Hard Points Optimization Design of Air Spring Suspension Based on Sensitivity Analysis

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Abstract — According to a company developed twelve meters bus, using the multi-body dynamics software ADAMS, they reached the conclusion that the change scope of the toe angle, and camber in the guide mechanism with wheel vertical, affects the operation stability. Utilizing: i) the air suspension guide mechanism of spatial position as design variables, ii) the front wheel alignment parameters as the objective function based on the insight module, iii) sensitivity analyzed in relevant hard point coordinates, and iv) experimentally designed and regression analyzed the variables, we optimized the hard points to improve the performance of the suspension system. The results show that the variation range of the front beam and camber angle decreases clearly with the optimized air suspension test. Through real vehicle verification, the simulation results are consistent with the trend of the real vehicle test, and the operation stability of the passenger car has a certain level of improvement.

Key words: Air spring suspension; Virtual Prototype Technology; Simulation analysis; Hard point optimization

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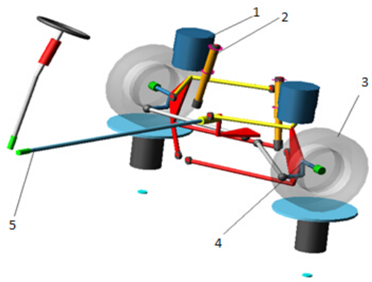
I .BACKGROUNDS

According to the relevant regulations, the large buses and senior middle and medium buses manufactured after 2010 were required to install air suspension, causing the need to improve the stability and comfort of the bus. Whether the installation of air suspension, has became one of the main criteria for the evaluation of commercial vehicle. Air suspension can transfer force better, attenuate the vibration, making the air suspension more "soft", which improved the comfort performance greatly. At the same time, the air suspension needs to guide the transmission structure, which increases the difficulty of match, if the matching is not properly, it may lead to tire wear seriously. A company in the trial of a 12 m length aluminum bus, found that there is a situation that the tire has been excessive wear, the prediction may be that the hard points of guide mechanism unreasonable, making tire wear seriously.

The development of virtual prototyping technology provides a quick way to analyze the new models, which greatly shorten the product development cycle[1]. In the development of automobile chassis, we need to optimize the suspension parameters and the relationship between the wheel jump, so as to improve the performance of the vehicle. Based on the three-dimensional model, the analysis of the suspension structure and the structure of the movement relationship. Aiming at the problems in the actual production, the hard point is optimized by using the multi-body dynamics module in the study. After optimization, the relevant parameters matching are more reasonable, and the variation of the parameters is also decreased, which reduces the wear of the tire. The simulation results can provide a theoretical basis for the design and analysis of the suspension.

II. SUSPENSION MODELING

In the process of establishing the model, followed the order of Point > Part > Geometry > Attachments. Measures related to hard point parameters in UG 3D bus model and create component based on a hard point. Finally, according to the actual motion relations between components, create connection (kinematic pair) between components. This paper according to the specific design of the type of bus, coupling including spring curve, parameters, established accurately car front suspension system, steering system, and the eventual establishment of the passenger car suspension simulation test model. As shown in fig.1:



1.Air spring assembly 2.Shock 3.Tire 4.Steering knuckle connecting bracket 5.Steering assembly

Fig.1 Front suspension system model

III. PARALLEL WHEEL TRAVEL SIMULATION AND EXPERIMENTAL VERIFICATION

Set the suspension parameters according to the actual vehicle conditions, including sprung quality, tread, tire free radius, position of center of mass, and do the shakedown test to the model. Carry on the parallel wheel travel, setting the height 100 mm. Related curves can be obtained by Processor Post in ADAMS/CAR. The change law of the parameters and the range of the parameters can be shown in the fig. Passing the changes of curve and gradient of toe and camber angle, to see the change range of the initial value of the design and the wheel travel whether to big or not. Finally analyze the causes of the excessive wear and the handling stability of the tire.

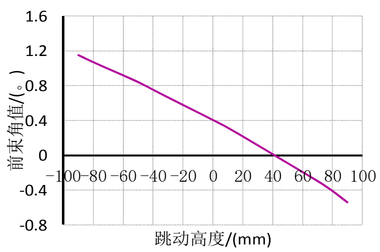


Fig.2 Change curve of toe angle with the wheel

The Effect of toe angle is reducing or eliminating the reason that the adverse consequences of the front wheel camber, which caused by rolling out during the car forward[2]. The toe angle increases with the increase of camber angle, and decreases with the decrease of the camber angle[3]. The variation range of the graph is from 1.15 to -0.5 degree. While in The Car Chassis Design[2,3], the recommended range of 0 to 0.6 degrees Although the negative slope of the former is more reasonable, the change range is too large, which made the tire wear increasing to a certain extent. It means that the toe angle needs further optimization.

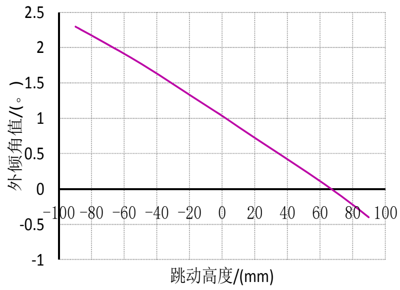


Fig.3 Change curve of camber angle

Camber angle made the front wheel having the trend of turning outward. The toe angle have trend to making the wheels turn inward, which can be brought about by camber offset the adverse effects, making the wheels and linear rolling without transverse skid phenomenon, ultimately reduce tire wear[5]. Taking the steering performance into account, the range of camber angle was recommended from 0 to 1.5 degrees in Automotive Chassis Design. Figure 3 shows that the change range of camber angle is -0.4 to 2. 3 degrees. Negative change of the negative slope to improve the adhesion, but the change range is too large, meaning that needing for further optimization.

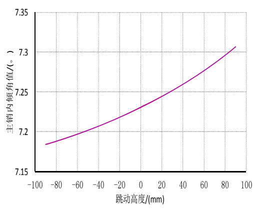


Fig.4 Change curve of kingpin inclination angle

The function of the kingpin inclination angle is to reduce the steering force, reduce the rebound and the phenomenon

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of deviation, and improve the stability of linear driving vehicle [4]. From Fig. 4 we can see that the angle of the main pin is 7.19 ~ 7.31 degrees while wheel travel. In the actual design, the range of parameters was rest with the structure of the different decisions. In the Automotive Chassis Design, the recommended range of 7 to 13 degrees, means that the range is in reason, the kingpin inclination angle will not need optimization.

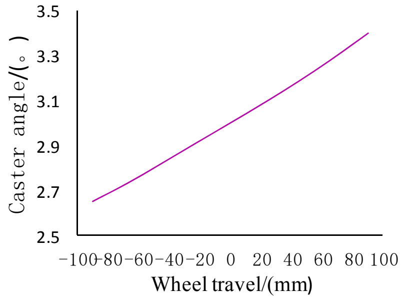

Fig. 5 Change curve of caster angle

The main role of the caster angle is to reset the wheel and improve the stability of the straight line, resulting in a positive torque to make the car in the driving force if the occasional external force can automatically return to the car steering wheel [5]. In the Automotive Chassis Design, the recommended range of 2 to 4 degrees, the range of variation is 2.65 to 3.4 degrees can be achieved in fig 5. The result shows that the caster angle meet the design requirements.

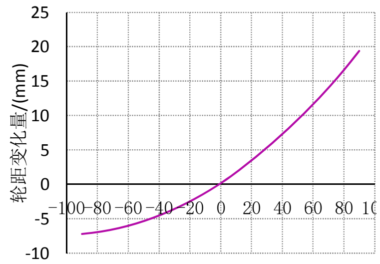

Fig.6 Change curve of wheel rate

From the graph 2-5, there is a negative trend in the toe angle and camber angle, meanwhile, the kingpin inclination and caster angle are the positive slope, which meets the design requirements. At the same time, the position of the wheel is zero, that is, the initial position of the toe angle is 0.39 degree, which is 7.3 mm in conversion with tire rolling radius. Camber angle is 0.99 degrees; the caster angle is 3 degrees; the kingpin inclination angle is 7.23 degrees; wheelbase is 2000 mm. By the auto design manual chassis design article[7] and automobile chassis design edited by Wang Xiaofeng[6] and other documents, listing for the recommended ranges of large bus front wheel alignment parameters as shown in Table I.

TABLE I. RECOMMENDED VALUES AND INITIAL VALUES FOR FRONT WHEEL ALIGNMENT PARAMETERS

Positional Parameter	**Recommended Value**	**Design Value**	**Whether Reasonable**
Camber angle (°)	0～1.5	1	reasonable
Toe angle (°)	0～0.6	0.39	reasonable
Caster angle (°)	2～4	3	reasonable
Kingpin inclination (°)	3～9	7.23	reasonable

Analysis results from table 1 are as follows:

The variation range of the camber angle is from -0.4 to 2.3 degrees;

The variation range of the toe angle is from -0.5 to 1.15 degrees;

The variation range of the caster angle is from 2.65 to 3.4 degrees;

The variation range of the kingpin inclination angle is from 7.19 to 7.31 degrees;

The maximum value change of wheel rate is 19.46 mm.

Table I shows that the design of the position parameters of the suspension is basically reasonable, and it is within the scope of industry recommendation. But from the point of view of kinematics, the change of the toe and camber angle is slightly larger, and the

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change trend is not gentle. The change range of the toe and camber angle is too large, which may cause the abnormal wear of the tire, such as the outer side of the shoulder come into being tire wear shape of sawtooth.

IV. HARD POINT OF DOE OPTIMIZATION DESIGN

A.Optimization Method

Analysis that the outside of the front tire wear seriously, the front suspension guide mechanism hard point layout unreasonable is indeed. This is consistent with the previous speculation. In this paper, we use the method of the full factorial experiment design. The full factorial experimental design was used to form different experimental conditions, which were involved in the experiment. The experiment was conducted under the condition of two or more than two times of the experiment [5].

B. Optimization Scheme

The front air suspension in this study is a symmetrical layout scheme. Taking the parallel wheel travel test into consideration, the change of the tripod inertia frame and the upright inertia frame is very insensitive. At the same time, the kingpin with the upper and lower arm of the movement of the toe angle and camber angle and other changes have no direct impact during the wheel travel, the hard point can be ignored. Combined with the layout scheme of the vehicle, the hard point coordinates of uca_outer_x, lca_outer_x, uca_outer_ y, lca_outer_y, lca_front_z, uca_oute_zr, lca_outer_z, uca_front_z were determined. Taking into account the design requirements and installation requirements, the final design variable is determined (-10--10) mm.

C. Optimization Objective Setting

In this paper, the design variables of the front suspension correlation are used as design variables, and the sensitivity analysis is based on the experimental design. The factors that may influence the changes of the toe and the camber angle are designed to determine the final optimization direction, and optimization direction is ensured finally. Due to the role of guide rod system, ensure the tire ground point caused by the track changes is relatively small beening required in test, which would prevent excessive tire wear.

According to the related parameter changes caused by the tire wear and power consumption that influence: toe angle, camber angle and the wheel rate changes caused by tire wear certain relationship exists and can be used as the weights of the objective function optimization. Among them, the change of the toe angle and wheel rate has great effect on the tire wear, which is about 2.57 times of camber angle. Finally, the weight value of the optimized target is 1:2.57:2.57.

After multiple virtual experiments, the sensitivity of the specific factors to the optimization objective is determined as Tab II:

TABLE II. SENSITIVITY ANALYSIS

Variable	Design Initial Value (mm)	Toe Angle（%）	Camber Angle(%)	Wheel Rate(%)
uca_front_z	147	-2.6	10.9	-15.7
lca_front_z	-220	-13.7	6.8	-20.9
uca_outer_x	1.5	4.2	11.2	-27.1
uca_outer_y	672	10.9	22.8	3.4
uca_outer_z	138.5	39.2	36.7	23.6
lca_oute r_x	1.5	-1.6	-0.5	8.1
lca_outer_y	703	9.8	23.0	8.9
lca_outer_z	-226.5	-17.8	-24.2	29.9

The analysis of the uca_outer_z is the most sensitive to the sensitivity of the toe angle, the second is lca_outer_z，which is -17.8% ; and the third is the lca_front_z,which is -13.7%. The sensitivity of uca_outer_z (upper arm external point) to the front camber angle is the 36.7% largest, followed by -24.2% (lca_outer_z), and lca_outer_y (lower arm) is 23%. Lca_outer_z to wheel rate changes affect the sensitivity of 29.9% is the largest, followed by uca_outer_x (upper arm exterior point) -27.1% ,third is uca_outer_z (upper arm exterior point) ,which is 23.6%.

D. Optimization Objective Function

Combined with the above analysis, the final objective function is shown in formula (4-1):

$$F(x) = \min\left[|m(x)| + 2.57|n(x)| + 2.57|s(x)|\right] \quad (4\text{-}1)$$

Among them, m (x), n (x), s (x) respectively camber angle, toe angle and wheel rate change quantity, f (x) is the objective function of the test.

· According to the actual situation of the bus structure, the constraint conditions are obtained:

$$\begin{cases} 0 < m(x) < 2 \\ 0 < n(x) < 0.6 \\ -10 < s(x) < 10 \end{cases}$$

The toe angle coefficient of the regression function is obtained by ADAMS/Insight, which is finally obtained as follows：

$$\begin{aligned} n(x) = {} & 4.6237 + 0.001234x_1 + 0.000832x_2 - 0.002984x_3 + 0.004872x_4 \\ & + 0.03792x_5 - 0.005228x_6 - 0.004764x_7 + 0.10977x_8 + 0.0061786x_1x_3 \\ & + 0.0023712x_1x_4 - 0.009981x_2x_4 + 0.007012x_2x_5 + 0.101982x_5x_7 \\ & - 0.007312x_5x_8 - 0.0019261x_1x_2x_5 - 0.010101x_5x_7x_8 + 0.0010593x_2x^{5^2} \\ & + 0.010378x_5x^{8^2} - 0.000239x_5^3 + 0.000094x_8^3 \end{aligned} \quad (4\text{-}2)$$

In formula (2), $x_1, x_2 \ldots\ldots x_8$ is the variable quantity,the scope of them are（-10--10）mm.

The regression function of camber angle and wheel rate would also obtained in the same way.

V. OPTIMIZATION RESULTS ANALYSIS AND EXPERIMENTAL VERIFICATION

A. Optimization Results Analysis

In this paper, there will be three times of fitting optimization to the variable quantity. After optimization, the maximum change of the hard point coordinates is 7.2 mm, and the following hard point position is adjusted. The model is established and the Wheel Travel Parallel 100 mm is performed. The comparison between initial value and optimized value of hard point is as shown in Table III:

TABLE III. OPTIMIZATION AND COMPARISON OF THE COORDINATES BEFORE AND AFTER OPTIMIZATION

Variable	Initial Value（mm）	Optimized Value（mm）
uca_front_z	147	149.2
lca_front_z	-220	-226.8
uca_outer_x	1.5	-0.84
uca_outer_y	672	679.2
uca_outer_z	138.5	139.7
lca_oute r_x	1.5	-0.79
lca_outer_y	703	699.7
lca_outer_z	-226.5	-229.6

The simulation of the optimized model was in progress. After the experiment, the corresponding parameters were compared before and after optimization respectively in the post-processing module, and the corresponding parameters were analyzed to achieve the optimization objective. Main analysis results are shown as follows:

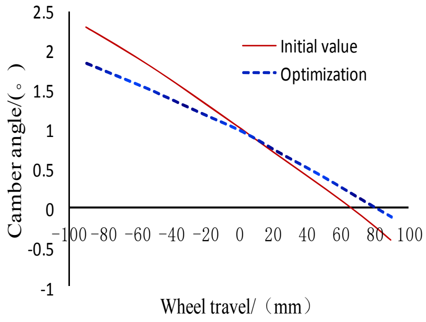

Fig. 7 Comparison of camber angle

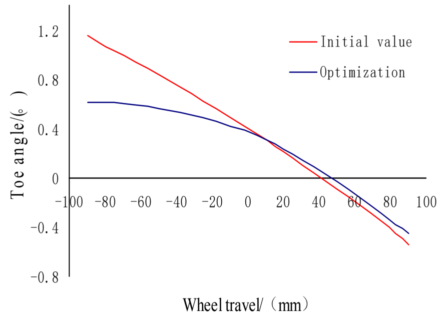

Fig.8 Comparison of toe angle

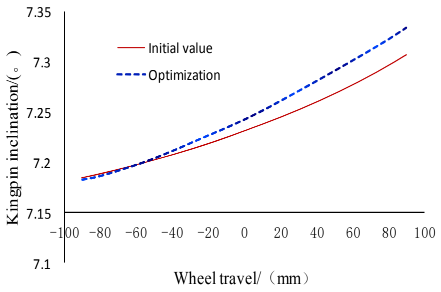

Fig.9 Comparison of kingpin inclination angle

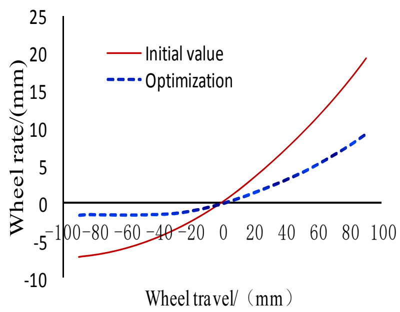

Fig.10 Comparison of wheel rate

TABLE IV. IMPROVEMENT RESULTS CONTRAST

Parameters	Initial Value	Optimization	Change	Change Range
Camber angle(°)	-0.4～2.3	-0.12～1.83	-0.75	-27.8%
Toe angle(°)	-0.5～1.15	-0.42～0.62	-0.61	-36.9%
Kingpin inclination angle(°)	7.19～7.31	7.18～7.32	+0.02	+16.7%
Caster angle(°)	2.65～3.4	2.64～3.46	+0.07	+9.3%
Wheel rate changes(mm)	-7.37～19.46	-1.79～8.76	-16.28	-60.7%

From figure 7--10 achieve the conclusion that the two curves of the graph represent the variation of the parameters of the corresponding parameters. In Figure 7, camber angle changes from the original range -0.4 to 2.3 degrees down to --0.12 to 1.83 degrees. In the same way, the variation range of the toe angle is from -0.5 to 1.15 down to the range of -0.42 to 0.62 degrees. In Figure 7,the variation range of the kingpin inclination angle is changed from 7.19 to 7.31 degrees 7.18 to 7.32 degrees. The variation of wheel rate changes from -7.37 ~ 19.46 mm down to -1.79 mm ~ 8.76 mm. By the analysis, it can be concluded that the variation range of the front camber angle is reduced by 27.8%, and the range of the front toe angle is reduced by 36.9%. The change of the caster and kingpin inclination angle is slightly larger, but it still meets the design requirements. The wheel rate change range is greatly reduced, the biggest changes in the amount of 8.76 mm, reduced by about 60%. Finally got the conclusion: the tire wear is reduced and the life-span is increased.

B. Test Verification

According to the optimization results of hard point coordinates in Table IV, adjusting the structure of the real vehicle, and the correctness of the virtual prototype is verified by real vehicle test and model simulation. Using the four wheel alignment instrument for real vehicle verification. As shown in Figure 11 - 12:

Fig.11 Front wheel clamps and probe

Fig.12 Corner plate

Analysis results are shown in figure 13:

The fold line represents the actual vehicle beam value verification curve. The simulation results are consistent with the trend of the real vehicle test that can be found in Figure 13. The error is kept within 13%, which meets the design requirements.

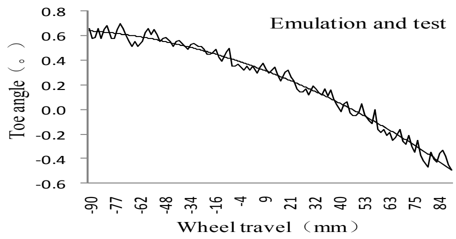

Fig. 13 Test validation of toe angle

VI.CONCLUSIONS

Established a simulation model, and the change curve of front wheel alignment parameters and its regular pattern are obtained by analyzing the model, which verifies that the front suspension steering mechanism hard point layout are unreasonable, which directly leads to a severe tire wear.

Aiming at the excess variation of toe angle and camber angle during the bus in the design, this paper utilized the air suspension part of the hard point of optimization by ADAMS/INSIGHT, reduced the scope of its changes ultimately to a certain extent.

The related hard point of optimization in this research is only relative to the best value in a certain extent. The optimization result resolved the major parameters of the front wheel alignment of the commercial vehicle in the case of considering the vehicle body layout. The results show that the optimization designs reduce the tire wear and operation stability of the bus.

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