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Effect of control arm position parameters on vehicles used for competition

Fabian Fuentes Martinez[1], Albert Miyer Suarez Castrillon[2] and Sir-Alexci Suarez Castrillon[3]

[1,2] *Faculty of Engineering and Architecture. University of Pamplona, Colombia.*
[3] *Engineering Faculty, University Francisco of Paula Santander Ocaña, Colombia.*

Abstract

The simulation and analysis of the design is intended to provide a solution to the drawbacks of the rigid axle suspension used by competition cars on oval tracks, with a new suspension system that offers better configuration characteristics, alignment versatility, performance, weight and space, meeting the requirements of the car. The design of these systems varies according to the characteristics of the car, although it is based on the same theories that allow determining the geometry parameters according to the type of suspension, there are no unique or standard suspension geometries, that is why the parameters of the position of the control arms used in this type of competitions are analyzed.

Keywords: Control arms, suspension, competition car.

1. INTRODUCTION

When talking about the geometry, it refers to how the unsprung mass of the car is connected to the sprung mass, these connections not only describe the relative motion trajectory, but also control the forces that are transmitted between them [1]. For the development of the analysis of the suspension geometry, it is necessary to analyze the trajectories described by the control arms, for this purpose the length of the control arms both upper and lower together with the length of the stub axle are assumed, since they are the parameters selected a priori for such analysis [2], this is because there is no methodology that allows defining these parameters exactly, so it is necessary to resort to assumptions or seek the solution using a graphical method.

Simulation is one of the possibilities to configure the geometry, but it can also be performed by means of scale vehicles, which offer a quick configuration of tests and adjustments, while the data can be taken to a real scale, giving the possibility to make adjustments such as changing the shock absorbers [3], [4]. Each configuration varies according to the make, function and terrain of the car to be analyzed, that is why the investigations become specific to each car [5], [6]. The studies are focused on cars used for rallying and racing on ovals or open tracks, but it is increasing its study in individual cars that can reduce traffic, such as narrow cars, which have a drawback in the tilting suspensions, trying to check the mathematical model through simulation and control systems [7]. For three-wheeled vehicles or tricycles the load that supports when turning is the maximum, if it is possible to vary an angle before the turn could decrease the centrifugal force and keep in good condition the tires and stability, it is therefore necessary to use a tilting suspension system [8]-[10].

In this research the vehicle suspension is analyzed by means of its geometry, applied to vehicles that use the same chassis in Nascar series races, making a 3D design using Catia to study the kinematics, in order to calculate the parameters that define the position of the control arms of the system and the effects it has on the car.

2 METHODOLOGY

The location of the instantaneous center (IC) in the front view allows to control the following parameters: the roll center height (RCH), the camber angle variation, and the wheel contact point (*Scrub*). The IC can be located on the inside or outside of the wheel, it can be located above or below the level of the running surface of the track, such position is at the discretion of the designer or according to the desired characterization for the design of the suspension system.

2.1 Construction of instantaneous centers.

The instantaneous center is used to describe and determine some common parameters in suspension. The word "instantaneous" is understood as the particular position of connection or articulation of both lower and upper control arms, and "center" refers to an imaginary projected point that is effectively the pivot point of the control arms at the instant.

The instantaneous centers in Figure 1 are obtained by drawing a line between the upright connection and through the centerline of the control arms, projecting it across the plane for both the lower and upper control arm (white colored lines), intersecting at a point (yellow dots), this intersection is the instantaneous center of the control arms. If this projection is made in the front or rear view, the instantaneous center defines the *camber* angle variation and roll center information.

2.2 Roll center. The roll center establishes the coupling between the suspended and unsprung mass. When the car takes a curve, the centrifugal force acting at the center of gravity has

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a reaction at the wheels. The lateral force at the center of gravity can be transferred to the roll center. The lower the roll center, the smaller the lateral acceleration moment that must be resisted by the springs.

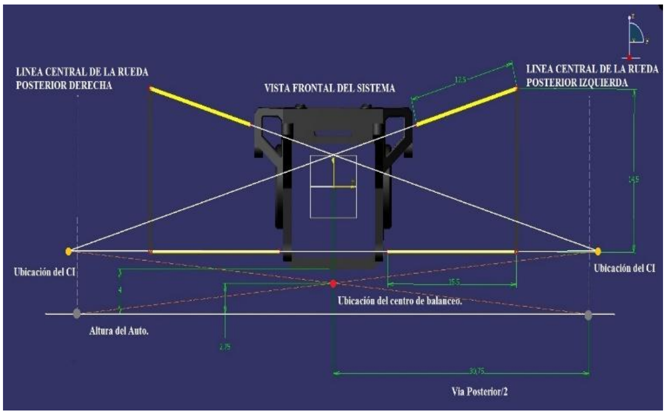

Figure 1. Location of instantaneous centers, front view.

However, another factor in setting the desired roll center height, if the roll center is above the level of the track running surface, the lateral force on the wheel will generate a moment at the instantaneous center, this moment presses the wheels down and lifts the suspended mass, an effect which is called *jacking*, Figure 2. If the roll center is below ground level (possible with SLA suspensions), then the force will press on the suspended mass causing it to descend. In either case, the suspended mass will have a vertical deflection due to the lateral force.

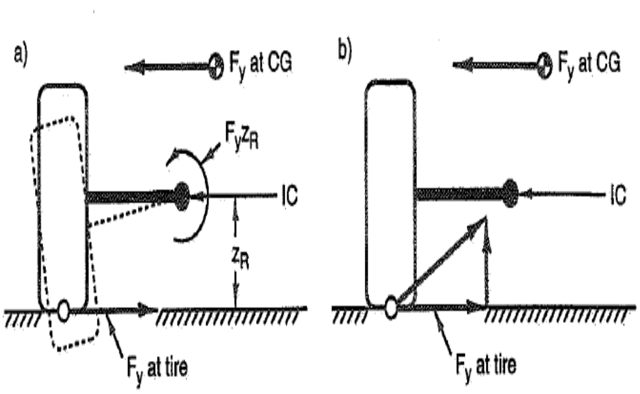

Figure 2. *Jacking* effect**. Source:** [11]

2.3 Camber angle variation. While the roll center is a function of the height of the car and length of the fvsa, length referring to the distance measured from the instantaneous center to the centerline of the wheel, the variation of the camber angle is only a function of the length of the fvsa, as seen in Figure 3.

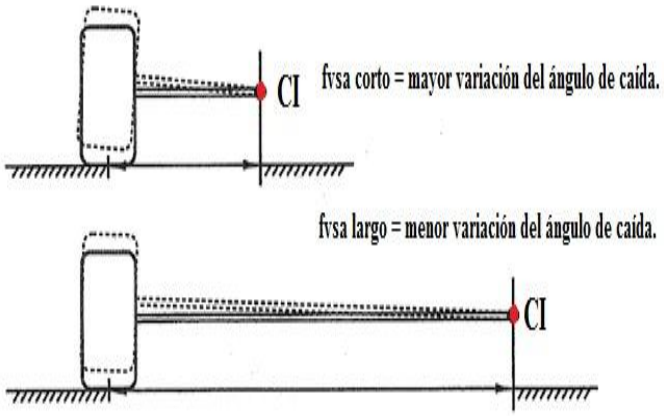

Figure 3. Ways of varying the angle of repose**. Source:** [11]

If you have an instantaneous center located at a greater distance from the centerline of the wheel you obtain a small variation of the camber angle and as the instantaneous center is located at a smaller distance from the centerline of the wheel the variation of the camber angle will be greater, this variation is achieved per inch or mm of suspension travel or if the length of the fvsa is known, equation 1 is used.

$$\text{Ángulo de caída} = \tan^{-1}\left(\frac{1}{\text{fvsa}}\right), \ \text{Grados }(°)/_{\text{in}} \qquad (1)$$

For an independent suspension, whether front or rear, the control arm assembly is intended to control the movement of the wheel relative to the car chassis in a single prescribed path. That path could have camber angle increase, caster angle (front suspensions) and toe change, but its path remains the same as it moves up and down. The alignment parameter concerning toe-in and toe-out will not be taken into account in this chapter, because it has more influence on front suspensions and will be discussed later in terms of settings.

2.4 *Scrub:* Another variable in the front view is the point of contact of the wheel with the running surface of the track (*scrub*). This is the lateral movement relative to the ground that results from the vertical movement of the wheels. *Scrub* occurs in all suspension systems. The *scrub* value is a function of the absolute and relative lengths of the control arms and the positions of the instantaneous center of the front sight relative to the ground. When the instantaneous center of the front sight is at any position other than ground level this parameter is increased and affects irregular wheel wear.

Figure 4 shows other cases of the *scrub* parameter depending on the location of the instantaneous center, if the IC is below ground level and inward, the wheel will move inward during travel.

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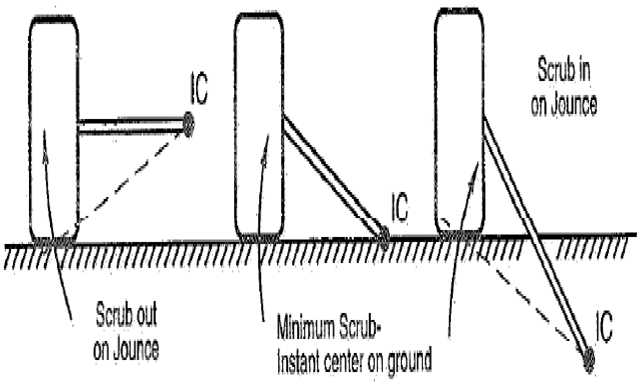

Figure 4. The *scrub* parameter as a function of instantaneous center height**. Source:** [11]

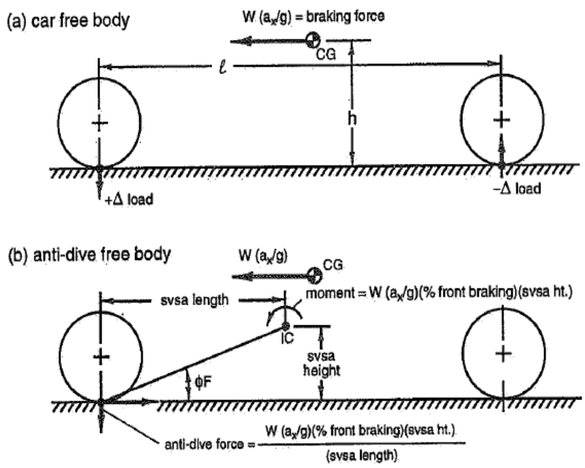

Figure 5. Anti-features**. Source:** [11]

2.5 SUSPENSION GEOMETRY IN SIDE VIEW

The swing arm in the side view (svsa) (distance from the IC location to the wheel centerline) controls the movements and forces resulting from accelerations and decelerations. Typical suspension parameters are anti-dive, anti-lift, anti-squat and wheel path. The position of the svsa can be in front of (or behind) and above (or below) the wheel center, these are the possible solutions for an independent front and rear suspension. Typically, the instantaneous center is behind and above the wheel center in front suspension, and it is in front and above in most rear suspensions. For the case study, due to the parallelism of the control arms, there is no definite instantaneous center since it tends to infinity.

Anti-features. The "Anti or Contra" effect in suspensions is a term that actually describes the longitudinal and vertical coupling forces between the suspended mass and the unsprung mass. The result is only the angle or inclination of the swing arm in the side view. The *Antidive/Anti-Squat* characteristic in the suspension system under study does not change the steady-state load transfer. The total longitudinal load transfer at constant acceleration or braking is a function of the wheelbase (ℓ), center of gravity (CG) height, and braking force (car weight) x (a_x/g), as shown in the free body diagram of the car in Figure 5.

Figure 6 shows a free body diagram for a car with external brakes describing the geometry of the opposing *anti-dive* and *anti-lift* forces The percentage (%) of brake distribution (or brake balance) determines the actual force as a fraction of the total longitudinal force.

For the suspension under study, it is evaluated having a neutral *anti-squat* parameter, i.e. 0%, which is produced when Θ_R (angle of inclination of the IC is equal to zero. If the suspension has 0% anti-squat, all the load transfer is supported by the spring-damper assembly causing the suspension to deflect proportionally to the wheel speed when the car accelerates or applies the brakes. Using equation 2 the value of the anti-characteristic is obtained.

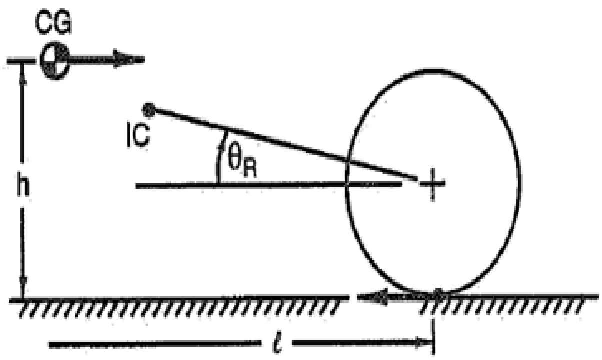

Figure 6. Anti-characteristics of a rear-wheel drive SLA suspension. **Source:** [11]

$$\% \text{ Anti} - \text{Squat} = \frac{\Theta_R}{\left(\frac{h}{\ell}\right)} \text{x } 100\,\% = 0 \tag{2}$$

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$$\theta_R = 0°$$

$$\% \text{ Anti} - \text{Squat} = \frac{\tan(0°)}{\left(\frac{14\ in}{110\ in}\right)} \cdot 100\ \% = 0 \qquad (3)$$

Now the suspension is evaluated with 100% anti-sway characteristics, all longitudinal load transfer is supported by the control arms and not by the suspension spring-damper assembly, so there is no deflection in the suspension during braking or acceleration.

$$\%\ Anti - Squat = 100\%$$

$$\Theta_R = tan^{-1}\left[\frac{14\ in}{110\ in} \cdot \frac{100\ \%}{100\ \%}\right] = 7.25\ ° \qquad (4)$$

This parameter is available when required for the location of both lower and upper control arms in order to reduce the travel advance if any during accelerations and decelerations as they have great importance the forces acting on the center of gravity of the car, which tend to cause a crushing action (*anti-dive/anti-squad*) in the rear section during acceleration and lift during decelerations of the suspension; the correction is usually done by modifying the inclination of the suspension control arms.

In the real world, the mechanical components that supply the constraints are not "perfect" in the sense of constraining the motion for a given degree of freedom. Therefore, for the independent suspension geometry calculated above, it is important to select the connecting elements to determine the degree of freedom describing their motion and the constraints that are attached to the system. Figure 7 shows the joints at the ends and by force elements.

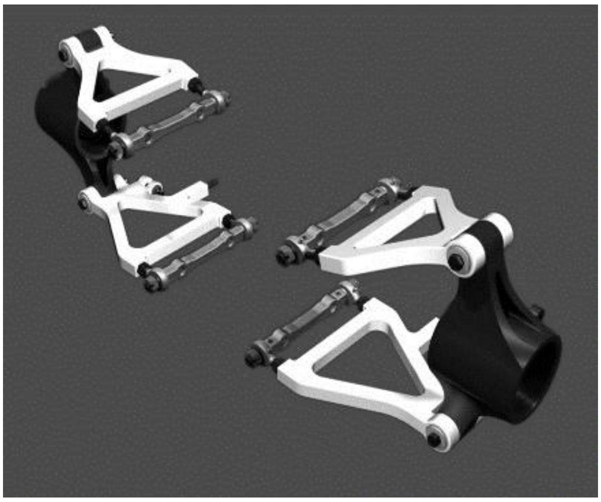

Figure 7. Degrees of restriction given by connection elements.

3. RESULTS

Tables 5 and 6 below detail the final results for the geometry to be modeled with the studied parameters *scrub*, roll center height and roll angle variation.

Table 5. Olley's constants

Calculation of Olley's constants			
P1	**Q1**	**P2**	**Q2**
0.015483871	-0.013548	0.86180645	2.933419
U1	**V1**	**U2**	**V2**
0.137419355	-0.1202419	7.648532	26.0341

Table 6. Calculated parameters for suspension geometry

Output data					
Wheel travel (in)	**Camber**	**Camber variation**	**Scrub (in)**	**Scrub Variati on**	**Height of balancing center**
z	**γ (rad)**	**dγ/dz (rad)**	**|γ3|**	**dγ3/dz**	**RCH (in)**
-2.0	0.000	0.001	0.000	-0.199	6.133
-1.9	0.000	0.001	0.020	-0.194	5.961
-1.8	0.000	0.001	0.039	-0.188	5.789
-1.7	0.000	0.001	0.057	-0.183	5.618
1.6	0.000	0.001	0.075	-0.177	5.447
-1.5	0.001	0.001	0.093	-0.172	5.276
-1.4	0.001	0.001	0.110	-0.166	5.106
-1.3	0.001	0.002	0.126	-0.161	4.936
-1.2	0.001	0.002	0.142	-0.155	4.767
-1.1	0.001	0.002	0.157	-0.150	4.597
-1.0	0.001	0.002	0.172	-0.144	4.429
-0.9	0.002	0.002	0.186	-0.139	4.260
-0.8	0.002	0.002	0.200	-0.133	4.092
-0.7	0.002	0.002	0.213	-0.128	3.924

-0.6	0.002	0.002	0.226	-0.122	3.756
-0.5	0.002	0.002	0.238	-0.117	3.588
-0.4	0.003	0.003	0.249	-0.111	3.421
-0.3	0.003	0.003	0.260	-0.106	3.255
-0.2	0.003	0.003	0.271	-0.100	3.088
-0.1	0.004	0.003	0.281	-0.095	2.922
0.0	0.004	0.003	0.290	-0.090	2.756
0.1	0.004	0.003	0.299	-0.084	2.591
0.2	0.004	0.003	0.307	-0.079	2.425
0.3	0.005	0.003	0.315	-0.074	2.261
0.4	0.005	0.003	0.322	-0.068	2.096
0.5	0.005	0.004	0.329	-0.063	1.932
0.6	0.006	0.004	0.335	-0.057	1.768
0.7	0.006	0.004	0.341	-0.052	1.604
0.8	0.007	0.004	0.347	-0.047	1.440
0.9	0.007	0.004	0.351	-0.042	1.277
1.0	0.007	0.004	0.355	-0.036	1.114
1.1	0.008	0.004	0.359	-0.035	1.070
1.2	0.008	0.004	0.362	-0.030	0.914
1.3	0.009	0.004	0.365	-0.020	0.628
1.4	0.009	0.004	0.367	-0.015	0.466
1.5	0.009	0.005	0.369	-0.010	0.305
1.6	0.010	0.005	0.370	-0.005	0.143
1.7	0.010	0.005	0.371	0.001	-0.017
1.8	0.011	0.005	0.371	0.006	-0.178
1.9	0.011	0.005	0.371	0.011	-0.338
2.0	0.012	0.005	0.370	0.016	-0.498

The design parameter has a margin of error of 0.2% being equal to the Figure of the roll center location and the same centerline height.

4. CONCLUSION

A softer damper setting will increase cornering grip (improve grip). If the suspension is stiffened, it causes understeer, making it more difficult for the car to make tight turns on fast circuits. The harder this setting is, the firmer and stiffer the car will be, although it will also be more unstable. Use a stiffer setting to reduce understeer by making the car take the track line at higher speed. As for the anti-roll bar, the stiffness should be increased to reduce the roll movement of the chassis. But too much stiffness can cause the inside wheels to separate from the racing surface during cornering (circuits with a number of right and left turns).

REFERENCES

[1] S. Biswal, A. Prasanth, R. Udayakumar, S. Deva, and R. Singh, "Design of a suspension system and determining suspension parameters of a medium downforce small Formula type car," *MATEC Web of Conferences*, vol. 124, p. 07006, Jan. 2017, doi: 10.1051/matecconf/201712407006.

[2] N. Roner, "Optimum Suspension Geometry for a Formula SAE Car," 2018, doi: 10.15760/HONORS.542.

[3] M. A. Fuentes, A. Navarro, and A. Vázquez, "Data acquisition system for 1:10 scale radio control vehicle implemented in a programmable logic device," Jan. 2015, doi: 10.18687/LACCEI2015.1.1.061.

[4] M. A. Fuentes, A. Navarro, and A. Vazquez, "Analysis of the left front suspension of 1:10 scale radio controlled AWD all-terrain vehicle," *undefined*, 2013, Accessed: Apr. 25, 2021. [Online]. Available: /paper/An%C3%A1lisis-de-la-suspensi%C3%B3n-frontal-izquierda-de-todo-Fuentes-Navarro/2a7abf9b4194ac694bf856bf3257655f8e603329.

[5] P. Sandoval and C. Andrés, "Analysis and implementation of the intelligent suspension system in the Corsa Sport 1. 4 vehicle," 2018, Accessed: Apr. 25, 2021. [Online]. Available: http://repositorio.utn.edu.ec/handle/123456789/8356.

[6] M. Levesley, S. Kember, D. Barton, P. Brooks, and O. Querin, "Dynamic Simulation of Vehicle Suspension Systems for Durability Analysis," *Materials Science Forum - MATER SCI FORUM*, vol. 440-441, pp. 103-110, Nov. 2003, doi: 10.4028/www.scientific.net/MSF.440-441.103.

[7] F. Maldonado-Páez, E. A. Cedeño, G. Guerrón-López, and J. Rocha-Hoyos, "Characterization of tilting suspension design for personal mobility vehicles," *Informacion Tecnologica*, vol. 31, pp. 87-102, Jun. 2020, doi: 10.4067/S0718-07642020000300087.

[8] B. Haridas, A. Jayaprakash, B. Benny, J. Kuriakose, and J. V. George, "Design and Fabrication of Tilting Trike," 2019, Accessed: Apr. 25, 2021. [Online]. Available: /paper/Design-and-analysis-of-an-integrated-suspension-for-Tang-He/a553bd4def09fffcf2a9659b7fd00033fd5ea55c.

[9] C. Tang, L. He, and A. Khajepour, "Design and analysis of an integrated suspension tilting mechanism for narrow urban vehicles," 2018, doi: 10.1016/J.MECHMACHTHEORY.2017.09.025.

[10] C. Tang, A. Goodarzi, and A. Khajepour, "A novel integrated suspension tilting system for narrow urban vehicles," *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, vol. 232, no. 14, pp. 1970-1981, Dec. 2018, doi: 10.1177/0954407017738274.

[11] W. F. Milliken and D. L. Milliken, *Race car vehicle dynamics*. Warrendale, PA, U.S.A: SAE International, 1995.