The Mark Ortiz Automotive

CHASSIS NEWSLETTER

PRESENTED FREE OF CHARGE
AS A SERVICE TO THE
MOTORSPORTS COMMUNITY

September 2006

WELCOME

Mark Ortiz Automotive is a chassis consulting service primarily serving oval track and road racers. This newsletter is a free service intended to benefit racers and enthusiasts by offering useful insights into chassis engineering and answers to questions. Readers may mail questions to: 155 Wankel Dr., Kannapolis, NC 28083-8200; submit questions by phone at 704-933-8876; or submit questions by e-mail to: markortiz@vnet.net. Readers are invited to subscribe to this newsletter by e-mail. Just e-mail me and request to be added to the list.

ROLL CENTER IN TRAILING ARM FRONT SUSPENSION

How does one determine the roll center of a trailing arm front suspension as in the VW beam front end? I have a Formula V with link pin trailing arms front suspension with 2 degree negative camber offset bushings. How can the roll center be raised or lowered in this type of front end? I'm thinking it is similar to a straight axle, because the intersection of the lines drawn through the two trailing arms is at infinity. Please correct me if my premise is wrong.

The short answer to the first part is that the roll center is at ground level, and you cannot adjust it or move it. As to the second point, the suspension behaves like a beam axle in ride, but it is very different in roll. A beam axle has a roll center well above ground level.

That's the short answer, and for practical purposes it's fairly close to correct. However, while both of the above statements are close to correct, they aren't quite perfectly accurate. A beam axle suspension can theoretically have a roll center at or even below ground level, but the linkage required to do this is unusual, and I have never seen an actual beam axle suspension with a roll center that low. And when the VW trailing arm suspension is in a rolled condition, the roll center, properly assigned, isn't exactly at ground level. That is, in the real world, the suspension actually generates a small geometric anti-roll moment when cornering. When the car has two or three degrees of roll, the anti-roll moment from the front suspension is sufficient to produce a significant modeling error if we imagine the roll center to be at ground level.

In any independent suspension, the front (or rear) view force line is an instantaneous perpendicular to the path that the contact patch center travels as the suspension moves. In the VW front suspension, the contact patch always moves straight up and down in front view, relative to the sprung mass. This is also true of many (though not all) beam axle suspensions, in ride, but not in roll. That makes the VW front suspension similar to a beam axle in ride, but not in roll.

When the contact patch moves straight up and down in front view, relative to the sprung mass, the force line, being perpendicular to the line of travel, is always horizontal relative to the sprung mass.

We can also view the trailing arm suspension in terms of short-and-long-arm (SLA) suspension principles. Looking at it this way, the front view projected control arms are: parallel to each other; horizontal relative to the sprung mass; and infinitely long. There is no instant center, or we might say that the front view virtual swing arm is infinitely long. The force line, which would normally run through the contact patch center and the instant center, becomes parallel to the front view projected control arms, and therefore horizontal – again, relative to the sprung mass. It remains horizontal at all ride heights.

The two front view force lines for the two wheels lie right on top of each other: they coincide. They can thus be said to have an infinite number of intersections, or to have no single definable intersection. If we try to define the roll center as the force line intersection, and if we believe that the lateral position of that intersection is significant, this is a very problematic case. The traditional expedient is to make a special rule for this case and say the roll center is on the centerline – but that's obviously arbitrary.

If that weren't enough, suppose we consider that nothing in the real world is made to zero tolerance, so there is a good chance that a real VW's force lines won't be exactly horizontal or parallel. That means they probably do have an intersection, somewhere – and it could be miles from the car, and way above or below the ground. Horrors! The car's behavior is completely unpredictable!

But it's not, of course, and VW front ends are no less tolerant of minor production and setup variations than any others. We merely have a case here that shows up the analytical deficiencies of taking the force line intersection as the roll center. If, instead, we look at the system in terms of individual wheel anti-roll geometry, and assign a roll center – a notional coupling point between suspension system and sprung structure, for lateral forces only – things start to make sense.

The height of each force line's vehicle longitudinal centerplane intercept remains at ground level for any pure ride displacement. The suspension thus has what I call a Mitchell index (and Bill Mitchell calls an incline ratio) of zero, throughout its travel. This means that the suspension's anti-roll geometry does not change in ride, but it changes in roll. In roll, the outside or loaded wheel gains anti-roll, and the inside wheel loses anti-roll.

In the pure trailing arm suspension, the anti-roll is zero on both wheels in the unrolled condition. So as the car rolls, the outside wheel assumes some positive anti-roll, and the inside wheel assumes some negative anti-roll, or pro-roll. The force lines remain parallel to each other (disregarding real-world variabilities), but they are no longer parallel to the ground. They both are horizontal relative to the sprung mass, but the sprung mass is leaning with respect to the ground, and therefore the force lines also both lean with respect to the ground, a similar amount. With respect to the ground, the outside wheel's force line slopes up toward the vehicle centerline, and the inside wheel's force line slopes down toward the vehicle centerline.

Okay, the inside wheel has pro-roll, the outside wheel has anti-roll, the force lines have the same absolute angle relative to the ground, and they intercept the vehicle centerplane equal distances above and below ground – so can't we just average the height of those centerplane intercepts and say the roll center is still at ground level and there is no net anti-roll or pro-roll?

No. The reason we can't do that is that although the force line angles are equal, the forces at the contact patches, that are acting along those force lines, are not equal in magnitude. In most cases, the more heavily loaded tire is making more lateral force than the unloaded one, so the vertical component of the resultant along the force line is correspondingly greater, if the force line angles are equal. We have more cornering force on the outside wheel than the inside wheel, and therefore greater anti-roll force than pro-roll. Net effect: net anti-roll; roll center height above ground.

How far above ground? It depends on two things: the amount of roll, and the amount of front load transfer.

My video, "Minding Your Anti", provides an illustrated explanation of how to assign a roll center height for known force lines and a known or assumed tire-pair lateral force distribution. I like to think I can paint pictures in a reader's mind with words, but explaining this method without illustrations is beyond my powers. Bill Mitchell's method for finding the force-based roll center agrees pretty well with mine.

Anyway, let's take one case as an example: a VW front end with a 52-inch track, rolled 3 degrees, with the inside front wheel at the point of impending lift – all load on the outside tire. Using my method, the roll center height is 2.72 inches. That's significantly different from ground level.

Returning to the original question, you can't adjust the roll center, but it does vary somewhat depending on how much of your total load transfer occurs at the front end and how much the car rolls.

Just as an aside, toward the very end of the era of trailing-arm front suspensions, Porsche experimented with inclined upper-arm pivot axes, to get more anti-roll and camber recovery. That's a story for another time, and not applicable to Formula V, but it is interesting to note that there are ways to make a trailing-arm front end that does not have quite the properties we usually think of.

Also speaking of Porsche – but now referring to Dr. Ferdinand Porsche himself, rather than the modern company – I often wonder what he thought he was getting in terms of roll center with that suspension. It may be that he thought the roll center was at the height of the springs, as with a beam axle. His choice would have made more sense on that basis.

One other note: although a VW front end does generate some anti-roll in real-world cornering conditions, most of the front roll resistance in a Formula V is elastic. The rear roll resistance, assuming we have one of the so-called "zero roll" suspensions, is entirely geometric, if we disregard frictional and unsprung components. The car thus provides an interesting case demonstrating the difference between elastic and geometric roll resistance, and also demonstrating that roll resistance is roll resistance, and you can get roll resistance in different ways, but the tires don't know how you did it. They respond to the amount of load they're carrying, but they can't tell where it came from.

MORE ON ROLL AXIS AND RELATED ANALYTICAL CONCEPTS

In your article in RCE April '06 [drawn from the February '06 newsletter] *you state that "So the end where we lowered the roll centre has less geometric load transfer and the same elastic load transfer as before - hence less load transfer overall."*

This I do not understand: assuming the CG remains in place, lowering the RC means that the distance between RC and CG increases, and the roll moment will increase. Lowering the RC will decrease the geometric load transfer indeed, but it will increase the roll-moment and hence increase the elastic load transfer in my opinion. In the end the total load transfer remains the same, but the way it is divided in elastic- and geometric load transfer differs.

Do you agree, or where am I wrong?

This is the passage in question, from the earlier newsletter:

If we raise one end of the roll axis and lower the other, leaving its height at the c.g. unchanged, the total geometric roll resistance is unchanged, but we increase the geometric roll resistance at one end and lower it at the other. The elastic elements – the springs and anti-roll bars – are not affected by this.

So the end where we lowered the roll center has less geometric load transfer and the same elastic load transfer as before – hence less load transfer overall. This will make that tire pair grip better, because they will be sharing the work more equally. At the opposite end, the elastic component will likewise be unchanged, but the geometric component will be increased – hence more load transfer overall.

The context of my statement is important. I am referring to a case where we lower one roll center and raise the other, so that the roll axis height at the c.g. remains similar. In that case, the total geometric and elastic weight transfers remain the same, and so does the roll angle. The elastic weight transfer at each end of the car remains unchanged. The geometric weight transfer decreases at the end where we lower the roll center; increases at the end where we raise the roll center; and remains unchanged in total. Therefore, the total weight transfer for the end where we lower the roll center decreases; the total weight transfer for the end where we raise the roll center increases; and the total weight transfer for the whole car is unchanged.

Remember, the sprung structure is one essentially rigid mass. It has one c.g., and it is the moment arm of this single c.g. about the roll axis that determines the overall elastic weight transfer. Treating the car as if it had a swivel in the middle will lead you to all sorts of erroneous conclusions.

Addition:

In the same article in RCE you gave a number of reasons why many cars have a nose down roll axis. Another reason I once heard was: to avoid torsion in the chassis you want the same roll moment in both the front axle and rear axle. As the weights front and rear differ, the distance from CG to RC have to differ to get the same roll-moment front and rear.

Torsion costs energy, and since there is only one source of energy - the engine - chassis torsion costs power and consequently speed. So to save speed you want as little torsion as possible in the chassis, and a (more or less) similar torsion-moment in the front and rear will help.

Do you agree with this vision?

No, I disagree. That is, I agree that similar roll resistance front and rear reduces torsional loading on the frame, or at least tends to in hard cornering, but I disagree that this is necessarily what it takes to make the car go fast. In certain cases it coincidentally is. In other cases, such a roll resistance distribution is disastrous.

Again, the sprung mass does not have a front and a rear piece that can act independently. The car is not a tractor and trailer, a locomotive and tender, or two men in a horse suit. The sprung structure is made up of many pieces, all of them having weight and inertia. Each of the components has its own center of mass, and they attach to the frame in various places. All of them apply inertia loads to the frame as the car undergoes accelerations. These loads stress the frame in all sorts of ways, including torsionally. No part of the sprung mass is truly rigid, nor is any other object, but we customarily ignore the deflections within the sprung mass for simplicity. We generally get away with this, because the sprung mass is relatively rigid, compared to the suspension and tires, in most cases. To the extent that the sprung structure is rigid, it is properly treated as one mass, with one center of gravity – most definitely not two independent masses with two centers of gravity.

If the car has 50/50 front/rear weight distribution and identical tires front and rear, and if the car has good aerodynamic balance, and speed is moderate, meaning the rear tires are not using most of their traction to propel the car, then equal overall roll resistance front and rear will get us close to a balance that will please most drivers. If the situation is identical except the car has 55% rear, and we make the roll resistances equal, we will have godawful oversteer. If we make the front and rear roll resistances proportional to the front and rear weight percentages, i.e. 45/55 front/rear, we will have even worse oversteer. Now, if we make the roll resistance inversely proportional to the weight distribution – 55/45 – we'll be closer to right, but there is no guarantee that that will be enough front roll resistance. Probably it will not be.

If we now put smaller tires on the front or larger ones on the back, things change again. Now, the car will not need as much front roll stiffness compared to rear as with equal tires.

If the car is at Daytona or Talladega, running near its top speed, with full power going to the rear tires just to maintain speed, it will need more front roll resistance than it would at lower speeds.

If the car is running American autocross, and the turns are tight, the front wheels will track significantly outside the rears even when the tires are sliding, and the car will need more rear roll stiffness to prevent understeer. In really tight turns, as in Formula SAE/Formula Student, the magnitude of this effect is quite startling.

My point is this: overall roll resistance should not necessarily be equal at both ends, nor should it be proportional to weight distribution. In fact, given equal tires front and rear, roll resistance generally needs to vary inversely with weight distribution: the light end needs more, not less. Even this is a highly non-linear relationship, and only applies if we are comparing otherwise similar cars and conditions. Across a broad range of tire, track, speed, and aerodynamic conditions, required roll resistance distribution has no simple relationship to weight distribution at all!

Now as to the idea that loading the frame in torsion due to unequal front and rear roll resistance absorbs energy and slows the car down, that's mostly nonsense. Yes, the frame does absorb a small amount of energy as it twists and then straightens out again, but the amount is negligible compared to tire and aerodynamic drag.

The frame also twists a bit as we go over bumps, both while cornering and while running straight. However, since the frame is essentially undamped, most of the energy we put in when we deflect it is returned as soon as the load is removed and the frame springs back. Movement of the suspension, on the other hand, does absorb energy, because the dampers oppose both a deflection and the recovery afterwards. We do save energy by softening the damping, but that doesn't necessarily improve handling, and of course controlling the car takes priority over this relatively small energy saving in most applications.

It is also true that at least in some cases we can make the car lighter if we can reduce torsional loadings, and that can raise cornering speed and also save energy. However, usually we are not at liberty to make this a dominant design priority. In passenger cars, there is a case for a live rear axle with a high rear roll center instead of an independent rear with more elastic roll resistance, from the standpoint of weight reduction. With ample elastic roll resistance at both ends, the car needs more torsional rigidity to avoid shakes and creaks over bumps, and this is one reason why cars with independent rear suspension tend to weigh more than cars with beam axle rear suspension.

Really, though, that is an argument for high roll axes rather than sloping ones, and for suspension with a soft wheel rate in warp, rather than an argument for equal overall roll resistance front and rear or for roll resistance proportional to weight distribution.