DrRacing's Blog

A race engineer and his race cars
Posted by: **drracing** | November 2, 2018

Load transfer and anti effects

Hi everybody,

as promised, i am here again, this time finally with something a bit more nerdy than last time.
I normally don't write too much about theoretical topics explained in vehicle dynamics books, mainly because there would be nothing original in such an article and its content would not be directly connected to something i dealt with in one of my projects.
Anyway, as i mentioned in one of my previous articles, since January I am supporting a bit the Racing Line (http://theracingline.net/), a very nice website dealing mainly with sportscars. Being part of the "staff", i am probably not the best to talk about what this website offers, because my opinion is surely not totally objective. Anyway, what can be honestly said is that we try to skip traditional news reporting (we could never keep up with professionals like the guys behind Dailysportscar or 365Sportscar, just to name two of many very good websites) and we do our best to offer original contents to the community, in my case focusing on the technical side. Our hope is that, trying to share our perspectives and opinions, we could initiate or stimulate a discussion where every fan can hopefully take place and learn something new.
In accordance with this philosophy, in my contents (mainly racecars engineering basics and race analysis) I try to offer my own perspective (which can well be wrong!) and to keep everything as simple as possible, with the aim to allow a broader audience to enjoy the articles without falling asleep.
This differentiates a bit what i write for theRacingLine with what i publish here, as the topics i deal with for tRL are not necessarily connected to any of my projects.

Keeping explanations of complex topics simple while trying to make them interesting also for people with broader knowledge and experience, is not easy at all, but it is an extremely motivating exercise: to explain something in an easy way, you have to have sure you have understood it fully. This means, you are forced to take some time and review also topics you take somehow for granted and also in areas you thought you knew all what there is to know. Actually (and luckily) there is always something new we can learn!

The physics of a racecar is actually pretty simple and there are relatively few areas that are not fully known and understood . Normally, these areas are fields where testing still doesn't allow to investigate in details or fully objectively which parameters influence a certain phenomenon and how.
Nonetheless, experience has proved to me that I am dumb enough to often learn something new, anytime i tackle a certain topic one more time.

So many words to say, that trying to explain racecars topics in an easy way is an extremely interesting challenge and i found myself really liking it.
It is also a good justification to read more and buy more books. A full library makes always a good impression to somebody visiting your place!

One of the series i prepared for the Racing Line focuses on racecar handling, as i mentioned in the past. As probably all the people reading this article know, load transfer is a key player in a car behavior. With respect to this, extremely important is the role played by suspension geometry, with particular focus on what are normally identified as anti effects.
Now, this is indeed an easy topic where i (but i suspect also other people) got confused very easily when i first came into contact with vehicle dynamics. The point is that, sometimes, even the best books don't go into details in explaining exactly not only how to calculate anti effects or load transfer but, more important, because certain assumptions are supposed to work. Most of the times, this happens either because this would require a lot of pages or because the writers assume that the reader already knows the basics on which their approach is based. Fair enough.

This article will be about anti effects, with particular focus on the lateral dynamic of a race car.

I started thinking about this after reading and getting in contact with some very smart guys writing in the FSAE (Formula SAE) forum, some years ago. Although I afterward analyzed some of the things that were discussed in the forum (with regard to anti effects for example) myself, I want to give full credit to the many smart people writing there, because reading their comments has opened a completely new perspective to me, when considering the effects of suspension kinematics on load transfer and suspension motion in braking, accelerating and cornering phases.
In particular, it has been very enlightening for me to read what Erik Zapletal ("Z" in the forum), Tim Wright and Henning Olsonn wrote. Look for them on the internet, they are really smart cookies.
In general, that Forum is extremely interesting and you can find some good reading material there, because most of the times they deal with more generic topics than something merely related to Formula Student / SAE competitions.

Beside the forum, as far as i know other people (much smarter than me) approach these areas of vehicle dynamics slightly differently than the "classical approach" presented in some books, see for example Mark Ortiz, regular contributor of Racecar Engineering and Chassis Consultant and William Mitchell, who sold for years one of the easiest and most effective suspension kinematic tools ever and sadly passed away some years ago.
Also the author Damian Harty shortly mentions in his book "The Multibody Systems Approach to Vehicle Dynamics" how he prefers dealing with anti effects in a similar way to the one i will discuss here.

Again, a lots of words to say that what i will write here is nothing new or revolutionary, simply tries to tackle this topic in an (hopefully) easier way.

The most popular books i know (beside Harty's one) normally deal with anti effects using a different approach for longitudinal and lateral direction.
For the longitudinal direction, normally they define percentage parameters, as antisquat/antidive (an effect working against suspension jounce/compression motion respectively at the front and rear axle) and antilift (an effect working against suspension rebound/extension motion respectively more often considered at the rear axle).
For the lateral direction, the most popular approach consists in considering the roll center height, at least in static condition.

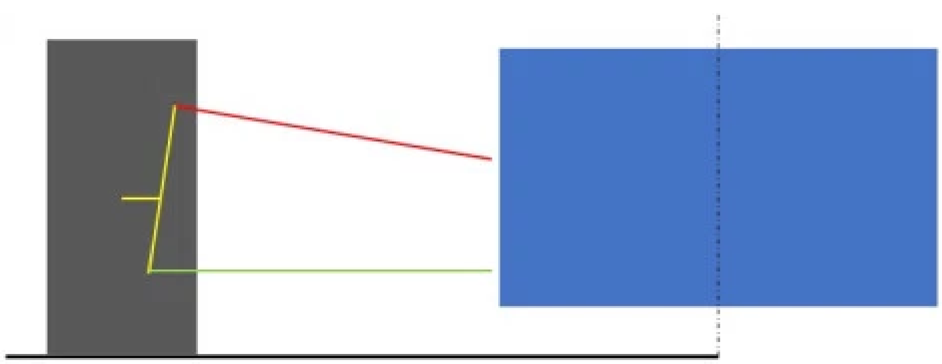
I will not go into details about how to calculate these parameters, because this has been done comprehensively in said books. I am sure most of the people crazy enough to read here know these concepts very well anyway.

The point about the way anti effects are conventionally treated is that it is sometimes hard to really visualize what these numbers means. There are easier ways to analyze, at least at the beginning, the importance and the effect of suspension geometry on roll and pitch motion and how its kinematics influence load transfer.
Talking about lateral load transfer and roll centers, in particular, they completely lose their significance in specific situations, like when the car has only two wheels on the road or when the "migration" (the movement of this specific point) is particularly big.

The approach i will describe shortly is nothing else than the application of very basic mechanics principles.
When i wrote the articles about lateral handling for the Racing Line, i tried to expand what can be found in the FSAE forum by calculating all the most important parameters without using the roll centers.
The results i obtained (numerically) using the two approaches are exactly the same, given the same assumptions and boundary conditions.
Anyway, this "more generic" approach, works also when the roll centers migrates by many hundred meters or when one wheel of an axle is not on the ground.
By the way, i inserted the calculation of the critical parameters to deploy this approach also in my excel suspension kinematics tool.

I want to stress once again, that i don't take any merit for this. What i talk about here is no new idea or concept and i simply tried my best to understand and formalize (mainly for my own use and interest) what other people already use since years and, incidentally, has its foundations on very simple mechanics principles, without using any abstract concepts.
The point is that many excellent books don't explain fully what really means to change suspension geometry in terms of load transfer and load transfer distribution effects, although this would make the description of this phenomena much easier to understand. For me, at least, this made everything much clearer.

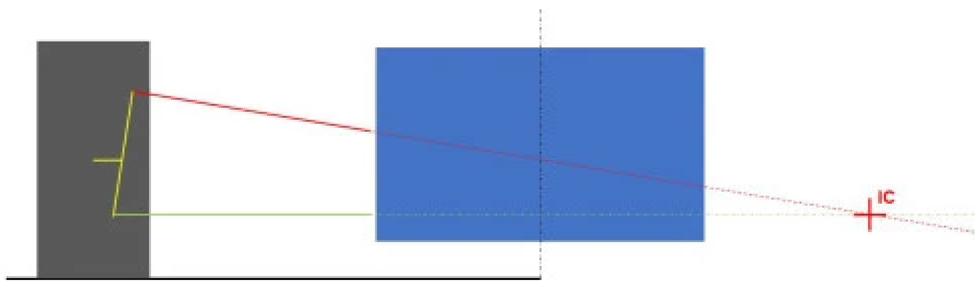
Let's start from the very beginning, using a simplified approach to make things easier to explain and to analyze.
A racecar suspension can be simplified in two dimensions (which means, projecting every element on a plane passing through the two tyres contact patches and seating vertically with respect to the ground) in what is called a "**four bars linkage**". The four links in questions are, from left to right in the following picture, the wheel/upright assembly, in contact with the ground at the contact patch, the upper control arm, the lower control arm and car's chassis.



Actually, most often a racecar suspension cannot be considered a 2D mechanism, but this simplification is very useful to keep the explanation simple.

We normally define **wheel travel** the distance covered vertically (or in Z direction, in the coordinate system we introduced previously) by the wheel (or by any point of it, although common practice is to consider the center of tyres contact patch or the wheel center).

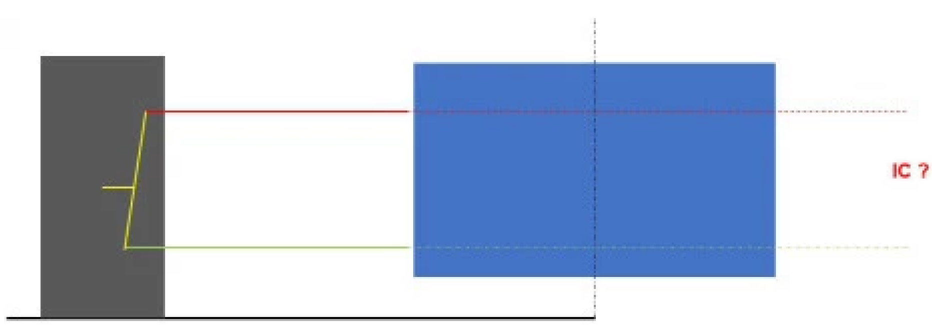
In a 2D four bars linkage, where we consider car's chassis to be fixed and the wheel/upright assembly to be free to move with respect to it, the wheel assembly will move on a path that, in each instant, will be circular with its center at the intersection point of the lines obtained elongating the control arms segments.



This means, the center of this circular path can be considered a pin point in the movement of the wheel (contact patch) with respect to the body. Its position will change, depending on the wheel travel, but in each instant we can represent our suspension using a bar connecting the contact patch (or each point of interest belonging to the wheel/upright assembly) and this point.

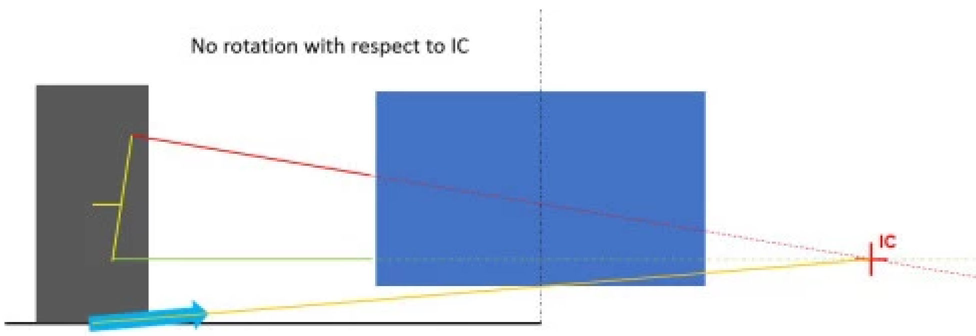
This point is normally called **instant center** (IC in our previous picture) and, as we will see, it is an extremely important concept.

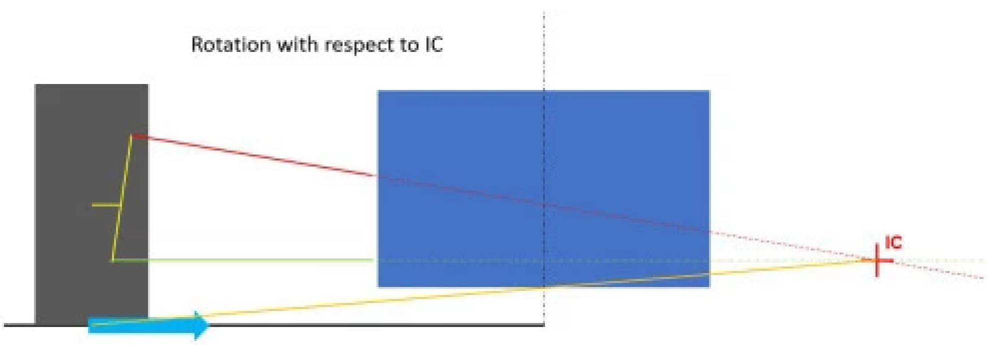
At this point, somebody has surely already recognized how the instant center would be undefined when the upper and lower control arms are (in a certain suspension position) perfectly parallel to each other.



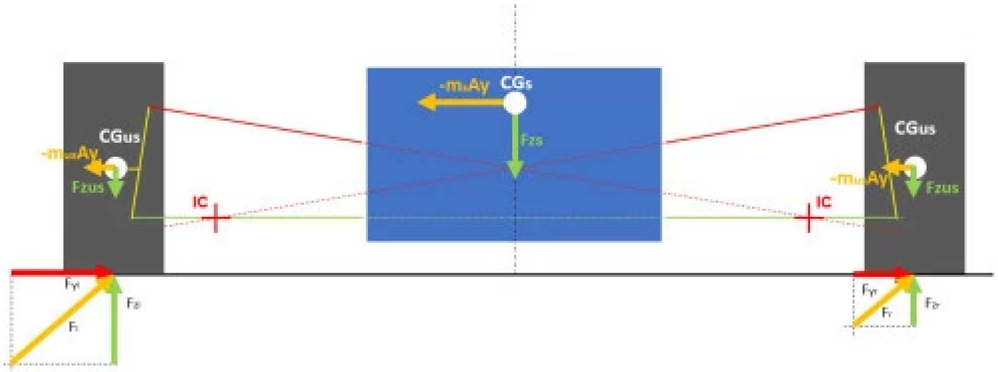
The instant centre position has not only an effect on suspension kinematics (its position determines the arc on which the wheel moves and this influences how parameters like camber and track width change), but it is also important because the line connecting the contact patch to the instant centre defines the only direction that a force applied to the wheel (tyre) at the contact patch can act on, without causing a rotation of the wheel itself around the instant centre (roll).

The following picture help to understand this point, which is crucial to figure out how suspension kinematics affects lateral load transfer. If we have a bar (which we suppose to be rigid) with a free end and another end located in a pin point, we can apply a force to the bar avoiding its rotation only if said force has the same direction of the bar.

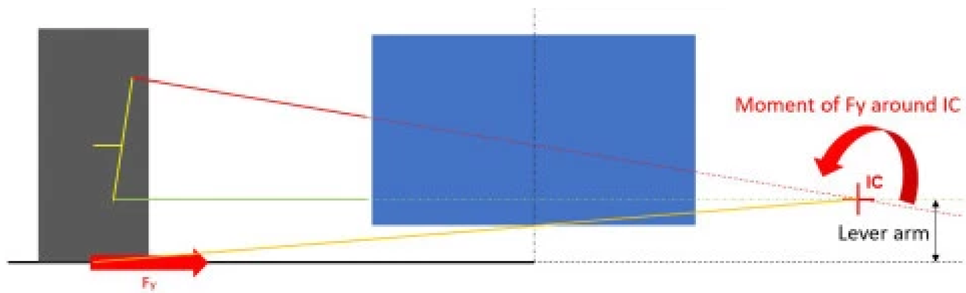




Generally, the force applied by the road to a tyre in a pure cornering situation acts in a different direction than the one of the above-mentioned line. Our 2D suspension will normally experience a resultant force at the contact patch that can be decomposed in two components: a lateral one, which we named already in several occasions $\mathbf{F_y}$ and a vertical one, $\mathbf{F_z}$.



Let's focus for a moment only on the lateral component, $\mathbf{F_y}$. We immediately recognize how the lateral components of the forces acting at the contact patches generate a moment about the relative IC and, hence, an effect that tends to rotate our "idealized swing arms".

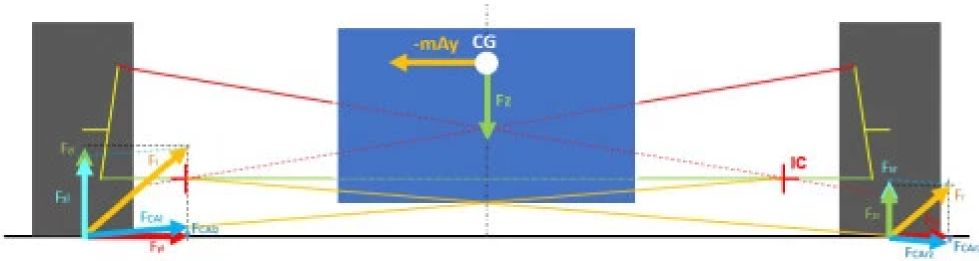


What the reader should keep in mind is that, if the IC seats as shown in the picture, these moments will move the suspension in the opposite direction than the one in which the body would roll, under the action of the centrifugal force (acting at the CG). De facto, as long as the IC seats as in the picture, the two moments generated by each side cornering force will tend to unroll the car, extending the outer suspension and compressing the inner. This is why, if the IC seats as in the above picture we can assume their effect to be a kind of **antiroll** action. If the IC seats below the ground or outboard with respect to the wheel, on the other hand, the cornering forces moments will have an opposite sign and a kind of pro-roll effect. As we will see, these are the basics to understand a mythical vehicle dynamic concept, namely the **roll centre**. More about this later.

Looking at the complete picture of the forces acting on the vehicle in a pure cornering situation, we will find, beside the already mentioned tyres contact patch forces also the centrifugal force, acting on car CG (we ignore for a moment the separation between sprung and unsprung mass) and any aerodynamic action. For the sake of simplicity, we will focus only on the vertical component of the resultant aerodynamic force (downforce). This will couple together with the weight/gravity action, to compose a resultant vertical force, acting at the CG.

As we saw, the centrifugal force, acting at the CG (or the portion of this force taken one of the two axle) and coupling with the contact patches lateral forces, rolls the car, compressing the outer springs and tyres, extending the inner ones and twisting the antiroll bars, while leading to a load transfer that will increase the vertical load acting on the outer tyres and lower inner tyres one.

At each contact patch, we will have a resultant force with a generic direction, generally different than the one of our idealized swing arm. As we said, this force can be decomposed in a lateral (**F_y**) and a vertical (**F_z**) component. Anyway, we can also decompose it differently, considering the component lying on the line connecting the contact patch and the IC.



This approach allows us to consider not only the magnitude of the resultant force transmitted directly to the control arms (**F_{CA}**), whose lateral component (**F_{CAy}**) is equal to **F_y**, but also to separate the vertical force **F_z** in two summing elements, **F_{CAz}** and **F_s**: the first one is the vertical component of **F_{CA}** and is often referred as Jacking Force, because it tends to lift the car (at least if the IC seats as in our previous pictures); the second one is the portion of **F_z** that cannot be transmitted through the control arms and will be reacted by suspension elastic elements (see the springs and anti-roll bars).

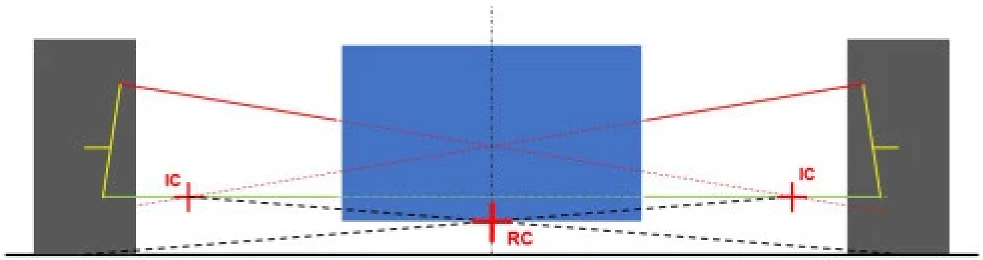
It is easy to see how the relative magnitude of **F_{CAz}** and **F_s** depends strictly on the position of IC or, in other terms, on the inclination of the line connecting the contact patch to IC. Some call this line "**n-line**". Anyway, what is really important is that, given an **F_z**, the bigger the inclination of the **n-line**, the bigger the magnitude of **F_{CAz}** and the smaller the magnitude of **F_s**. This means, the bigger the inclination of the n-line, the smaller will be the force that the elastic elements of car suspensions have to react and, hence, the smaller will be the roll angle of the car for a given stiffness of these elastic elements.

The bigger the inclination of the **n-line**, the bigger will be the antiroll torque generated by the **F_y**, which will play against the car rolling under the influence of the centrifugal force.

F_{CAz} points upward if we consider the outer side (suspension) of the car and downward on the inner side; anyway, since on the outer side its magnitude is bigger than on the inner side (mainly because of the bigger vertical load acting at tyre contact patch, but sometimes also because of suspension movement in roll, leading to a movement of the IC), we will have a net effect pushing the car upward, in a case with the ICs seating as in our picture.
For a given amount of **lateral load transfer** experienced by the car and a given roll stiffness, the steeper are suspensions n-lines, the smaller the roll angle, because there will be a bigger antiroll action generated by the lateral forces acting at the contact patches. Similarly, the **total lateral load transfer distribution**(**TLLTD**) will be influenced both by the relative roll stiffness distribution of an axle with respect to the other (see, how stiff are front springs and antiroll bar compared to their rear counterparts) and by the relative magnitude of the antiroll torque generated by an axle with respect to the other (see, how big is the inclination of the n-lines of one axle with respect to the other, assuming the IC seats as shown in the picture).

This whole discussion simply shows a different approach to what many experts and books identify with the term **roll centre (s)**.

The roll centre of a suspension is defined as the point where the resultant lateral force of an axle can be applied without causing any roll of the body. This point is nothing less than the intersection of the n-lines relative to the left suspension and the right suspension of an axle. If the car doesn’t roll and the suspensions have a symmetric design (the left side is a mirrored version of the right side), the roll centre lies on the car centerline/plane (longitudinal middle plane).



It is worth to come back shortly to the case where the two control arms are parallel to each other and to the ground. In that case, the instant centre IC is undefined and the roll centre lies on the ground.

As probably everybody reading this article knows, the logic behind roll centres definition can be understood if we imagine sliding our $\mathbf{F_{CA}}$ along the n-line till we meet the middle plane of the car. Assuming that both left and right $\mathbf{F_{CA}}$ have the same magnitude (hence a situation where the car negotiates a corner with very low centripetal acceleration, thus experiencing very low load transfer), we will have a resultant force with a lateral component two times the magnitude of $\mathbf{F_{Cay}}$ and a vertical component given by the vector sum of the left and right $\mathbf{F_{CAz}}$ (that normally have opposite signs). This latest force is normally identified as Jacking Force, as we mentioned.

The roll centre concept works very well as long as we assume a symmetrical case, meaning we consider the car having the same suspension design (only mirrored) on both sides, not rolling and having contact patches lateral forces with the same magnitude on both sides. This is, of course, a simplification that proves to produce bigger errors as soon as lateral acceleration (and, hence, the lateral load transfer and roll angle) grows, because we move away from the symmetric case we initially assumed. Moreover, the reader has to keep in mind that the roll centre is a point that moves in space depending on suspension position, for example in roll. With certain design the roll centre could migrate meters (if not kilometres) away from its initial position, thus losing its significance. On the other hand, the inclination of n-lines is something that always remains easily definable and measurable.

Nonetheless, roll centers can be successfully used to get a picture (at least statically) of what to expect in terms of load transfer (and hence, of handling), because under the correct assumptions they are an indicator of how much suspension geometry influences load transfer at the front axle, with respect to the rear.

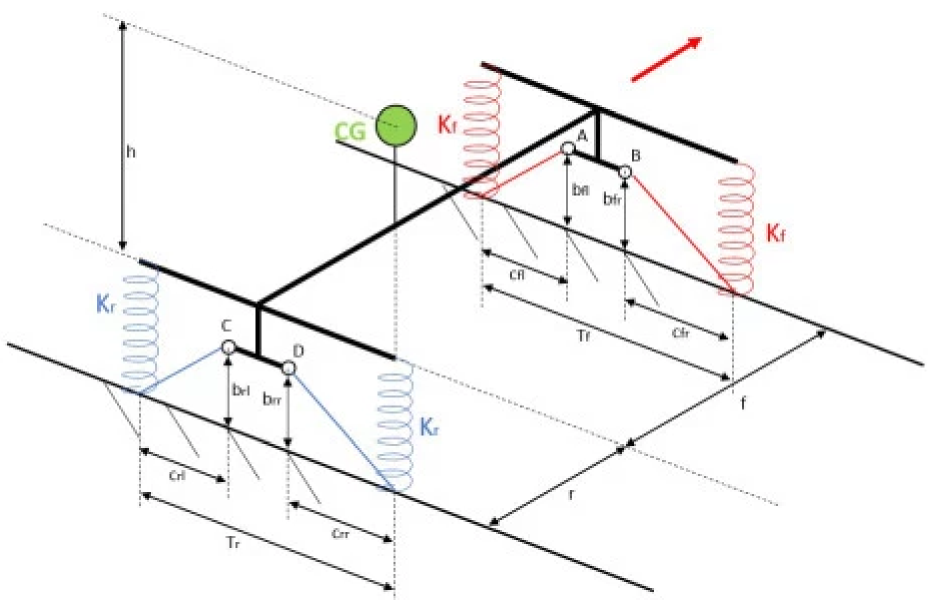
Let’s now analyze how can we put all of this down into equations, with the aim to calculate the load transfer and load transfer distribution of our car.

The load transfer each axle experiences can be divided in a sprung and unsprung mass contribution. Anyway, since the unsprung mass load transfer is not a tunable parameter, to keep our analysis as easy as possible we will simply incorporate its contribution in the overall axle load transfer. Anyway, one thing to mention (and that many books don't explore) is that the unsprung mass has also an influence on the roll angle (although often small), because the forces the unsprung mass experiences (think for example to a wheel assembly) will be transmitted to the chassis through the control arms.

We will consider a simplified vehicle as shown in the following picture, where also all the main dimensions are shown. The chassis is represented by the thick black lines, at which the suspensions, in the form of swing arms, are connected through ball joints. Front suspension arms are shown in red, while rear ones are in blue. Said swing arms connect the contact patch to the chassis, so their constraints to the chassis are simplified representations of the **Instant Centers** we introduced in our previous entries. In the picture we identify these points as A, B, C and D.

$\mathbf{K_f}$ and $\mathbf{K_r}$ are respectively front and rear wheel rates in roll, calculating considering springs, antiroll bars and tyres stiffness. The CG seats at a height "**h**" above the ground and at a distance "**f**" from the front axle and "**r**" from the rear.

Front and rear track width are $\mathbf{t_f}$ and $\mathbf{t_r}$ respectively. The terms named **b** and **c** identify the vertical and lateral distance of suspension ball joints / IC to the contact patch. The astute reader can already recognize the connection between these values and the n-lines slope.
The red arrow indicates the forward direction of motion of our car.

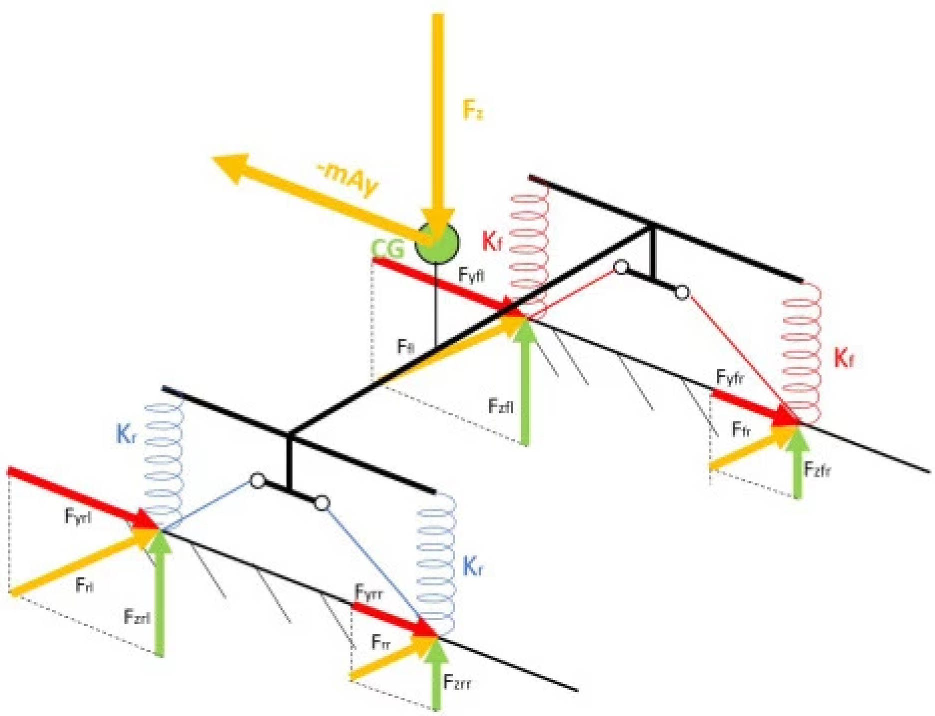


Front and rear roll stiffness can be calculated as previously shown:

$$K_{rollf} = K_f \frac{{t_f}^2}{2}$$

$$K_{rollr} = K_r \frac{{t_r}^2}{2}$$

In the following picture, the same vehicle model is shown, this time including all the external forces acting on the car. Beside the centrifugal force **–mA$_\mathbf{y}$** and the vertical force **F$_\mathbf{z}$** (combination of both weight and downforce, with the car here assumed having the CoP at the same position as the CG, for the sake of simplicity) acting at the **CG** (we ignore aerodynamic drag as it doesn't conceptually influence what we want to show), we have the four contact patches forces, decomposed in their lateral and vertical components: front left tire force, **F$_\mathbf{fl}$** (composed by **F$_\mathbf{yfl}$** and **F$_\mathbf{zfl}$**), front right tire force, **F$_\mathbf{fr}$**(composed by **F$_\mathbf{yfr}$** and **F$_\mathbf{zfr}$**), rear left tire force, **F$_\mathbf{rl}$** (composed by **F$_\mathbf{yrl}$** and **F$_\mathbf{zrl}$**) and rear right tire force, **F$_\mathbf{rr}$**(composed by **F$_\mathbf{yrr}$** and **F$_\mathbf{zrr}$**).



If we consider the equilibrium of the vehicle (in lateral and vertical direction and to the rotation around longitudinal axis) we can write the following three equations:

$$F_{yfl} + F_{yfr} + F_{yrl} + F_{yrr} = -mA_y = F_c$$

$$F_{zfl} + F_{zfr} + F_{zrl} + F_{zrr} = F_z$$

$$(F_{zfl} - F_{zfr})\frac{t_f}{2} + (F_{zrl} - F_{zrr})\frac{t_r}{2} = F_c h$$

Since we know that:

$$(F_{zfl} - F_{zfr}) = (F_{zf0} + \Delta F_{zf}) - (F_{zf0} - \Delta F_{zf})$$

where $\mathbf{\Delta F_{zf}}$ is front load transfer and $\mathbf{F_{zf0}}$ is the static weight acting on each front tyre (the CG is assumed lying on the car centerline, so the static loads acting on each wheel of the same axle are equal), we can derive:

$$(F_{zfl} - F_{zfr}) = 2\Delta F_{zf}$$

And hence:

$$\frac{(F_{zfl} - F_{zfr})}{2} = \Delta F_{zf}$$

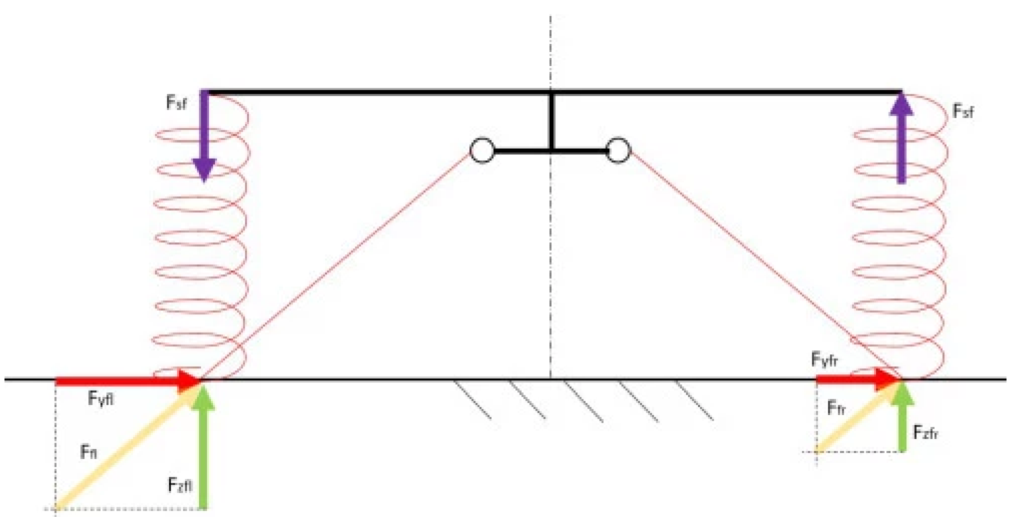
$$\frac{(F_{zrl} - F_{zrr})}{2} = \Delta F_{zr}$$

where $\mathbf{\Delta F_{zr}}$ is rear load transfer.

So we can conclude:

$$F_c h = \Delta F_{zf} t_f + \Delta F_{zr} t_r$$

To derive front and rear load transfers, we can consider the rotational equilibrium of each suspension, indicated in our schematic representation by each swing arm. To start, we will consider the front left suspension, shown in the left portion of the following picture. Please consider that the violet arrows represent here the variation of spring force (F_{sf}) magnitude due to load transfer, with respect to static conditions.



The rotational equilibrium of the front left suspension will be given by:

$$F_{yfl}b_{fl} + F_{sf}c_{fl} - F_{zfl}c_{fl} = 0$$

This can be rewritten as:

$$F_{yfl}\frac{b_{fl}}{c_{fl}} + F_{sf} - (F_{zf0} + \Delta F_{zf}) = 0$$

Analogously, the rotational equilibrium of the right suspension can be expressed as:

$$F_{yfr}\frac{b_{fr}}{c_{fr}} + F_{sf} + (F_{zf0} - \Delta F_{zf}) = 0$$

As we have seen, front axle roll stiffness is given by:

$$K_{rollf} = K_f\frac{{t_f}^2}{2}$$

We also know that:

$$F_{sf} = K_f\Delta s_f$$

where Δs_f is the suspension wheel travel. Combining the two equations we can derive:

$$F_{sf} = \frac{K_{rollf}\emptyset}{t_f}$$

Where Φ is car's roll angle.

Front left and right suspension rotational equilibrium can then be rewritten as:

$$F_{yfl}\frac{b_{fl}}{c_{fl}} + \frac{K_{rollf}\emptyset}{t_f} - (F_{zf0} + \Delta F_{zf}) = 0$$

$$F_{yfr}\frac{b_{fr}}{c_{fr}} + \frac{K_{rollf}\emptyset}{t_f} + (F_{zf0} - \Delta F_{zf}) = 0$$

Summing the two equations together, we obtain:

$$F_{yfl}\frac{b_{fl}}{c_{fl}} + F_{yfr}\frac{b_{fr}}{c_{fr}} + 2\,\frac{K_{rollf}\emptyset}{t_f} - 2\,\Delta F_{zf} = 0$$

And hence (using also the same approach for the rear suspension):

$$\Delta F_{zf} = \frac{K_{rollf}\emptyset}{t_f} + \frac{1}{2}\left(F_{yfl}\frac{b_{fl}}{c_{fl}} + F_{yfr}\frac{b_{fr}}{c_{fr}}\right)$$

$$\Delta F_{zr} = \frac{K_{rollr}\emptyset}{t_r} + \frac{1}{2}\left(F_{yrl}\frac{b_{rl}}{c_{rl}} + F_{yrr}\frac{b_{rr}}{c_{rr}}\right)$$

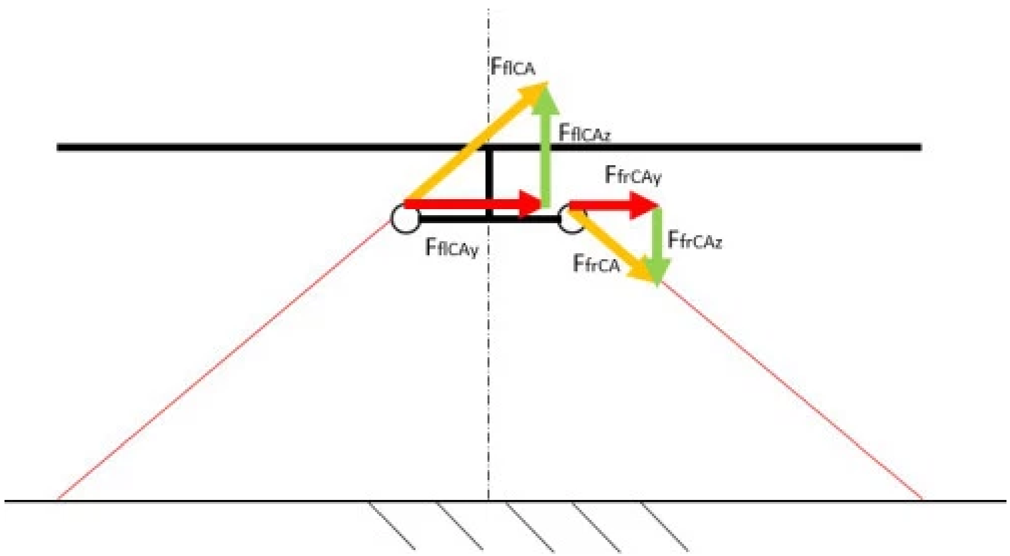
Till now, the roll angle remains unknown. Anyway we can calculate it as follows:

$$K_{rollf}\emptyset = \frac{M_{roll}}{K_{roll}}$$

where:

$$K_{roll} = K_{rollf} + K_{rollr}$$

M_{roll} can be derived considering the equilibrium of the forces applied to the chassis by the control / swing arms.



We know that:

$$F_{flCAz} = F_{flCAy}\frac{b_{fl}}{c_{fl}} = F_{yfl}\frac{b_{fl}}{c_{fl}}$$

Chassis rotational equilibrium can then be written as:

$$M_{roll} = F_c h - F_{yfl}b_{fl} - F_{yfl}\frac{b_{fl}}{c_{fl}}\left(\frac{t_f}{2} - c_{fl}\right) \text{... and analogously for each other suspension}$$

Hence we derive:

$$M_{roll} = F_c h - F_{yfl}\frac{b_{fl}}{c_{fl}}\frac{t_f}{2} - F_{yfr}\frac{b_{fr}}{c_{fr}}\frac{t_f}{2} - F_{yrl}\frac{b_{rl}}{c_{rl}}\frac{t_r}{2} - F_{yrr}\frac{b_{rr}}{c_{rr}}\frac{t_r}{2}$$

With this in mind we can finally obtain:

$$\Delta F_{zf} = \frac{K_{rollf}}{K_{roll}}\frac{M_{roll}}{t_f} + \frac{1}{2}\left(F_{yfl}\frac{b_{fl}}{c_{fl}} + F_{yfr}\frac{b_{fr}}{c_{fr}}\right)$$

$$\Delta F_{zr} = \frac{K_{rollr}}{K_{roll}}\frac{M_{roll}}{t_r} + \frac{1}{2}\left(F_{yrl}\frac{b_{rl}}{c_{rl}} + F_{yrr}\frac{b_{rr}}{c_{rr}}\right)$$

These two equations allow to calculate the load transfer that each axle experiences, basing on the cornering force that each tyre of an axle exchange with the road. Some readers may have already recognized how the terms "**b/c**" multiplying each $\mathbf{F_y}$ are nothing else than the slope of the n-line of each suspension. As we said already, n-lines slopes are closely related to the concept of roll centers and, from a pure conceptual perspective, they are nearly synonyms.

The first term of the equation is what we is often called "**elastic load transfer**", because its value depends on the roll stiffness contribution of each axle to the overall roll stiffness and, hence, on the elastic elements of the suspension (tyres, springs and antiroll bar). It shows how, increasing the roll stiffness of only one axle produces a bigger load transfer on that axle, because the ratio between that axle roll stiffness and overall roll stiffness will increase. In steady state, this means that, increasing front roll stiffness we will create more understeer (or less oversteer).
The second term is what we initially defined as geometric load transfer. It basically shows how, if the n-line of a suspension gets a bigger slope, as identified by the term "**b/c**" (or, in other terms, we increase roll center height), the axle under analysis will experience a bigger load transfer. Again, this means if we increase the height "b" of our front pivot points while "c" stays the same (thus increasing our front roll center height), we will create more understeer (or less oversteer).

Out of ΔF_{zf} and ΔF_{zr} we can easy calculate the Total Lateral Load Transfer Distribution (TLLTD):

$$TLLTD = \frac{\Delta F_{zf}}{\Delta F_{zf} + \Delta F_{zr}}$$

As known, the bigger the TLLTD, the more the car will tend to understeer in steady state.

This formulation generates exactly the same results as the one using roll centers, under the same assumptions (cornering forces having the same magnitude on each side of the same axle and summing up to generate simply a F_{yf} for the front and a F_{yr} for the rear). Anyway, the equations we wrote are more generic and allows to obtain useful results also when considering non-equal cornering forces on the two side of the same axle (in extreme cases a wheel could be in the air, for example and hence produce no lateral force) or even when facing situations where the roll center migrates to very unpractical locations.

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