

Comparative Analysis on a Rectilinear Independent Suspension with Traditional Ones

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ABSTRACT

Suspension plays an essential role in vehicle's handling stability and riding comfort. This paper discusses a novel suspension that has the capacity to trace a straight line in theory. Therefore it is called rectilinear suspension. So the alignment parameters are invariable during jounce and rebound if the elasticity of suspension components is ignored. According to the structure characteristics of the rectilinear suspension, it is suitable to mount on the rear axle. To evaluate its performance, the dynamics model is established through ADAMS. Moreover, a comparison of the rectilinear suspension with the twin-trapezoidal link suspension is carried out. Further, the K&C test results show that the alignment parameters of the rectilinear suspension are almost invariable compared with MacPherson suspension.

KEYWORDS: Rectilinear Suspension, MacPherson suspension, Twin-trapezoidal Link Suspension, Kinematic and Compliance, K&C Test

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INTRODUCTION

Handling stability and ride comfort of a vehicle are closely related to the performance of suspension system. Hence, stringent requirements of the suspension design are demanded as high speed safety and comfort are aspired today. At present, vehicle suspension system mainly includes independent suspension and dependent suspension whose mainly difference is the guidance mechanism, which determines the alignment parameters of the wheels. It is the major goal of suspension guidance mechanism synthesis to ensure that the wheel alignment parameters remain a range as small as possible [1]. To achieve this purpose, researchers mainly focused on two research aspects, one is the kinematics analysis and optimization of suspensions [1, 2, 3, 4, 5] and the other one is the innovation design of suspension mechanisms [6, 7, 8]. Kinematics analysis and optimization mainly cover kinematic and compliance (K&C) simulation [1, 2, 3] and optimization of the hard points [4, 5] with ADAMS. On the other hand, Zhao et al. [6, 7, 8] proposed a kind of suspension mechanism consisting of four identical 3-revolute joint (4-RRR) chains with the rectilinear guidance function.

At present, the alignment parameters of traditional suspensions are changing within an interval during jounce and rebound. The kinematic characteristic is extremely sensitive to the hard points of suspension mechanism. Conversely, the new suspension mechanism proposed in [6, 7, 8] can overcome the shortage of traditional ones.

This paper synthesizes a four Revolute-Prismatic-Revolute chain (4-RPR) suspension and verifies its rectilinear guidance function by mobility analysis first. And then its dynamics model is established through ADAMS to investigate the K&C (kinematic and compliance) characteristics. Moreover, a comparison of this suspension with the

twin-trapezoidal link suspension is carried out. Further, the K&C test results show that the alignment parameters of the rectilinear suspension are almost invariable with respect to MacPherson suspension.

SYNTHESIS OF RECTILINEAR MECHANISM

The design target of the guidance mechanism for a suspension is to keep the alignment parameters changing within a range as small as possible. So the ideal suspension guidance mechanism should have a rectilinear guidance function. That is to say, the ideal suspension mechanism should constrain 5 degrees of freedom of the knuckle, including two translations, and three rotations. For the 4-RRR suspension mechanism, every RRR kinematic chain provides 2 constraint moments and 1 constraint force. Based on the screw theory, replacing the middle revolute joint of RRR kinematic chain with a prismatic joint yields an RPR kinematic chain, which has the identical terminal constraints and free motions with those of RRR kinematic chain. So the 4-RPR suspension mechanism has the same rectilinear guidance function with 4-RRR mechanism.

As Fig. 1 illustrated, kinematic chains ABC and DEF are both RPR kinematic chains. Both revolute joints A and F are located on the XOY -plane, and they are symmetrical about the YOZ -plane. The distance between the center of revolute joint A and point O is R . Revolute joints C and D both connect with the end-effector, and their axes are parallel to those of the revolute joints A and F , respectively. The axes of prismatic joints B and E are perpendicular to the axes of its neighboring revolute joints shown in Fig. 1. The subtended angle of the plane which is perpendicular to the axis of revolute joint A and the YOZ -plane is $\theta \in [0, \pi]$. The angle between the RPR kinematic chain and the XOY -plane is $\alpha \in [-\pi/2, \pi/2]$.

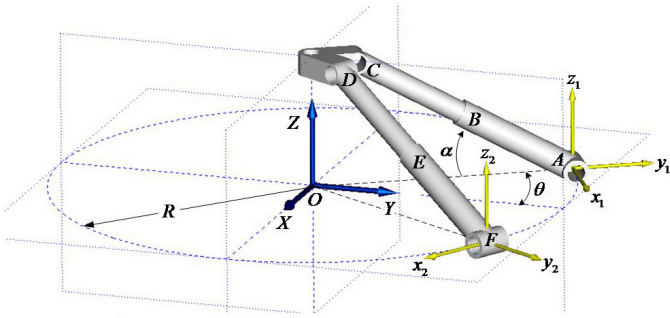


Fig. 1. Geometry model of a 2-RPR mechanism

The local coordinate system, $Ax_1y_1z_1$, is established whose x_1 -axis superimposes with the axis of revolute joint A , z_1 -axis is parallel with the Z -axis, and the y_1 -axis is obtained according to the right-hand rule. Assume that the length of AC is l , then the kinematic screws of the revolute joints of the kinematic chain ABC are expressed as:

$$\begin{aligned} \mathcal{S}_A &= [1 \ 0 \ 0 \ 0 \ 0 \ 0]^T, \\ \mathcal{S}_B &= [0 \ 0 \ 0 \ 0 \ -\cos \alpha \ \sin \alpha]^T, \\ \mathcal{S}_C &= [1 \ 0 \ 0 \ 0 \ l \sin \alpha \ l \cos \alpha]^T. \end{aligned}$$

Based on the reciprocal screw theory, the terminal constraints of the kinematic chain ABC are obtained

$$\mathcal{S}_1^r = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}^T$$

According to the expression, the terminal constraints of the kinematic chain ABC are one constraint force along the x_1 -axis and 2 constraint moments about the y_1 -axis and the z_1 -axis, respectively. Similarly, the terminal constraints of the kinematic chain DEF in the local coordinate system $x_2y_2z_2$ can also be obtained. As a result, the terminal constraints of the platform in $OXYZ$ can be expressed in matrix form:

$$\mathcal{S}^r = \begin{bmatrix} \cos \theta & \sin \theta & 0 & 0 & 0 & -R \\ 0 & 0 & 0 & -\sin \theta & \cos \theta & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ \cos \theta & -\sin \theta & 0 & 0 & 0 & -R \\ 0 & 0 & 0 & \sin \theta & \cos \theta & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix}^T \quad (1)$$

Therefore, when $\sin \theta \neq 0$ or $\cos \theta \neq 0$, that is $\theta \neq 0$ or $\pi/2$ or π , the rank of constraint matrix $\mathcal{S}^r$ is 5. Accordingly, the kinematic screw of the end-effector is

$$\mathcal{S}_{CD} = [0 \ 0 \ 0 \ 0 \ 0 \ 1]^T \quad (2)$$

Equation (2) shows that the 2-RPR mechanism merely has one degree of freedom along the Z -direction. So the 2-RPR mechanism can be used as the guidance mechanism of an independent suspension. Considering large loading capacity of suspension mechanism in

application, an extra 2-RPR mechanism is added to form a 4-RPR overconstrained suspension mechanism, which can improve the structure strength and loading capacity.

DYNAMIC MODELING AND SIMULATION

Dynamic Modeling

To evaluate the K&C characteristics of the rectilinear suspension, its dynamic model is established. So does the dynamic model of a twin-trapezoidal link suspension and MacPherson suspension equipped on Haima Family.

The vehicle parameters of the Haima Family are listed in Tab.1. The model parameters of two suspensions are the same except the hardpoints and rubber bushings. The stiffness of spring is $K_s = 19.61 \text{ N/mm}$. The equivalent torsional stiffness of the anti-roll bar is 24337.5 N.mm/deg . The MacPherson suspension is used as front suspension system. The spring stiffness is $K_{sm} = 21.9 \text{ N/mm}$, the torsional stiffness of the anti-roll bar is 85624.9 N.mm/deg . Tables from 2, 3, 4, 5, 6, 7 show the hardpoints and rubber bushings' stiffness of these suspensions.

Accordingly, the dynamic models are established. First, the suspension template model and anti-roll model are established. And then, the suspension assembly model is built up. For the sake of simplicity, more details about the dynamic modeling are not expressed here. The dynamic models of suspensions are shown in Fig. 2.

Tab. 1. Vehicle parameters of the Haima Family

No.	Name	Number
1	Wheel base(mm)	2610
2	Wheel track(mm)	Front 1470
		Rear 1470
3	Mass(kg)	Front 347.5
		Rear 227.5
4	Alignment parameters of front wheel	Toe angle(deg) 0.1
		Camber angle(deg) -0.8
	Alignment parameters of rear wheel	Toe angle(deg) -0.1
		Camber angle(deg) -0.5

Tab. 2. Hard points of the rectilinear suspension

No	Hard points	Coordinate(mm)			Symmetry
		X	Y	Z	
1	Bufo_Hi	2461.01	547.005	365	R/L
2	Bufo_Ho	2560	646	365	R/L
3	Bufo_Hi	2385	471	365	R/L
4	Bufo_Ho	2528.54	614.543	365	R/L
5	Buro_Hi	2758.99	547.005	365	R/L
6	Buro_Ho	2660	646	365	R/L
7	Buri_Hi	2835	471	365	R/L
8	Buri_Ho	2691.46	614.543	365	R/L
9	Blfo_Hi	2461.01	547.005	215	R/L
10	Blfo_Ho	2560	646	215	R/L
11	Blfi_Hi	2385	471	215	R/L
12	Blfi_Ho	2528.54	614.543	215	R/L
13	Blro_Hi	2758.99	547.005	215	R/L
14	Blro_Ho	2660	646	215	R/L
15	Blri_Hi	2835	471	215	R/L
16	Blri_Ho	2691.46	614.543	215	R/L
17	wheel_center	2610	735	290	R/L
18	strut_up_mount	2610	555.7	895	R/L
19	strut_down	2610	555.7	379	R/L
20	spring_down	2610	580.9	632.4	R/L
21	subframe_to_body	2610	0	290	S
22	subframe_front	2560	470	350	R/L
23	subframe_rear	2660	470	350	R/L
24	antiroll_strut	2682.42	528.7	386	R/L
25	antiroll_subframe	2502.6	300	234.64	R/L
26	antiroll_external	2682.42	536.93	245.32	R/L
27	antiroll_conner	2502.6	536.93	234.64	R/L

Notes: B-branch, H-hard point, i-inner, o-outer, m-middle, u-upper, l-lower, f-front, r-rear