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Simulation analysis and optimization design of front suspension based on ADAMS

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1. Introduction

Suspension system transfers force and torque between wheel and automobile frame, guides and controls the relative motion of the wheel and automobile body, eases up the impact from road surface and the shock from attenuation system [1-3]. Kinematic characteristics of the suspension refer to variation regularities of parameters such as kingpin inclination angle, camber angle, caster angle and the like when automobile wheels jump up and down, namely, wheels and vehicle body have relative movement in the vertical direction. Rational choice of suspension structure and performance parameters has a great and direct influence on the traveling smoothness, steering stability and comfortableness of the automobile [4]. Therefore, a suspension system is one of important assemblies for modern automobiles. An analysis on the kinematic characteristics of the suspension is a precondition for rational selection of the suspension and geometric parameters of a suspension guide mechanism. The double wishbone independent suspension is one of the suspension types that are widely used in light cars. Tread and front wheel alignment parameters change can be controlled within acceptable limited scope by selecting proper upper and lower wishbone lengths and rational deployment, thus to ensure the automobile is of good running stability. Currently, the double wishbone independent suspension has been widely used in the front and rear suspensions of Sedan [5].

Since the suspension system is a comparatively complex multibody system, the motion relation of its components is also complicated. This makes the traditional method for computing and analyzing various suspension features quite difficult. Currently, with the application and development of the virtual sample machine simulation technology, as well as people's continuously improved requirements to the automobile performance, using ADAMS provided analysis method to design parameters and verify automobile motion performance during the automobile independent suspension system optimization design can effectively improve the suspension design speed and quality [6]. For the double wishbone independent front suspension system, this paper adopts ADAMS/View to establish the virtual sample machine simulation model to analyze the geometrical parameters of the suspension and the wheel alignment parameters, and carry on parameter optimization design, thus to improve the suspension system motion performance [7].

2. Establishment of suspension simulation model

ADAMS software describes the space configuration of an object based on Cartesian Coordinates and Eulerian angle parameters; a problem on the solution of sparse matrix is solved by means of Gear rigid integration; ADAMS/Solver provides a plurality of solvers with mature functions, which can perform kinematics, statics and kinetics analysis on an established model [8].

2.1. Selection of generalized coordinates

In ADAMS software, the selection of generalized coordinates has a direct influence on the solution speed of a kinetic equation. Eulerian angle which reflects orientation of a rigid body and mass centric Cartesian coordinates of rigid body i are regarded as generalized coordinates, namely, $q_i = \left[x, y, z, \psi, \theta, \varphi\right]^T$, $q = \left[q_1^T, \ldots, q_n^T\right]$. Each rigid body is described by six generalized coordinates. Due to the application of generalized coordinates which are not independent, kinetic equation sets of the system have the biggest number; they are differential-algebraic equations which are coupled at a highly sparse degree, applicable for methods of sparse matrixes for efficient solution [9].

2.2. Establishment of kinetic equation

An ADAMS program applies a Lagrangian multiplier method to establish a motion equation of a system; mass centric Cartesian coordinates of a rigid body and an Eulerian angle which reflects orientation of a rigid body are regarded as generalized coordinates in the ADAMS, namely, $q_i = \left[x, y, z, \psi, \theta, \varphi\right]^T$; set $R = \left[x, y, z\right]^T$, $\gamma = \left[\psi, \theta, \varphi\right]^T$ and $q = \left[R^T, \ldots, \gamma^T\right]$, wherein q is a mass centric Cartesian coordinate; R is a mass centric position coordinate; γ is a mass centric Eulerian angle coordinate; three unit vectors of the coordinate system are axes of above three Euler's rotations respectively; therefore the three axes are not vertical to each other. A coordinate transformation matrix from the coordinate system to a mass centric coordinate system of a component is as follows

$$B = \begin{bmatrix} sin\theta\, sin\psi & 0 & cos\theta \\ sin\theta\, cos\psi & 0 & -sin\theta \\ cos\theta & 1 & 0 \end{bmatrix}$$

Angular velocity of the component can be expressed as $\omega = B\dot{\gamma}$; a variable ω_e in introduced in ADAMS as a component of the angular velocity in an Euler's rotation axis coordinate system $\omega_e = \dot{\gamma}$.

In consideration of a constraint equation, ADAMS takes advantages of an energy form of a Lagrangian equation of the first category with a Lagrangian multiplier to result the following equation

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_j}\right)-\frac{\partial T}{\partial q_j}=Q_j+\sum_{i=1}^{n}\lambda_i\frac{\partial \psi}{\partial q_j} \quad (1)$$

where T is kinetic energy expressed in generalized coordinates of the system; q_j is generalized coordinates; Q_j is a generalized forced in the direction of generalized coordinates q_j; the final item relates to a constraint equation and a Lagrangian multiplier which express a constraining force in the direction of the generalized coordinate q_j [10].

2.3. Establishment of suspension simulation model

Physically abstract the practical structure of the double wishbone front suspension to get the space structure topology as is shown in Fig. 1.

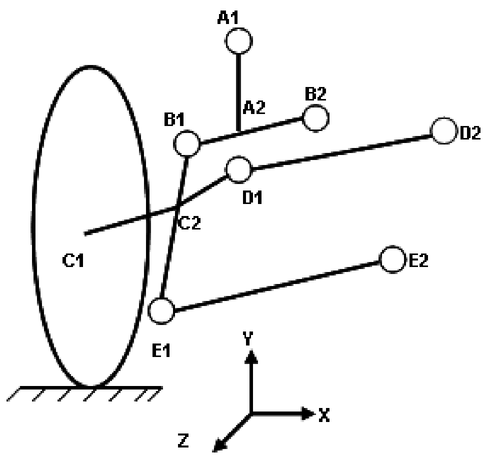

Fig. 1 Left-side of double wishbone suspension (E_2 – cylindrical link)

The shock absorber is simplified as linear spring, and the friction within kinematic pairs shall be ignored; the tyre is simplified as rigid body. Considering that the automobile is basically a vertical symmetrical system, this paper sets up the left side of suspension model. The coordinator value of the key design points of the left suspension system are as shown in Table 1.

Table 1
Main design point coordinate of the left side of the double wishbone suspension

Hard point	Coordinate (mm)		
	X	Y	Z
A1	174.6	637.89	24.85
A2	174.6	347.89	24.85
B1	52.38	330.72	10.23
B2	496.07	87.47	-79.78
C1	-236.92	103.18	2.44
C2	17.04	107.61	3.33
D1	439.55	181.19	-252.5
D2	-26.95	100	-170.71
E1	0	0	0
E2	386.4	389.62	33.95

The main alignment parameters of double wishbone suspension are shown as in Table 2.

Table 2
Main alignment parameters of double wishbone suspension

Parameter name	Kingpin inclination (°)	Kingpin caster (°)
Parameter value	9	1.75
Parameter name	Front wheel camber (°)	Toe angle (°)
Parameter value	1	0.2

3. Simulation analysis

During the suspension kinematics simulation, the kinematic pairs and linear drive shall be selected and added between the test platform and ground to represent the ground's excitation effect to the wheel; the selected wheel hop-up and hop-down travel shall both be 50 mm [11-13]. The excitation function of the ground to the wheel is as follows $F = 50\times sin(360d\times time)$. Setting simulation time is 1 s, working step is 100.

This paper mainly simulates four main alignment parameters of the front suspension and the relation of front wheel side slippage changing with the vertical direction wheel run-out. Through the simulation curve from Fig. 2 to Fig. 5, it can be seen that the variance of kingpin inclination, kingpin caster, front wheel camber, and front wheel toe angle before and after suspension optimization is small during the wheel up-and-down run-out process: change scope of kingpin inclination angle is 9° ~ 9.3° as shown in Fig. 2; kingpin caster change scope is 1.51° ~ 2.32° as shown in Fig. 3; front wheel camber change scope is 0.71° ~ 1° as shown in Fig. 4; and front wheel toe angle change scope is -0.07° ~ 0.21° as shown in Fig. 5.

It can be seen from Fig. 6, front wheel side slippage has comparatively large variance during the up and down run-out, -6.92 mm ~ 9.89 mm. However, large front wheel side slippage variance will result in tyre abrasion and damage to operating stability. Next, the optimization computing for suspension parameters will be carried out based on the target of minimizing the front wheel side slippage variance during wheel run-out process.

4. Model optimization and analysis

For the problem of too large front wheel side slippage and serious tyre abrasion, this paper takes upper suspension length, lower suspension length, and the landscape planar dip angles of the upper and lower suspensions in the automobile as the design variables, and takes minimizing front wheel side slippage change as the target of the optimization in the ADAMS/View module. In order for the convenience of optimization, this paper takes the absolute value of target function front wheel side slippage.

Considering the limiting of suspension elements layout in real vehicle, the standard value and upper and lower limits of the design variables are as shown in Table 3.

The rules of change of main alignment parameters and front wheel side slippage after optimization are as shown in Figs. 2 - 5. The change scope of kingpin inclination, kingpin caster, front wheel camber, and front wheel toe angle before and after suspension optimization is small. The comparison of the main alignment parameters of the suspension before and after optimization is as shown in Table 4.

Table 3
Standard value and upper/lower limit of design variables

Variable name	Standard value	Minimum value	Maximum value
Upper suspension length, mm	340	300	400
Lower suspension length, mm	510	480	550
Upper suspension land-scape planar dip angle, °	10	8	15
Upper suspension land-scape planar dip angle, °	10	5	15

Table 4
Data comparison before and after optimization

Front wheel alignment parameters	Before optimization	
	Change scope	Variance
Kingpin inclination, °	9 ~ 9.3	0.3
Kingpin caster, °	1.51 ~ 2.32	0.81
Front wheel camber, °	0.71 ~ 1	0.29
Toe angle, °	-0.07 ~ 0.21	0.28
Front wheel alignment parameters	After optimization	
	Change scope	Variance
Kingpin inclination, °	9 ~ 9.24	0.24
Kingpin caster, °	1.56 ~ 2.2	0.64
Front wheel camber, °	0.77 ~ 1	0.23
Toe angle, °	-0.01 ~ 0.21	0.22

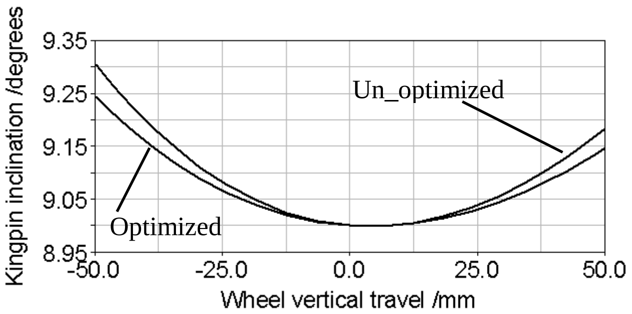

Fig. 2 Kingpin inclination and wheel bouncing distance

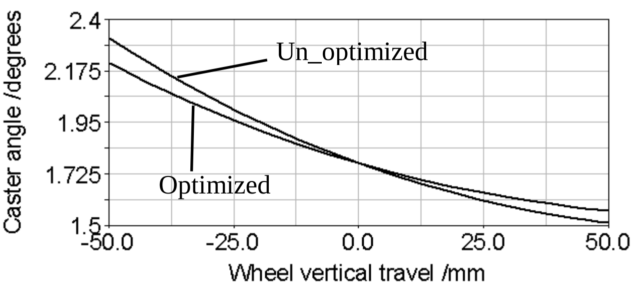

Fig. 3 Caster angle and wheel bouncing distance

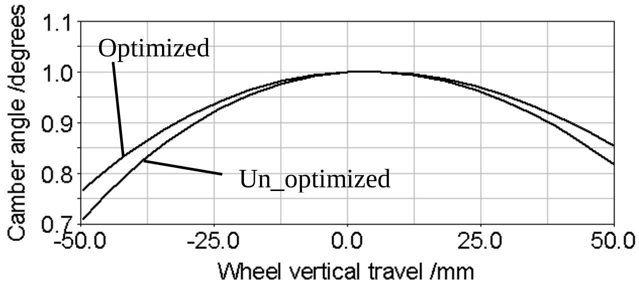

Fig. 4 Camber angle and wheel bouncing distance

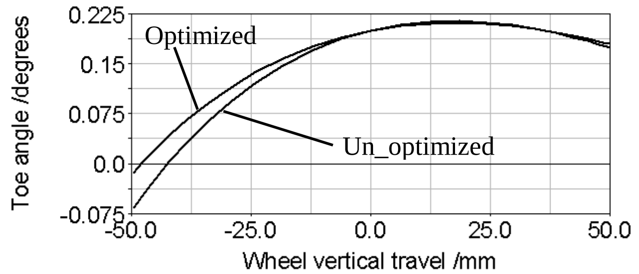

Fig. 5 Toe angle and wheel bouncing distance

Fig. 6 is the change curve of the absolute value of the front wheel side slippage before and after optimization. The optimized front wheel side slippage change scope is 0 mm-1.17 mm with only 0.17 mm variance.

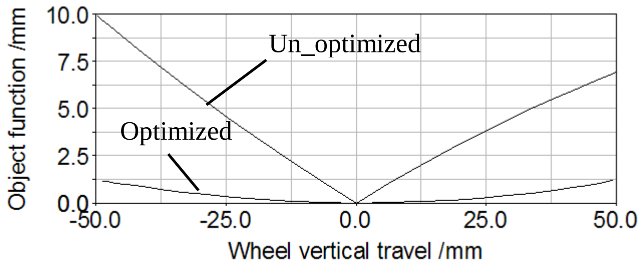

Fig. 6 Absolute value of wheel lateral travel and wheel bouncing distance

5. Conclusion

In this paper, the ADAMS/View module is adopted to set up double wishbone independent suspension kinematics model for simulation. For the problems found in the analysis, it optimizes suspension design point coordinates for many times based on the target of minimizing the wheel side slippage and get the optimized result. It satisfies the design requirement by adjusting relative alignment parameters to settle the problems of too large alignment parameter change and serious tyre abrasion during suspension run-out. But due to the limit of automobile body deployment, the change to the design point coordinates can only be limited in a small scope so that the optimized value is only a relative value not absolute optimized value. However, the virtual sample machine technology can effectively simulate environment, reduce physical sample machine experiment times, and greatly reduce the development fee and cost. To some extent, this technology has certain meaning of guidance [14-17].

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PRIEKINĖS PAKABOS IMITAVIMO ANALIZĖ IR PROJEKTAVIMO OPTIMIZAVIMAS NAUDOJANTIS ADAMS PROGRAMA

R e z i u m ė

Straipsnyje naudojama ADAMS programa sukurti dviejų svirčių pakabos kinematikos analizės modeliui. Juo nustatyti priekinių ratų padėties parametrų kitimo dėsningumai minimizuojant ratų išdilimą ir atsižvelgiant pakabos sistemos eksploatacinių duomenų racionalumą. Siekiant minimizuoti per didelį šoninį slydimą ir didelį priekinės pakabos ratų išąčdilimą, atliktas pakabos optimizavimo skaičiavimas. Gauti optimizavimo rezultatai tam tikru mastu pagerina pakabos sistemos darbą.

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SIMULATION ANALYSIS AND OPTIMIZATION DESIGN OF FRONT SUSPENSION BASED ON ADAMS

S u m m a r y

It uses ADAMS software to establish the double wishbone suspension kinematics analysis model to analyze the rules of front wheel alignment parameter changing with wheel run-out and evaluate the rationality of the suspension system data. For the problem of too much side slippage and serious abrasion of front wheel of suspension, it carries out optimization computing for the suspension based on the target of minimizing the front wheel side slippage during wheel run-out. And the optimization result improves the suspension system performance in a certain extent.

Keywords: Double wishbone suspension, ADAMS, optimization, kinematics simulation.

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