International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:6, No:5, 2012

Kinematic Analysis of Roll Motion for a Strut/SLA Suspension System

Yung Chang Chen, Po Yi Tsai, I An Lai

***Abstract*—**The roll center is one of the key parameters for designing a suspension. Several driving characteristics are affected significantly by the migration of the roll center during the suspension's motion. The strut/SLA (strut/short-long-arm) suspension, which is widely used in production cars, combines the space-saving characteristics of a MacPherson strut suspension with some of the preferred handling characteristics of an SLA suspension. In this study, a front strut/SLA suspension is modeled by ADAMS/Car software. Kinematic roll analysis is then employed to investigate how the rolling characteristics change under the wheel travel and steering input. The related parameters, including the roll center height, roll camber gain, toe change, scrub radius and wheel track width change, are analyzed and discussed. It is found that the strut/SLA suspension clearly has a higher roll center than strut and SLA suspensions do. The variations in the roll center height under roll analysis are very different as the wheel travel displacement and steering angle are added. The results of the roll camber gain, scrub radius and wheel track width change are considered satisfactory. However, the toe change is too large and needs fine-tuning through a sensitivity analysis.

***Keywords*—**roll analysis, roll center height, steering, strut/SLA suspension, wheel travel

I. Introduction

THE kinematics of suspension components describes how important characteristics change as the suspension moves, typically in wheel travel, roll and steering. The kinematic relationship between the linkages and joints in an automotive suspension is very complex, and it has a significant influence on the stability, handling ability and ride comfort of the vehicle. Engineering tools, including dynamic analysis software and K&C rig testing, are usually used to design, test and tune the vehicle suspension concepts [1]. Kinematics has been defined in a general sense as the study of motion without reference to mass or force. Many design parameters relative to the static settings, such as the wheel alignment angles, roll center height, caster trail, scrub radius and spring motion ratio, basically need to meet the overall vehicle targets. Furthermore, the kinematics of the suspension, such as roll center height, track gain, camber gain, caster gain, Ackermann change with steering angle, roll steer and bump steer, also need to be verified [2].

Y. C. Chen is with the Automotive Engineering Department, National Pingtung University of science and technology, Taiwan. (phone: +886-918519261; fax: +886-8-7740398; e-mail: c123@ mail.npust.edu.tw).

P. Y. Tsai is a graduate student at the Automotive Engineering Department, National Pingtung University of science and technology, Taiwan. (e-mail: l23b23j23@hotmail.com).

I. A. Lai is a graduate student at the Automotive Engineering Department, National Pingtung University of science and technology, Taiwan. (e-mail: laiian@hotmail.com).

The strut/SLA suspension combines the space-saving characteristics of a MacPherson strut suspension with some of the preferred handling characteristics of an SLA suspension [3]. Although there are several dissimilar models, all strut/SLA suspensions have a few features in common, i.e., a lower and upper control arm, an integral-spring strut with a pivot bushing at its base and an anti-roll bar, in general, this is classified as a SLA suspension. The corresponding kinematic features of this type of suspension have received less attention.

The kinematics of an automotive suspension have been studied extensively [4]-[10]. These studies have generally evaluated variations in the corresponding parameters in regard to individual wheel vertical motion, rolling motion or steering. In this paper, a front strut/SLA suspension with steering system was modeled by ADAMS/Car software, and a roll analysis with parallel wheel travel and steering inputs was employed to investigate the rolling characteristics of this suspension. The results can further support the requirements of a tuning process through a sensitivity analysis.

II. Kinematic Parameters of Roll

Suspension motion can be viewed as a combination of suspension ride and suspension roll. In SAE's definition, suspension roll is the jounce or rebound displacement or velocity of a pair of wheels on the same axle, which is antisymmetric with respect to the vehicle plane of symmetry [11]. Several important kinematic parameters, such as the roll center and the related parameters of roll camber, toe change, scrub radius and wheel track width change, are described below.

A. Roll Center

The SAE defines the suspension roll center as the point at which lateral forces may be applied without producing rolling of the sprung mass. The roll center height is the distance from the roll center to the ground tire contact, measured on the vertical centerline of the wheel. The line connecting both the front and rear roll centers determines the roll axis. Every car rolls about the roll axis when subjected to a side force, and the amount of moment is related to the height of the roll center and the center of gravity [12].

The instant center is defined basically from a four-bar theory, and its location is determined by the steering and suspension geometries. Thus, it is only valid for one particular position of the wheel. The roll center is located at the vehicle centerline with a line connecting the instant center and wheel contact point, as shown in Fig. 1. An important assumption for the kinematic instant center and roll center is that the wheel, links, chassis and the virtual connecting lines are all considered as rigid. The kinematic roll center offers a design base for symmetric suspensions.

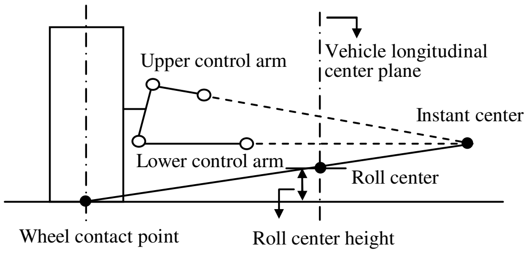

Fig. 1 Roll center and roll center height of the SLA suspension

B. Related Roll Parameters

Body roll at the chassis centerline affects camber gain or loss in the opposite direction to suspension bump. In general, body roll produces positive camber gain on the outside and negative camber gain on the inside, while outside bump produces negative camber gain and inside droop produces positive camber gain. The camber angle is a measure of the orientation of the wheel plane relative to the vehicle, and it is independent of road plane geometry. Fig. 2 shows the camber angle of the wheel.

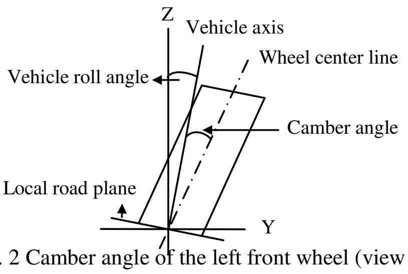

Fig. 2 Camber angle of the left front wheel (view from front)

The roll steer of the suspension is the change in steering angle resulting from a given suspension's roll angle. The toe change of the front wheels as the suspension goes from a normal ride height through full bump to full droop is called the bump steer. A large toe change in the roll or bump motion will cause a handling problem.

The kingpin axis offset at the ground is called the scrub radius. A larger scrub radius makes the steering stiff and increases tire wear. Moreover, the scrub radius being positive or negative will influence the toe specifications and braking stability.

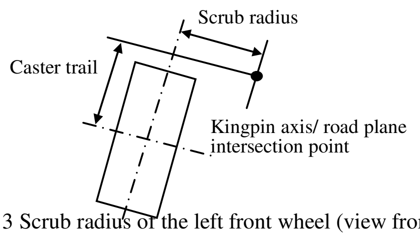

Fig. 3 Scrub radius of the left front wheel (view from top)

The wheel track width is the distance between the contact centers of a pair of tires on an axle. As the wheel moves up and down relative to the chassis of the vehicle, the wheel contact point rotates around an instantaneous center and passes through an imaginary arc in space with this rotation, as shown in Fig. 1. Thus, it may not only change the camber and caster angles, but also the wheel track width.

III. Kinematic Analysis

For this study, a front strut/SLA suspension of a production car was modeled by ADAMS/Car software as shown in Fig. 4, and the hard points were measured by a FARO arm. In addition, a rack and pinion steering system was also established with this half-car model. The gear ratio for the steering system was 0.11.

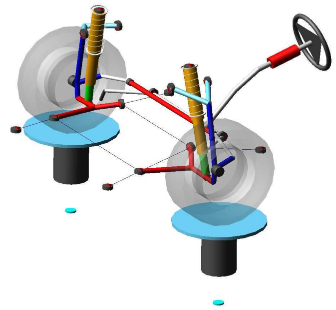

Fig. 4 Kinematic model of the strut/SLA suspension

A roll analysis was first performed under the chassis roll alone. Two real values of 5 and -5 were selected to fix the upper and lower limits of roll angle displacement. The orientation of the tables measures the roll angle. A positive value causes the left test-rig wheel to be moved upward and the right test-rig wheel downward.

Since wheel travel and steering are usually accompanied by the roll motion, the wheel's vertical displacement and steering angle were added separately to the roll analysis of the suspension test-rig.

The wheel jump range is based on the full load car, and a load case of bump movement of -50 mm~50 mm was selected for this study. A positive value represents the jounce, while a negative value represents the rebound. In jounce travel, the wheel and suspension components move upward and compress. In addition, the steering input was -500°~500°. A positive value means a left turn has been made.

After completing the roll analysis, the kinematic parameters of roll, such as roll center height, roll camber, toe change, scrub radius and wheel track width change under variations in wheel travel and steering angle, were evaluated and compared.

IV. Results and Discussion

Because the geometry of the left- and right-front strut/SLA suspensions are considered to be symmetric, only the left wheel data have been illustrated and analyzed in this study.

A. Roll Center Height

Fig. 5 shows the roll center height responses of the strut/SLA suspension under simultaneously applied the wheel travel and chassis roll. A linear range of roll center height and wheel travel during the full wheel motion was exhibited. The roll center height was 209 mm at the ride height, and the variation in roll center height from the complete rebound to complete jounce was approximately 140 mm. The roll angle of the car during lateral acceleration is dependent on the distance from the roll center to the center of gravity. The ratio of the front roll-couple to the rear roll-couple is one of several factors that determine the tendency of a car to either oversteer or understeer.

On an average strut suspension, the roll center usually lands about 25 mm to 75 mm from the ground. The SLA suspension typically places the roll center a bit farther up. In this case, the static roll center height and its variations seemed too large to have good handling. A high roll center transfers lateral force quickly and does cause jacking where the suspension rises over the contact patch of the tire.

In addition, the roll center height was lowered as the roll was added. It could be seen that the roll center height varied slightly (~10 mm) as the roll angle was within ±3°; after this roll angle level, the roll center height changed abruptly. The rapid movement of the roll center as the system experiences small displacements can lead to stability problems with the vehicle. In practical situations, the roll center may migrate away from the longitudinal plane of the car as weight is transferred from one side to another. When the car rolls by turning, the roll center height also changes somewhat with the steering angle, as shown in Fig. 6.

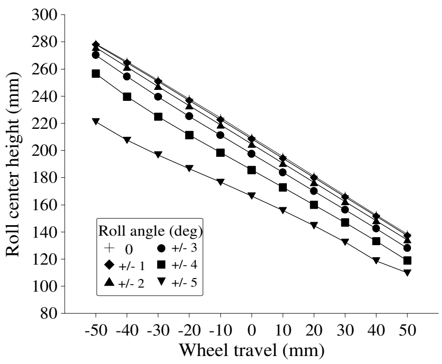

Fig. 5 Roll center height with respect to wheel travel under chassis roll

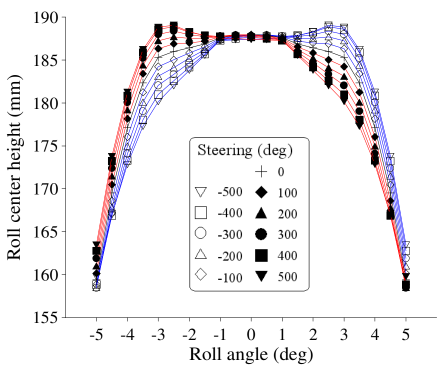

Fig. 6 Roll center height with respect to roll angle under steering

B. Roll Camber

Due to the non-zero caster angle of the suspension system, the camber roll effect results in the camber gain on front wheels when the vehicle is turning. Furthermore, the camber angle can change as the wheel moves through suspension travel and as the wheel turns about the steering axis. The amount of camber gain for wheel travel is determined by having different lengths and angles of upper and lower control arms. Furthermore, the camber changes more slowly with a longer front view swing arm, but can change drastically with a shorter one. A larger roll center height for this studied strut/SLA suspension actually decreased the camber gain. Figs. 7 and 8 demonstrate the roll camber responses of the left wheel with respect to the chassis roll under wheel vertical motion and steering inputs, respectively. Under the chassis roll input alone, it could be seen that the camber angle of the wheel went further negative with a small peak value near -0.25° in either a positive or negative chassis roll. As the suspension jounced, the net camber angle of the wheel increased in the positive direction with the increase of positive chassis roll, as shown in Fig.7. The camber angle increased in the negative direction with the increase of negative chassis roll. With the wheel under rebound, however, the net camber angle went further negative. The change rate increased gradually from negative chassis roll to positive chassis roll. The variation in the camber angle of the left wheel in this simulation was only approximately 1.3°.

The net camber varied with the steering input, as shown in Fig. 8. The left wheel exhibited positive net camber and negative net camber, respectively, as a left turn or right turn was added. It could be seen that the total change of the net camber angle in this measurement was about 2.5°.

In general, a suspension system must be designed to compensate for the camber angle change associated with chassis and wheel movements so that the maximum cornering forces are produced. The results, as shown in Figs. 7 and 8, suggested that the variation of the net camber angle of the wheel during the test was quite small.

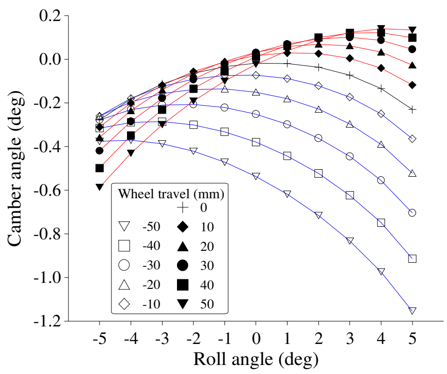

Fig. 7 Camber angle with respect to roll angle under wheel travel

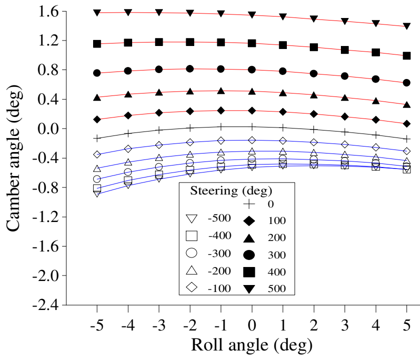

Fig. 8 Camber angle with respect to roll angle under steering

C. Toe Change

Bump steer is the term for the tendency of the wheel of a car to steer as it moves upwards, and it is dictated by the relative length and position of the tie rod and the upper control arm. Modern suspension systems are often designed with the geometrical intent of eliminating bump steer. In roll steer, one wheel rises as the other falls. Typically, this produces toe-in on one wheel and toe-out on the other, thus producing a steering effect.

Fig. 9 depicts the toe angle responses of the left wheel as a function of the roll angle with the variation of wheel travel. At the ride height, the wheel exhibited toe-in with a maximum angle of 0.6° under positive chassis roll, and a maximum toe-out angle of -1° under negative chassis roll. When the rebound was added, the toe angle of the wheel increased (toe-in or toe-out) with the increase of either a positive or negative chassis roll. However, the toe angle of the wheel went further toe-out with a peak value of -1.6° at the side of the negative chassis roll as the jounce was added.

The results showed the obvious toe-out of the front wheel as the suspension was compressed; therefore, the vehicle had a bump understeer. The total change in toe-out for the full bump approached -1.6°. From a design standpoint, this was not considered a good layout and a tuning process was needed.

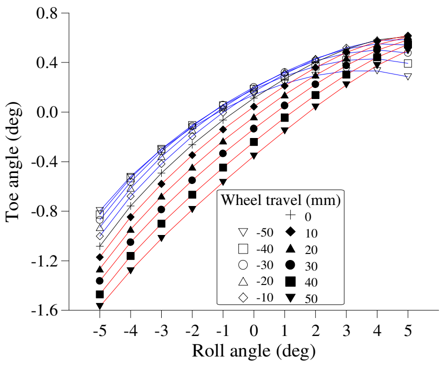

Fig. 9 Toe angle with respect to roll angle under wheel travel

D. Scrub Radius

The effect of the scrub radius is to provide a turning moment which attempts to turn the wheel away from the central position when the vehicle is in motion. A negative scrub radius decreases torque steer and improves stability in the event of brake failure. On a front-wheel-drive vehicle with a negative scrub radius, the vehicle's forward motion and the friction between the tire and the road cause a force which tends to move the front wheels back, which causes the wheels to toe-out.

Figs. 10 and 11 present the scrub radius responses of the left wheel in relation to the chassis roll under wheel vertical motion and steering inputs, respectively. Due to the change in the camber angle under the wheel's vertical motion, the scrub radius of the wheel also varied throughout its travel, as shown in Fig. 10.

As shown in Fig. 11, it can be seen the maximum scrub radius variation of 4.5 mm during a positive chassis roll was nearly the same as that for a negative chassis roll. There was only a small change in the scrub radius of the wheel when the steering motion was added. By SAE convention, positive roll is experienced in a left-hand turn where the right side is in bump and the left side is in droop. Thus, the scrub radius of the left wheel decreased with the increase of a positive chassis roll and increased during a negative chassis roll. The results suggested that the degree of scrub radius variation in the wheel travel and steering motion was acceptable.

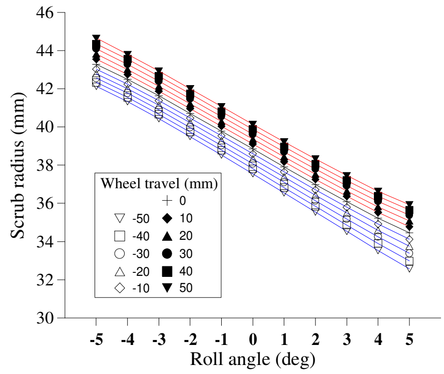

Fig. 10 Scrub radius with respect to roll angle under wheel travel

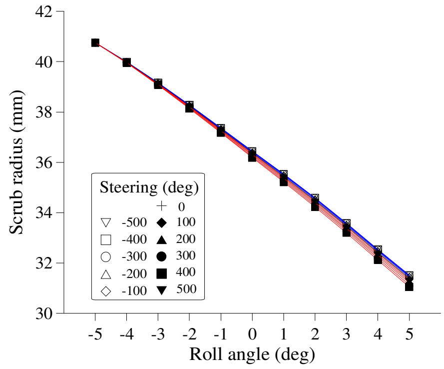

Fig. 11 Scrub radius with respect to roll angle under steering

E. Wheel Track Width Change

For a car with an independent suspension, the camber and the track width change when cornering or hitting bumps. Therefore, a lateral tire force occurs if the track width changes, which means that the maximum possible value of the longitudinal tire force decreases. In general, track width variation forces tires to slip; thus, anything which reduces grip must be kept to a minimum.

Fig. 12 shows the change in the wheel track width of the left wheel with respect to chassis roll under variations in wheel travel. It could be seen that the wheel track width decreased during wheel jounce and increased during wheel rebound. As the chassis roll reached 5°, the peak variations in wheel jounce and rebound were nearly -9 mm and 24 mm, respectively.

Fig. 13 illustrates the change in the wheel track width of the left wheel with respect to chassis roll under variations in steering motion. The track width change clearly increased as the positive steering motion increased. Generally, track width change is related to the layout of the control arms. The peak variations in this simulation test were considered to have met the requirements.

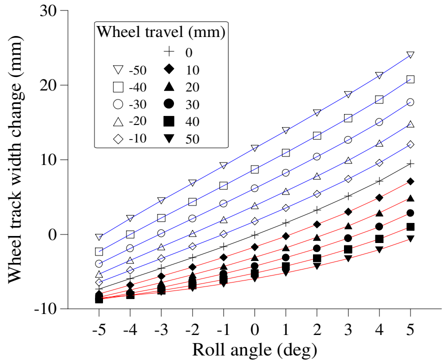

Fig. 12 Wheel track width change with respect to roll angle under wheel travel

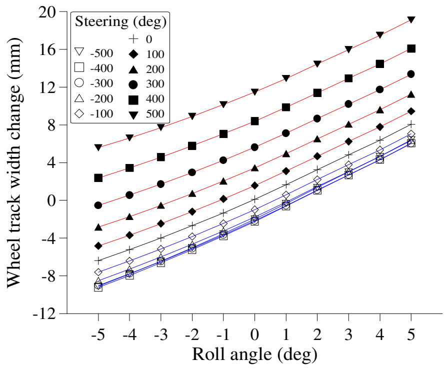

Fig. 13 Wheel track width change with respect to roll angle under steering

V. Conclusion

This paper has presented a roll analysis with the inputs of wheel travel displacement and steering angle for a strut/SLA suspension system.

The conclusions from this study are summarized as follows:

1. This strut/SLA suspension had a larger roll center height as compared to the strut or SLA suspensions; it may decrease the moment arm and thus reduce the roll effect. However, a higher roll center may cause jacking effects and erratic suspension movements.
2. The total change in toe angle for the full bump approached -1.6° during the negative chassis roll. This meant there was obvious front wheel toe-out as the suspension was compressed; therefore, the vehicle had a bump understeer. From the design perspective, this was not considered a good layout and tuning was needed.
3. Compared to the average car, the net camber responses under simultaneously applied chassis roll and wheel travel and chassis roll and steering were quite small. In addition, the amounts of change in the scrub radius and wheel track width were acceptable.

Acknowledgment

This work was supported by a grant from the National Science Council, Taiwan. (Project No.: NSC 100-2221-E-020-016)

References

[1] W. C. Mitchell, R. Simons, T. Sutherland, and K. L. Michael, "Suspension geometry: theory vs. K&c measurement," *SAE Technical Paper* 2008-01-2948, 2008.
[2] J. S. Hwang, S. R. Kim, and S. Y. Han, "Kinematic design of a double wishbone type front suspension mechanism using multi-objective optimization," *5th Australasian Congress on Applied Mechanics, Australia*, 2007.
[3] E. K. Janette, *Automotive Steering, Suspension, and Wheel Alignment.* San Jose, CA: Chek Chart, 1993, pp. 226–232.
[4] K. P. Balike, S. Rakheja, and I. Stiharu, "Synthesis of a vehicle suspension with constrained lateral space using a roll-plane kineto-dynamic model, *SAE Int. J. Mater. Manuf.*, vol. 3, pp. 305-315, 2010.
[5] P. Holdmann, and F. Berger, "Kinematics and compliance of sports utility vehicles," *SAE Technical Paper* 2001-01-0491, 2001.
[6] W. C. Mitchell, "Forced-based roll centers and an improved kinematic roll center," *SAE Technical Paper* 2006-01-3617, 2006.
[7] L. Li, C. Xia, and W. Qin, "Analysis of kinetic characteristic and structural parameter optimization of multi-link suspension," *SAE Technical Paper* 2007-01-3558, 2007.
[8] W. Lamers, "Development and analysis of a multi-link suspension for racing applications," *Master's Thesis, Eindhoven University of Technology, the Netherlands*, 2008.
[9] B. P. Minaker, and N. C. Nantais, "An eigenvector approach to roll centre analysis," *SAE Technical Paper* 2007-01-0859, 2008.
[10] Y. C. Chen, H. H. Huang, and J. B. Lin, "Application of Vector Finite Screw Analysis to Determine Roll Center from Wheel Points," *Proc. IMechE, Part C: J. Mechanical Engineering Science*, vol. 225, pp. 2586-2596, 2011.
[11] SAE, *Vehicle Dynamics Terminology*, SAE J670e, last revised 1976.
[12] W. F. Milliken, and D. L. Milliken, *Race Car Vehicle Dynamics*, SAE edition, 1995.