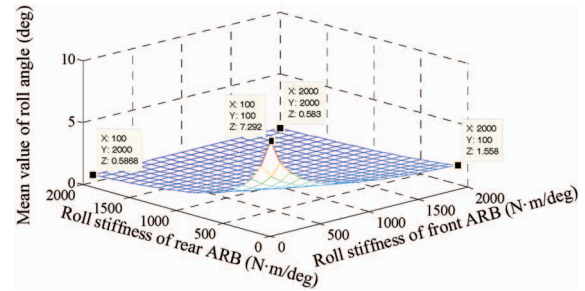


Figure 3. Roll angle mean value vs anti-roll bar roll stiffness.

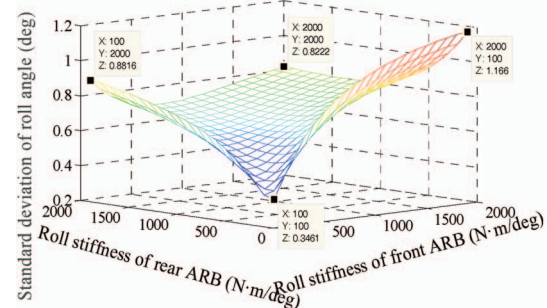


Figure 4. Roll angle standard deviation value vs anti-roll bar roll stiffness.

As is shown in Figure 4, the curve of roll angle standard deviation value vs anti-roll bar roll stiffness rises slowly as a whole with upwarps on both sides and forms a groove in the middle. So it is easy to discover that, the roll angle standard deviation value will increase as the sum of front and rear anti-roll bar's roll stiffness increases. What's more, the smaller the difference between roll stiffness of front and rear anti-roll bars, the smaller the roll angle standard deviation value will be.

According to the aforementioned analysis, there exists a contradiction between the improvement of roll angle standard deviation value and that of roll angle mean value in IAS vehicle, which means mere pursuit of either index will definitely lead to deterioration of the other. Hence, multi-objective optimization design of suspension is required so as to find a balance between handling stability and riding comfort. The target of optimization can be addressed as minimizing the weighted sum of mean value and standard deviation of body roll angle which can be calculated by means of statistics [12]. With the combination of vehicle time-domain model and genetic algorithm, the objective function of roll stiffness optimization for anti-roll bar is established as follows:

$$\min h(x) = \min(w_a f_\Phi(x) + w_b g_\Phi(x)) \quad (5)$$

where, $h(x)$ is the objective function, $f_\Phi(x)$ and $g_\Phi(x)$ represent respectively the roll angle mean value and standard deviation value when roll stiffness of anti-roll bar equals x , and the units are both deg; w_a and w_b stand for the weight coefficient of roll angle mean value and standard deviation value respectively.

3.2 Optimization Variables

In IAS system, little anti-roll effect is provided by air suspensions [13], and thus vehicle roll stability mainly relies on anti-roll bar, which will also exert direct influence on vehicle roll angle vibration characteristics when the vehicle is traveling on an uneven road. Therefore, the roll stiffness of front and rear anti-roll bars are selected as optimization variables, so as to improve vehicle roll stability and further ease the contradiction between handling stability and riding comfort. The optimization variables are shown as follows:

$$x = [K_{\Phi bf} \ K_{\Phi br}] \quad (6)$$

where, $K_{\Phi bf}$ and $K_{\Phi br}$ represent respectively the roll stiffness of front and rear anti-roll bar, both with a unit of $N \cdot m/\text{deg}$.

3.3 Constraint Conditions

In order to guarantee the functionality, practicality and safety of the suspension system, during the optimization design, the optimization variables must satisfy certain constraint conditions. According to the specific circumstances of the vehicle, the constraint conditions are established as follows:

- (1) For a passenger car, the travel limit of suspension $[f_d]$ is generally in the range of 7 cm~9 cm and 8 cm is chosen in this paper. When the root-mean-square of suspension dynamic travel is less than or equal to one third of the travel limit, i.e. $f_d \leq [f_d]/3$, the probability of suspension's hitting the limit block can be limited below 0.3%.
- (2) The relative dynamic load between the wheel and road surface, i.e. F_{zd}/F_{zs} , in which F_{zd} and F_{zs} are respectively wheel dynamic load and static load, has a considerable effect on the vehicle driving safety. When the relative dynamic load of a certain wheel is greater than 1, the wheel is likely to leave the ground, and the ground adhesion will be gone, and thus driving, steering and braking ability of the vehicle will be lost. This dangerous situation is not allowed during the vehicle driving process. Therefore, the relative dynamic load should not go beyond a reasonable range. When the root mean square of the relative dynamic load is less than 1/3, i.e. $F_{zd}/F_{zs} \leq 1/3$, the probability of wheel's jumping off the ground is not more than 0.15%, which means the wheels would hardly jump off the road surface [14].
- (3) According to reference [15], the ratio of the front to the rear suspension roll stiffness has a great effect on vehicle steady steering characteristics. In order to meet the requirements of vehicle steady steering characteristics, for passenger cars, the ratio is supposed to be within the range of 1.4~2.6 [15]. In case of no consideration for the rubber bushing, the roll stiffness of the suspension is the sum of the stiffness that provided by the spring and the stiffness of the anti-roll bar [16]. Since little roll stiffness is provided by air springs of IAS, the roll stiffness of IAS is seen as

equal to the anti-roll bar roll stiffness [17]. Therefore, the ratio of the front to the rear anti-roll bar roll stiffness shall meet the following requirement:

$$1.4 \leq \frac{K_{\Phi bf}}{K_{\Phi br}} \leq 2.6 \quad (7)$$

To sum up, the optimization model for the roll stiffness of anti-roll bars is obtained as follows:

$$\begin{cases} \text{Objective function: } \min h(x) = \min(w_a f_{\Phi}(x) + w_b g_{\Phi}(x)) \\ \text{Optimization variables: } x = [K_{\Phi bf} \ K_{\Phi br}] \\ \text{Constraint conditions: } f_d \leq \frac{8}{3}; \frac{F_{zd}}{F_{zs}} \leq \frac{1}{3}; 1.4 \leq \frac{K_{\Phi bf}}{K_{\Phi br}} \leq 2.6 \end{cases} \quad (8)$$

4. Optimization Methods and Optimization Conditions Analysis

4.1 Optimization Methods

4.1.1 Genetic Algorithm

Genetic algorithm is a highly efficient global optimization algorithm based on the theory of natural selection and heredity, which combines biological evolution mechanisms of survival of the fittest and random information exchange between chromosomes within the group [18]. Probabilistic transfer rules are obeyed when genetic algorithm is used to deal with group optimization matters, and genetic algorithm can also be used to optimizing more than one variables at the same time. Therefore, genetic algorithm is more efficient and can make it more convenient to analyze the relationship between parameters and evaluation indexes when compared with the traditional single point search method. In addition, in genetic algorithm what is operated directly is chromosome string, i.e. code of the variables rather than those variables themselves, so genetic algorithm would not be affected by continuity and differentiability of the objective function.

In this paper, genetic algorithm is used to solve the optimization problem of anti-roll bar in an IAS vehicle, and the solving procedure is displayed by the following flowchart.

4.1.2 Optimization Parameters of Genetic Algorithm

On the premise of ensuring enough accuracy for optimization results, combined with the purpose of reducing the optimization time, the initial population size is set as 60; the maximum generation is set as 200; the crossover probability is 0.6, and the mutation probability is 0.01. In order to unify the order of magnitudes of the sub-objective functions and conform to the fact that vehicles travel more often on straight roads than on curving roads, the weight coefficients ω_a and ω_b are set as 0.7 and 0.3 respectively. According to the actual circumstance of the vehicle, the range of optimization variables of is determined as 0~2000 N · m/deg.

4.2 Optimization Conditions Analysis

As the roll stiffness optimization of anti-roll bars proposed in this paper is based on the whole car dynamic simulation results, which are directly influenced by the simulation conditions. Therefore, choosing appropriate si-

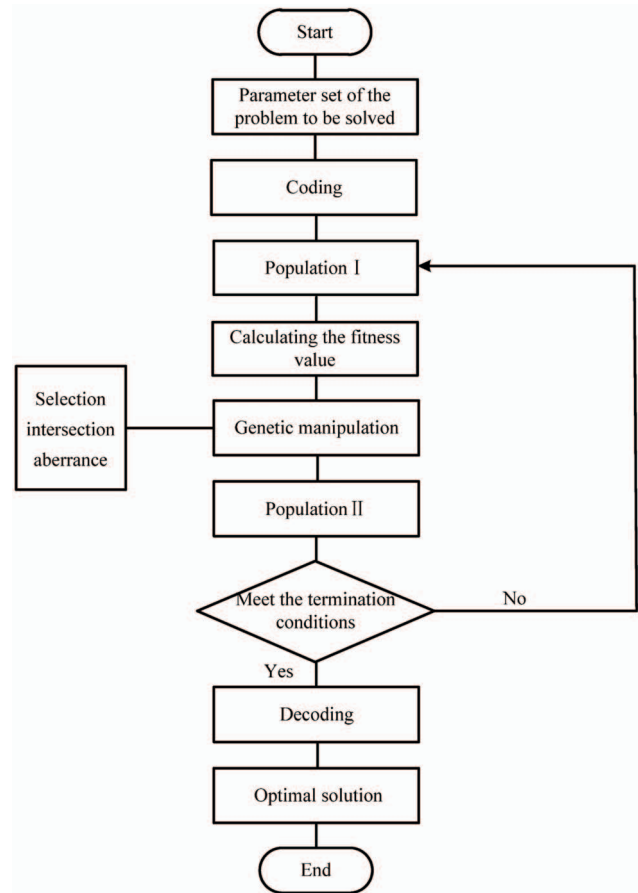


Figure 5. Operational flowchart of the genetic algorithm.

mulation conditions is crucial for enhancing the practical application value of the optimization results. Unlike most whole car models whose input variable are steering wheel angle, the vehicle model built in this paper use the lateral acceleration as the input of the system. The simulation results are mainly affected by three driving condition parameters: road grade, lateral acceleration and vehicle speed. Impact sensitivity analysis of the driving condition parameters is carried out, and the simulation conditions used in the optimization are determined based on the results of the analysis.

4.2.1 Road Grade

In the simulation process, lateral acceleration was set as 0.2 g, and the vehicle speed was 50 km/h. The random road grade was successively set as A, B, C, D, and other parameters were set as same as those in section 4.1.2. According to the international standard document entitled ISO/TC 108/SC2N67, geometric mean of road roughness coefficients of Grade A, B, C and D are respectively 16E-6 m³, 64E-6 m³, 256E-6 m³ and 1024E-6 m³. Roll characteristics simulation was conducted based on the Simulink model of IAS vehicle established in section 1.2. With the horizontal axis of geometric mean of road roughness coefficients and the vertical axis of objective function values, the curve of objective function vs road grade could be depicted as shown in Figure 6.

From Figure 6 it can be seen that the values of objective function show a growing trend with the increase of the road grade to some extent, and it has increased by

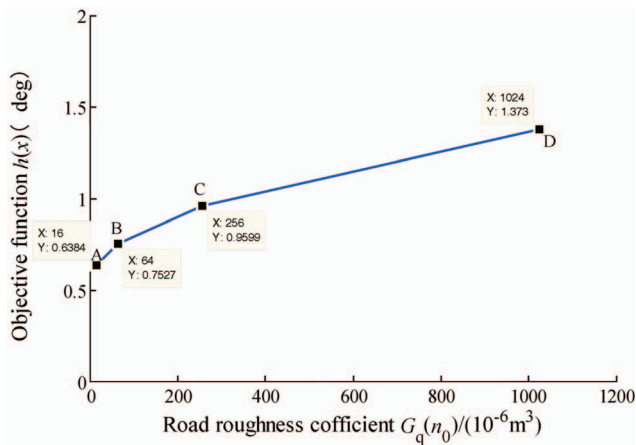


Figure 6. Objective function vs road roughness coefficient.

115.07% from grade A to grade D. But there is little difference between the objective function value of grade A and that of grade B, of which the gap is only 17.9%. Since the vehicle researched on in this paper seldom drives on the roads of grade D, and the road spectra of most highways in China lie in between grade B and grade C, the roads of grade B and grade C are selected as road conditions in the optimization process.

4.2.2 Lateral Acceleration

In the simulation process, road grade was set as grade B, and the vehicle speed was 50 km/h. The lateral acceleration ranged successively from 0 g to 1 g with the interval of 0.2 g, and other parameters were set as same as those in section 4.1.2. Vehicle roll characteristics simulation was then conducted, from which the curve of objective function vs lateral acceleration was obtained as shown in Figure 7.

As is shown in Figure 7, the value of objective function increases gradually with the rise of lateral acceleration of the body center of mass. And the objective function values at all sampling points do have large differences, except for those at 0.2 g and 0.4 g, of which the gap is only 7.46%. For instance, the objective function value at 0.2 g is 259.74% bigger than the value at 0 g. Owing to the fact that there are few chances that the vehicle travels with a lateral acceleration beyond 0.6 g, lateral acceleration of 0 g, 0.2 g and 0.6 g will be considered in the optimization.

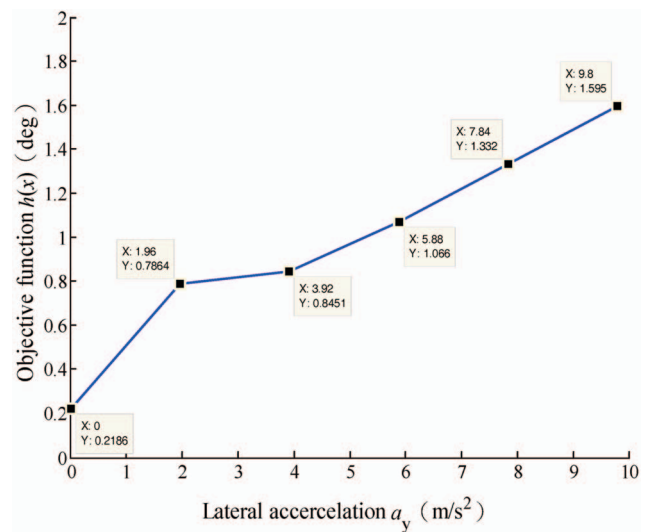


Figure 7. Objective function vs lateral acceleration.

4.2.3 Vehicle Speed

In the simulation process, road grade was set as grade B, and the lateral acceleration was 0.2 g. The vehicle speed ranged successively from 40 km/h to 110 km/h with the interval of 10 km/h, and The settings of other parameters were the same as those of the parameters in section 4.1.2. Then the curve indicating the impact of vehicle speed on objective function is given in Figure 8.

According to Figure 8, vehicle speed does have some influence on the objective function value, but the influence is very small. This can be proved by the fact that difference between objective function values at 40 km/h and 110 km/h is as small as 0.0556° . So the commonly used speeds of this car, i.e. 50 km/h and 70 km/h, were included in the speed conditions during the optimization process.

5. Optimization Results and its Verification

5.1 Optimization Results

According to the sensitivity analysis of the vehicle condition parameters in section 4.2, the optimization conditions of anti-roll bar roll stiffness are shown in Table 2.

All the conditions in Table 2 were taken into account in the optimization. And now the objective of the optimization can be redescribed as finding out a pair of roll stiffness of front and rear anti-roll bars that minimizes

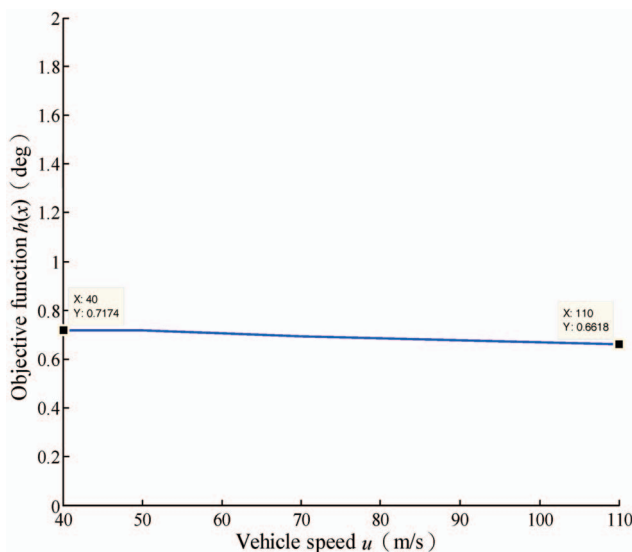


Figure 8. Objective function vs vehicle speed.

the weighted sum of roll angles' mean values and standard deviations under different conditions, namely, the general objective function can be depicted as follows:

$$\min H(x) = \min \sum_{i=1}^6 h_i(x) \quad (9)$$

where, $H(x)$ represents the general objective function; i represents the number of conditions; $h_i(x)$ represents the objective function value under condition No. i .

After running the genetic algorithm optimization program, the optimization results was finally obtained as follows:

$$K_{\Phi_{bf}} = 1998 \text{ N} \cdot \text{m/deg and } K_{\Phi_{br}} = 1402 \text{ N} \cdot \text{m/deg}$$

5.2 Verification of the Optimization Results

To verify the optimization results and validity of the proposed optimization strategy, the roll angle's time history before optimization was compared with that after optimization under two typical driving conditions when the interconnection switches are on and off respectively, as shown in Table 3.

And the results of comparisons can be shown in Figures 9 and 10.

In fact, roll stability and roll angle vibration characteristics are respectively part of vehicle handling stability and riding comfort, so they are contradictory to each other and can not be improved at the same time, and that means roll angle mean value and roll angle standard de-

Table 2. Optimization conditions

Number of conditions	Road grade	Lateral acceleration ($\text{m} \cdot \text{s}^{-2}$)	Speed ($\text{Km} \cdot \text{h}^{-1}$)
1	B	0	70
2	B	0.2 g	70
3	B	0.6 g	50
4	C	0	70
5	C	0.2 g	50
6	C	0.6 g	50

Table 3. Driving conditions

Number of conditions	Road grade	Lateral acceleration ($\text{m} \cdot \text{s}^{-2}$)	Speed ($\text{Km} \cdot \text{h}^{-1}$)
1	B	0.2 g	70
2	B	0.6 g	50

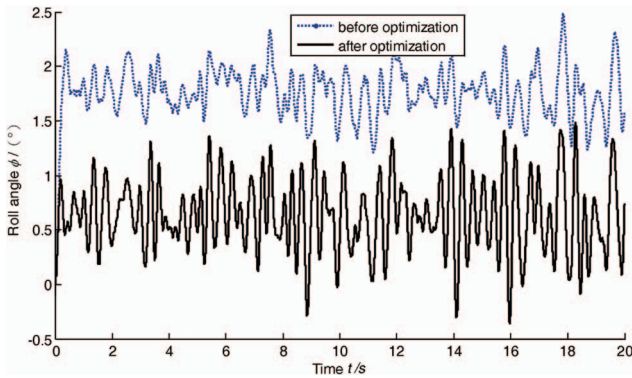


Figure 9. Roll angle time history under driving condition 1.

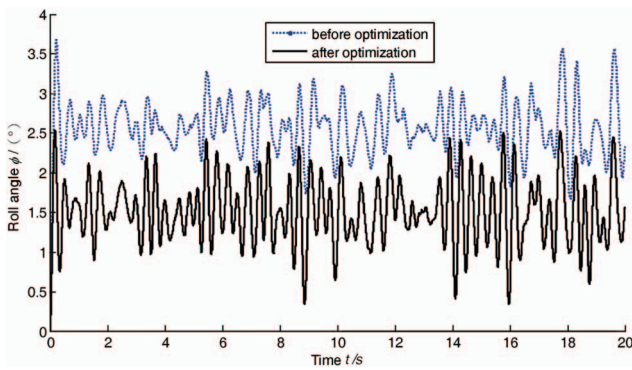


Figure 10. Roll angle time history under driving condition 2.

viation value may not be decreased simultaneously. In other words, a reduction of roll angle mean value will inevitably lead to an increase of roll angle standard deviation value. However, it can be clearly seen from the Figure 9 that, roll angle mean value of the IAS vehicle under driving condition 1 declines dramatically after the anti-roll bars are optimized, indicating that the proposed optimization strategy can improve vehicle roll stability significantly. Further calculation shows that with the optimization of anti-roll bar roll stiffness, roll angle mean value of IAS vehicle under driving condition 1 has reduced by 63.96% while the roll angle standard deviation merely goes up by 59.05%. Additionally, the gap between the reduction percentage of roll angle mean value and the increase percentage of roll angle standard deviation value even goes larger when the IAS vehicle mounted with optimal anti-roll bars is traveling under driving condition 2; the reduction percentage of roll angle mean value and the increase percentage of roll angle standard deviation value are respectively 41.45% and

20.08%. To conclude, the comparisons between optimal cases and non-optimal cases manifest that the optimization strategy proposed in this paper can improve the roll stability of IAS vehicles significantly with a relatively small loss of roll angle vibration characteristics. The proposed optimization strategy is effective.

6. Conclusions

- (1) Roll stability of IAS vehicle increases as the sum of front and rear anti-roll bar roll stiffness increases. Besides, the ratio of front anti-roll bar roll stiffness to rear anti-roll bar roll stiffness also has an impact on the vehicle roll stability and the larger the ratio, the stronger the roll stability will be.
- (2) Anti-roll bar will deteriorate the lateral angular vibration characteristics of IAS vehicle. The larger the sum of front and rear anti-roll bar roll stiffness, the more severe vehicle lateral angular vibration will be, which means the lateral angular vibration characteristics will get worse. However, narrowing the gap between front and rear anti-roll bar roll stiffness can help to improve the lateral angular vibration characteristics of the vehicle.
- (3) The anti-roll bar roll stiffness of an IAS vehicle was optimized based on genetic algorithm, from which the optimal roll stiffness of front and rear anti-roll bars was obtained, i.e. 1998 N · m/deg for the front anti-roll bar and 1402 N · m/deg for the rear one. Both vehicle roll stability and lateral angular vibration characteristics under different circumstances were taken into consideration during the process of optimization using the aforementioned method, and thus the optimized anti-roll bars can satisfy the needs of vehicle roll characteristics better.

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