



Optimal Roll Center Height of Front McPherson Suspension System for a Conceptual Class A Vehicle

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Abstract

In this paper, the effects of roll center height of McPherson suspension mechanism on dynamic behaviour of a vehicle are first studied, and then the optimum location of roll center of this suspension system is determined for a conceptual Class A vehicle. ADAMS/Car software was used for the analysis of vehicle dynamic behaviour in different positions of suspension roll center. Next, optimization process has been done using ADAMS/Insight. Results show significant effects of roll center location on body roll angle, body roll rate and steering response. Also, the contradiction between body roll and steering response can be observed.

Keywords: Vehicle dynamics, Roll center height, McPherson suspension mechanism.

1. Introduction

Roll center or instantaneous center of sprung mass and ground is an important parameter of independent suspensions and its determination is the prime point of suspension mechanism design. The height of roll center compared with sprung mass CG is an important issue. To design a suspension mechanism with optimum performance, it is necessary to have complete understanding of the behaviour of suspension mechanism and study the effects of mechanism's parameters such as roll center location on dynamic behaviour of the vehicle.

In 1981, Cronin described the kinematics of McPherson suspension mechanism by presenting a comprehensive 3D velocity and acceleration analysis [1]. In 2012, Babaeian et al. identified the effective parameters of kinematic and Elasto-kinematic behaviour of McPherson suspension mechanism by using sensitivity analysis. In their study, suspension bushing and geometry are optimized using genetic algorithm [2]. In 2009, Mohammadi et al. introduced a new method called TANGA for suspension system optimization. They used Taguchi method in experiment design, artificial neural networks for modelling and genetic algorithm for optimization process [3]. In 2012 Nemeth and Gaspar designed a new suspension mechanism with variable geometry [4].

In the present study, McPherson suspension mechanism analysis and optimization of roll center height have been investigated. The vehicle under study is a conceptual Class A vehicle that is designed and developed in *Road and Railway Vehicles Institute of Isfahan University of Technology*. This car has McPherson suspension system in the front and twist beam suspension in the rear. At first, dynamic model of the vehicle was created using ADAMS/Car. Then, by changing the front suspension roll center height, the vehicle behaviour in cornering maneuver is evaluated. The effects of suspension roll center height can be observed in the lateral load transfer, body roll angle, body roll rate and steering wheel angle (steering response). Finally, by defining the desired objective function, optimum location of roll center is calculated by means of ADAMS/Insight software.

2. Roll Center Theory

The virtual location of suspension mechanism that transfers the lateral forces of the tires to sprung mass is an important parameter of suspension systems. The location of this point, known as roll center, has a direct effect on vehicle cornering behaviour. According to SAE J670E, roll center is a point in a vertical plane that passes through wheel centers, and in which a lateral force application will produce no roll angle [5]. Fig.1 shows the roll center position in left cornering manoeuvre from the rear view. In this situation, it is assumed that axle and tires are fully rigid. The purpose of this model is load transfer analysis in cornering manoeuvre and investigating the effects of roll center height and CG height of sprung and unsprung masses on load transfer.

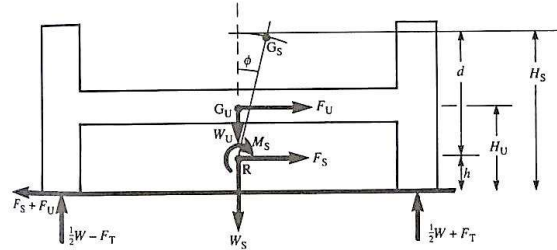


Fig. 1. Roll center and axle free body diagram in cornering [5]

As shown in Fig. 1, having a lateral acceleration a in cornering manoeuvre, lateral forces acting on sprung and unsprung masses can be expressed by Eq.1 and Eq.2 as below:

$$F_u = m_u a \quad (1)$$

$$F_s = m_s a \quad (2)$$

Also F_s and the force due to weight of m_s in place of G_s can be moved to roll center and the resultant moment M_s can be expressed by Eq.3:

$$M_s = m_s a d \cos \phi + m_s g d \sin \phi \quad (3)$$

In which:

$$d = H_s - h \quad (4)$$

Clearly, moment M_s that is faced with the resistance of springs causes the body roll while F_s in roll center does not cause any body roll. Accordingly, the total load transfer can be divided into three parts (Eq. 5):

- 1- Unsprung mass lateral force (F_{Tu})
- 2- Sprung mass lateral force in roll center (F_{TR})
- 3- Sprung mass moment (F_{TM})

$$F_T = F_{Tu} + F_{Ts} = F_{Tu} + F_{TR} + F_{TM} \quad (5)$$

Thus, for moment balance around a point on the ground and on vehicle vertical axis, total load transfer in cornering manoeuvre can be expressed as Eq. 6:

$$F_T = \frac{m_u a H_u}{w} + \frac{m_s a h}{w} + \frac{M_s}{w} \quad (6)$$

As can be seen, as the height of roll center increases, the load transfer due the m_s increases too. On the other hand, increasing the height of roll center by reducing the amount of d and moment M_s causes reduction of the load transfer. In the general case, despite having the suspension system in a vehicle, the lateral load transfer is increased in cornering manoeuvres. As a result, sport vehicles use relatively rigid suspension mechanisms for better cornering [6].

It seems for reducing the lateral load transfer in automobiles due to suspension system, the position of roll center should be chosen higher and near the CG. But by increasing h in this situation, the load transfer due to lateral force of the sprung masses increases, and because of greater effect of this term on total load transfer, usually suspension system with high roll center is not recommended for automobiles. However, in sport and racing cars, suspension mechanism is designed based on high roll center near or upper the CG [6, 7].

3. McPherson Mechanism Roll Center

In mechanistically viewpoint, the roll center of suspension system is the instantaneous center of the body (sprung mass) and ground [6]. Theoretically, instantaneous center (IC) in a mechanism is the common point of two levers in which the absolute velocity of IC on two levers is equal and levers do not have any relative velocity at the common point IC. According to Kennedy theorem, for three bodies moving relative to each other, three IC are placed on a line.

This theorem is used for determining the position of suspension roll center. Equivalent mechanism of McPherson suspension system is shown in Fig. 2. Vehicle body, wheel (unsprung mass) and ground can be considered as three independent bodies. The instantaneous center of the wheel and ground (I_{wg}) is the tire-road contact point and the instantaneous center of the wheel and the body (I_{wb}) obtained by a geometric method. Thus, the instantaneous center of the body and ground (I_{bg}) can be found by using Kennedy theorem. In fact, I_{bg} is the initial roll center of suspension mechanism or roll center in low lateral accelerations.

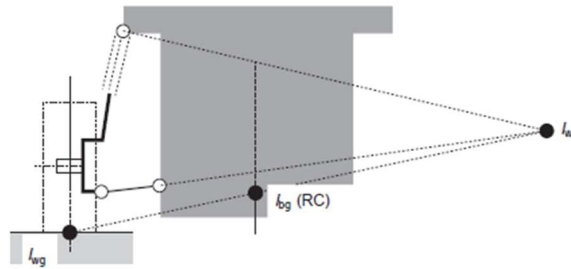


Fig. 2. Roll center determination of McPherson suspension mechanism [6]

Therefore, finding the roll center location of McPherson mechanism consists of three general steps [8]:

- 1- Finding the virtual reaction point of suspension's links.
- 2- Drawing a line from tire-road contact point to virtual reaction point.
- 3- Roll center is the intersection point of obtained line and vehicle vertical axis.

4. Dynamic Model of the Vehicle

Dynamic modelling of the conceptual car is done in ADAMS/Car software. ADAMS/Car is one of the strongest software in modelling and dynamic analysis of vehicles. ADAMS/Car can simulate various types of driving manoeuvres. Modelling process comprises three general stages:

- 1- Building parametric templates.
- 2- Building subsystems.
- 3- Subsystem assembly and development of full vehicle model.

For full vehicle modelling, all dynamic subsystems such as front suspension, rear suspension, steering, tires, power train, body, brake and anti-roll bar (ARB) are modelled. Also, rear suspension system that is semi-independent is modelled in flexible form using Viewflex package in ADAMS/Car. Front suspension and steering assembly are shown in Fig. 3. In addition, the full vehicle model is presented in Fig.4. The results of specific manoeuvres such as constant radius cornering (CRC) and step input steering of similar vehicles are used for the model validation. It should be noted that the car is still in the conceptual design stage and not prototyping.

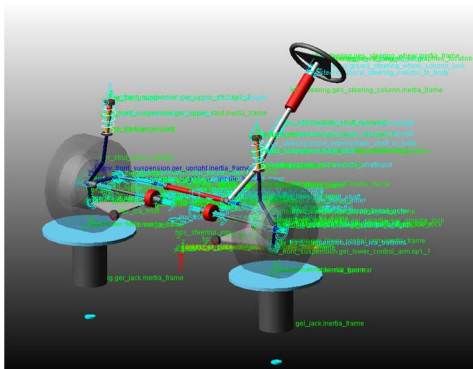


Fig. 3. Front suspension and steering system assembly

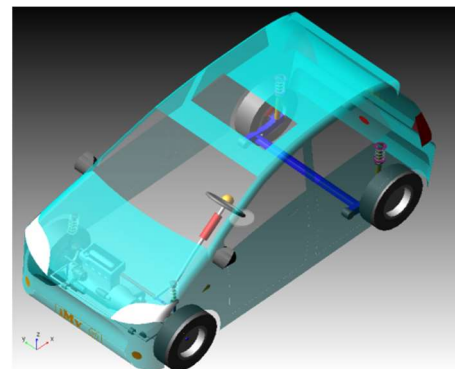


Fig. 4. Full vehicle model

In order to investigate the effects of roll center height on vehicle behaviour, three front suspension models with three different positions of roll center are developed. Roll center height relative to the ground in the default model is 80mm. To change the roll center height in the other two models, the coordinates of the joint of the suspension control arm and body are changed in vertical direction (Table 1).

Table 1. Roll center location for three models of front suspension

Roll center height relative to ground [mm]	Control arm joint coordinate in Z direction [mm]
80	181
177	217
-3	145

5. Effects of roll center height on vehicle dynamic behaviour

A constant radius cornering (CRC) simulation with 50 meters radius in left direction was performed to examine the effects of roll center position on vehicle behaviour. In this test, the speed of the vehicle was increased from 40 to 70 km/h within 8 seconds. The simulation was performed three times with three models of front suspension system. Body roll angle, body roll rate, steering wheel angle, vertical force of front tire-road contact point and tires lateral forces are the outputs of these tests. The results are shown in Fig. 5 to Fig. 11. Results indicate that increasing roll center height causes reduction of body roll angle and body roll rate, which is considered as a desirable outcome. On the other hand, Fig. 7 shows that the steering wheel angle for crossing the path is increased. This is due to increase of transfer of the vertical load from the inner wheel to outer wheel and changes in tires lateral force. Increasing inner tire lateral force and decreasing outer tire lateral force and slight reduction of total tires' lateral force have caused a greater steering wheel angle than is required in the default roll center position.

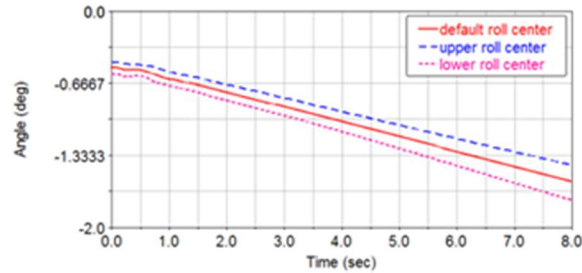


Fig. 5. Body roll angle

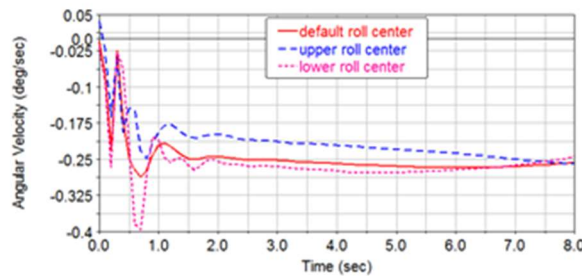


Fig. 6. Body roll rate

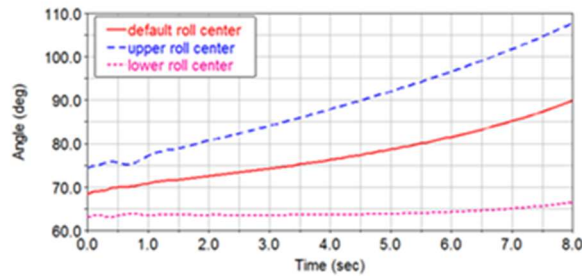


Fig. 7. Steering wheel angle

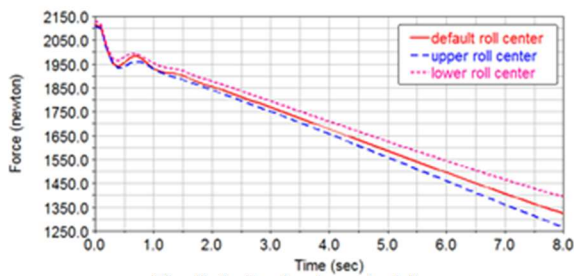


Fig. 8. Left wheel vertical force

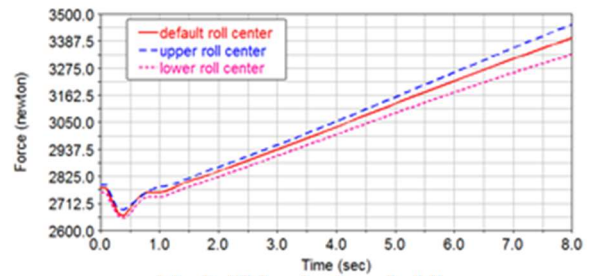


Fig. 9. Right wheel vertical force

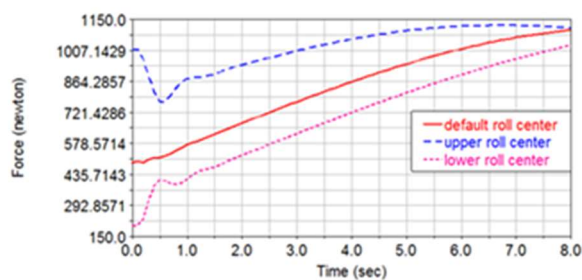


Fig. 10. Left tire lateral force

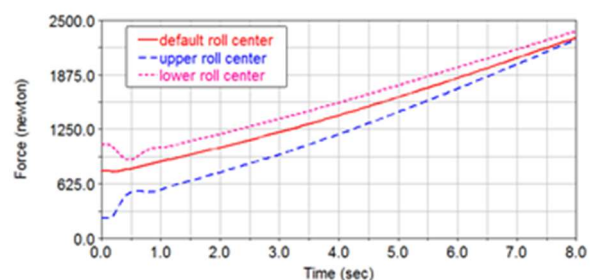


Fig. 11. Right tire lateral force

There are two important points:

The first point is about the graph of the steering wheel angle versus lateral acceleration. The slope of the steering wheel angle versus lateral acceleration somehow evokes the Understeer gradient, because lateral acceleration is increased during the time. So increasing the roll center height causes Understeer and decreasing the roll center height causes Neutralsteer and even Oversteer.

The second point concerns the change of lateral forces with the change of the vertical forces. It is expected that the lateral force increases with the increase of vertical force of the tire. This issue is not observed when the location of the roll center is changed. It should be noted that changes also occur in other parameters such as wheel camber and wheel side-slip angle. By decreasing the inner wheel vertical force, wheel camber will bend toward the positive angle and by increasing the outer wheel vertical force, wheel camber will bend toward the negative angle. Fig. 12 and Fig. 13 show the camber changes of the front wheels in CRC simulation. It is clear that camber changes is affecting the lateral force.

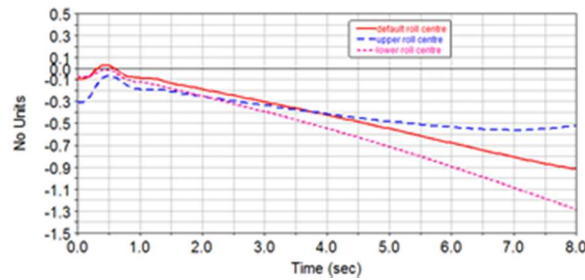


Fig. 12. Left wheel dynamic camber angle

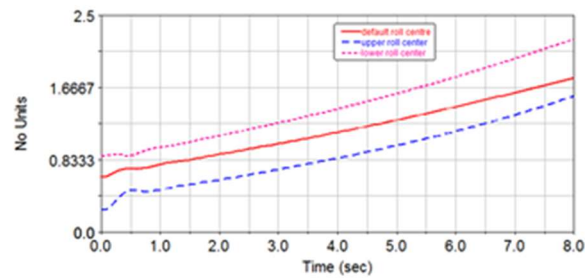


Fig. 13. Right wheel dynamic camber angle

6. Optimization Process

According to the above discussion, it is necessary to provide an optimum design for the suspension roll center location. The optimization process must satisfy the following objectives:

- 1- Minimizing steering wheel angle in the manoeuvre with Understeer condition.
- 2- Minimizing lateral load transfer in the manoeuvre.
- 3- Desirable body roll angle and body roll rate.

In order to perform this optimization, ADAMS/Insight coupled with ADAMS/Car are used.

Full factorial experiment with 64 runs is designed for the depicted variables of Table 4 (3 variables in 4 levels) and is applied to above-mentioned CRC simulation. A cubic polynomials form response surface is created by regression analysis on experimental results for outputs. Analysis of variance (ANOVA) is used to examine the goodness of fit (Table 2). R^2 and R^2_{adj} values indicate the goodness of fit [9]. When these two values approach one, it indicates a minimum error in polynomial fitting. Also, big ratio of range to variance (R/V) indicates the interpolationability of the regression model.

At the end, optimization is performed using SDI¹ algorithm. SDI algorithm is a kind of direct search methods. Table 3 shows the controlling parameters of the SDI algorithm in the present problem. The optimization constraint is that the body roll angle and the body roll rate do not increase higher than 10% of initial values (values before optimization).

Table 4 shows the optimization results. Fig. 14 to Fig. 16 show the vehicle behaviour before and after optimization process, too. Design variables in Table 4 are:

- Z_1 : Z coordinate of the joint between suspension control arm and body.
- Y_2 : Y coordinate of the suspension strut lower mount.
- Z_2 : Z coordinate of the suspension strut lower mount.

Results indicate 11% improvement in steering wheel angle in CRC manoeuvre. Also, improvement in lateral load transfer can be observed.

¹Stochastic Design Improvement

Table 2. Goodness of fit

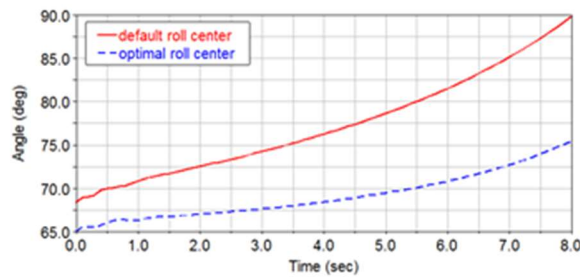
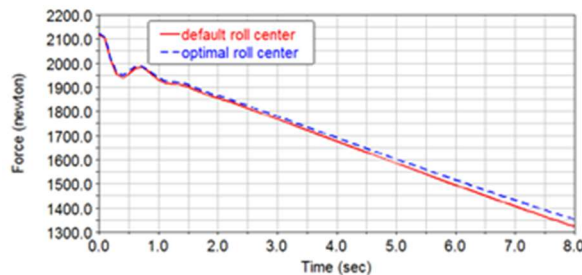
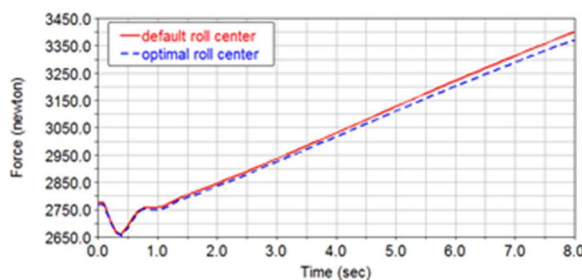
Outputs	R^2	R^2_{adj}	R/V
Steering wheel angle	1	1	4.41 e+3
Body roll angle	1	0.99	2.67 e+4
Body roll rate	1	0.99	1.08 e+3
Left wheel vertical force	1	1	1.96e+3
Right wheel vertical force	1	1	2.67e+3

Table 3. SDI algorithm parameters

Converge tolerance	Number of runs in each step	Step ratio
0.01	27	0.3

Table 4. Design variables and optimization results, [mm]

Design variable	Initial value	Level 1 (lower limit)	Level 2	Level 3	Level 4 (upper limit)	Optimum value
Z1	180.978	144.778	168.911	168.911	217.178	161.68
Y2	569.671	559.671	566.341	573.011	579.671	579.24
Z2	591.105	585.194	589.134	593.074	597.016	588.46

**Fig. 14.** Steering wheel angle improvement**Fig. 15.** Left wheel vertical force improvement**Fig. 16.** Right wheel vertical force improvement

7. Conclusion

Optimization results in Table 4 show that the optimal roll center height is about 40 mm from the ground. This means that the roll center height is reduced about 50%. Also, a slight improvement in lateral load transfer can be seen. But, these improvements are along with a slight increase in body roll angle and body roll rate (less than 10%) which can be modified by changing anti-roll bar (ARB) stiffness.

Another important point is the slight improvement of lateral load transfer. In this case, it should be noted that the lateral load transfer is affected by various parameters according to Eq. 6. Therefore, by reducing the roll center height, the body roll induced lateral load transfer is increased. In addition, the contribution of sprung mass lateral acceleration on total load transfer will be decreased. Moreover, because of the effects as a result of the changes of the suspension mechanism such as camber change and etc., steering wheel angle and handling behaviour of the vehicle are improved in CRC simulation.

Nomenclature

a	Lateral acceleration [m/s^2]	M	Translational Moment [N.m]
F	Lateral force [N]	R	Symbol of roll center location
F_T	Total lateral load transfer [N]	s	Symbol of sprung mass
G	Symbol of CG location	u	Symbol of unsprung mass
H	Height of CG [m]	W	Weight [N]
h	Height of roll center [m]	w	Wheelbase [m]
I	Symbol of instantaneous center	φ	Roll angle [deg]
m	Mass [kg]		

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