The Mark Ortiz Automotive

CHASSIS NEWSLETTER

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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.

VIDEOS STILL AVAILABLE

I still have available videos of my lecture, "Minding Your Anti", presented March 2003 at UNC Charlotte. Price is $50.00, which includes shipping and handling worldwide. North Carolina residents please add 7½ % sales tax.

RAISING MY RATES

For the first time in three years, I have decided to raise my hourly rate for consulting. The new rate will be $50/hour, which is still reasonable compared to what other consultants have told me they ask. Retainer rates will likewise go up proportionately. A month will be $300; a year will be $1500.

I was considering having the new rate take effect at the turn of the year, but what I'm going to do instead is offer the old rate of $40/hour, and the corresponding $240/month or $1200/year for all services paid for before March 1, 2004.

REVERSE ACKERMANN OR TOE-IN ON OVALS

I race stock cars and am from the old school of using about 1/8" toe-out. Recently, I've heard of successful stock car racers using significant straight-ahead TOE-IN (e.g. ½" toe-in). And racers who used to believe in running shorter steering arms on their left front spindle to get more Ackermann are now doing the opposite to get reverse Ackermann. They are using tire steering plates [turn plates] to adjust Ackermann and have the toe they want when the tires are steered, but still use a very small amount of straight-ahead toe-out. It's all related to tire slip angles, tire temps, optimized handling, etc. I would appreciate a newsletter addressing this topic.

The December 2002 newsletter did address Ackermann a bit, more in the context of road racing. To recap from that issue:

> There isn't a universally agreed way to express how much Ackermann (toe-out increase with steer) a car has. The closest thing we have is to take the plan-view (top-view) distance from from the front axle line to the convergence point of the steering arm lines, divide the wheelbase by that number, and express the quotient as a percentage. If the steering arms converge to a point on the rear axle line, that's said to be 100% Ackermann. If they converge to a point twice the wheelbase back, that's said to be 50%. If they converge to a point 2/3 of the wheelbase back, that's said to be 150%. If they are parallel, that's zero Ackermann. If they converge to a point twice the wheelbase ahead of the front axle, that's said to be –50%.
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> Supposedly, with 100% Ackermann, the front wheels will track without scuffing in a low-speed turn, where the turn center (center of curvature of the car's motion path) lies on the rear axle line in plan view. This is actually not strictly true, even for the simplest steering linkage, which would be a beam axle system with a single, one-piece tie rod. With either a rack-and-pinion steering system or a pitman arm, idler arm, and relay rod or center link, we can't fully predict what the Ackermann properties will be at all, merely by looking at the plan view geometry of the steering arms. The whole mechanism affects toe change with steer.
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> Even knowing what instantaneous toe we want in a specified dynamic situation is not simple. We don't necessarily want equal slip angles on both front tires. For any given steer angle, the turn center might be anywhere, depending on the situation. All the infinitely numerous possible situations will have different optimum toe conditions. Therefore, there is no relationship between steer and toe that is right for all situations.
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> The toe we have at any particular instant results not only from Ackermann effect, but also from static toe setting and toe change with suspension movement (roll and ride Ackermann).
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> Because of these complexities, there is no single obvious way to define what constitutes theoretically correct Ackermann. It is possible to come up with a rationally defensible definition for your own purposes, but there is no standard rule, and it is unlikely that there ever will be.

With oval track cars, we can have additional complexities. As the questioner notes, it is common to use unequal-length steering arms on oval track cars, usually shorter on the left. In fact, this is the only way to get positive Ackermann on a front-steer stock car, unless the rules allow fabricated spindles and we accept a much larger scrub radius than we'd like. A shorter left steering arm only gives positive Ackermann when the wheels steer left, at the expense of exaggerating negative Ackermann when steering right. This is not an option when the turns go both ways.

The questioner referred me to a website where an expert says that tire slip angle is a property of the tire, which should be available from the tire manufacturer. That is incorrect. Slip angle is an **operating condition** of a tire at a particular instant: the angle between the wheel's aim and its direction of actual travel. A tire does not have a single slip angle. It has a measurable slip angle, when tested on a machine, at a specific load (or normal force), a specific desired lateral force, a specific camber, a specific pressure, a specific temperature, a specific rim width, and a specific wear condition. Change any of these factors, and the slip angle changes. Change the properties of the road or simulated road surface, and the slip angle changes.

The body or sprung mass can also be said to have a slip angle, since it also has a definable direction of aim and direction of travel. An aircraft or watercraft also has this type of slip angle. Slip angle as measured by recently introduced GPS-based data acquisition systems is body slip angle. In a large-radius turn, body slip angle is very similar to rear wheel slip angle, provided there is no large amount of roll steer, ride steer, or static steer at the rear wheels. Front wheel slip angles can be very different from the body slip angle, since the front wheels can steer.

Defining the tire's direction of travel can be a bit enigmatic. At first blush, we might suppose it would simply be the same as the body's direction of travel. For situations where we don't need tremendous precision, and for large-radius turns where the vehicle's yaw velocity is small, this can be an adequate approximation. But when the body has a yaw velocity – and it must have some in any steady-state cornering condition – the tires will not be traveling in exactly the same direction as the body's origin point or CG.

As an example, suppose we are running on a quarter-mile oval where the turns are half of the lap distance. These would be fairly tight turns by oval track standards, about 105 foot radius. Let's suppose for simplicity that the car's CG, or our chosen origin point, is midway along the wheelbase and centered side to side. If the car has a 9' (108") wheelbase, the wheelbase subtends or spans an arc of about 5 degrees on a 105' radius. If the car is driving through this turn very slowly, the tires not sliding significantly, then the center of curvature of the car's path will lie on the rear axle line as seen from above. The body's origin or center of gravity will track outside the midpoint of the rear axle. The midpoint of the front axle will track still further out, and the front wheels will track outside the rears. The front wheels will be steered an average of about 5 degrees to the left. The left front will be leading the right front, and will need about ¼ degree more steer angle than the right front if we want least scuffing, tire wear, and rolling resistance. That's about 1/8" toe-out as measured with typical toe plates, or 1/16" total as measured at the wheel rims using a string or laser.

If we measure the car's body slip angle using a GPS-based data acquisition system, it will tell us the car has a negative slip angle: it is traveling about 2.5 degrees left of the direction it's pointing, while making a left turn!

Now suppose the car is going faster, and the rear wheels need to run at about 2.5 degrees of slip angle to keep the car on course at the speed it's running. The center of curvature will now lie on a line perpendicular to the car's centerline, intersecting the centerline at the CG or origin. The GPS will now tell us we have a slip angle of zero. That won't be what the tires are feeling.

The front tires will now need only half as much toe-out as in the previous case, or about 1/8 degree, for their slip angles to be equal. The rear tires will need about 1/8 degree toe-in for their slip angles to be equal. At the front, the left wheel leads the right slightly. At the rear, the left trails slightly.

Okay, now let's raise the speed again, to a point where the rear wheels have about a 5 degree slip angle. The GPS will now tell us we have a positive slip angle: the car is travelling about 2.5 degrees to the right of the direction it's pointing, while making a left turn. The center of curvature now lies on the front axle line. Neither front tire leads the other. For the front tires to have the same slip angle, they now need to have no toe-in or toe-out with respect to each other. At the rear, the left now trails enough so the rear tires would need ¼ degree toe-in to have equal slip angles.

If we go still faster, the center of curvature moves ahead of the front axle line. The front wheels now need toe-in for their slip angles to be equal.

Now suppose we try a similar comparison, except the track is 2 miles long, and the turns have a radius of 820 feet. The same car’s wheelbase now subtends an arc of only about 0.6 degree. The center of curvature is now on the front axle line at only 0.6 degree rear wheel slip or 0.3 degree body slip. Any time the car is cornering hard on this turn, the center of curvature is well ahead of the front axle line, and the front wheels will require toe-in for equal slip angles.

Do we want equal slip angles? Not necessarily.

Although we cannot describe a tire’s properties as they relate to slip angle behavior with a single number, we certainly can meaningfully discuss them, and also generalize about them to some extent. For a desired set of conditions, we can measure them on a tire testing machine. For a given set of conditions, a tire will have some slip angle at which it develops maximum lateral force. If we increase slip angle beyond this, lateral force drops off. This is what happens when a tire reaches its limit and breaks away. For street radials at typical loads and pressures, this occurs somewhere around 6 degrees.

In general, bias-ply tires develop peak cornering force at higher slip angles than radials. Narrow tires develop peak cornering force at higher slip angles than wide ones. Tires at low inflation pressures develop peak cornering force at higher slip angles than at high inflation pressures. Tires with thick or deep tread develop peak cornering force at higher slip angles than tires with shallower treads. This is particularly true with treaded tires, but the effect is measurable with slicks also.

As we increase normal force on a given tire, other conditions constant, the slip angle for peak lateral force increases. For most applications, this means that in a left turn, the right front tire develops peak lateral force at a higher slip angle than the left front. This is why we might want toe-in when cornering, or more toe-in than required for equal slip angles.

If both front tires are operating at peak cornering force together, that should be the greatest total cornering force available from the pair. However, we should also think about the longitudinal forces, as these also affect the car’s balance. If both front tires are optimized for lateral force, the right one will not only be making more lateral force than it would with more toe-out or less toe-in, it will also be making more drag. This will tend to add understeer. So the toe for least understeer may be a little different than the toe for greatest front lateral force.

We also need to remember that if both tires peak together, they break away together too. So part of the price for greatest peak lateral force is more sudden breakaway. That means that exploiting the lateral force capability may be harder for the driver.

Looking at effects on tire temperatures, in oval track racing the right front tire can easily overheat. The left front seldom gets hot enough to be a problem. Running less toe-out or more toe-in through the turns doesn't help this, it makes it worse.

Whatever slip angles we decide we want on the front tires, we can definitely say that we are going to want less toe-out or more toe-in as the center of curvature moves forward relative to the car. In other words, the larger the radius of curvature or the more the rear tires are sliding, the more toe-in or the less toe-out we want.

What are the implications of this for the optimum relationship between the steering arm lengths? Do we want more positive Ackermann, or less negative, when steering left, or when steering right? I think the tradition of having the left steering arm shorter, if anything, is sound. When the wheels are turned left the most, the center of curvature is furthest aft. When the car is crossed up, steered right in a powerslide or catching a slide, the center of curvature is ahead of the car and we need toe-in. Some experts say having toe-in or less toe-out when countersteering "pins the front end" and spins the car. This is based on the misconception that the wheels always drag more when toed in. As we have seen, this is not the case when the center of curvature is ahead of the car, and the left front is trailing the right front.

So here are my recommendations:

1. Whatever your strategy, the combination of static toe and Ackermann has to give you a good toe value for your prevailing conditions. Wrong Ackermann with a toe setting that compensates is better than improved Ackermann with static toe that doesn't suit.
2. For road racing or street use, where a wide variety of conditions will be encountered, a combination of substantial positive Ackermann and moderate static toe-in is the way to go. This is the prevailing factory approach for road cars, and also, I'm told, the approach taken by Cornell on their winning FSAE cars.
3. For pavement oval-track applications, toe-in may make sense, if combined with positive Ackermann when steering left, especially on high-speed tracks, provided that right front tire temperature considerations don't overrule the decision. Making the right steering arm shorter than the left does not make sense, although it may work as a way to crutch static toe-out on a high-speed track.

It is worth noting that this whole question has been subject to fads throughout the history of racing. In road racing, the first successful rear-engined cars used negative Ackermann. This was in the late 1950's and very early 1960's. The designers claimed this was because the more heavily loaded outside tire required a larger slip angle. In the early '70's, a bit of static toe-out, with zero Ackermann, was the most popular choice. By the 1990's, it was becoming commonly recognized, especially in CART, that high-speed ovals demanded much less Ackermann than road courses, and street circuits demanded the most. Far as I've heard, nobody has yet tried static toe-in for the high-speed ovals, but it's possible my latest information isn't entirely current.

MANAGING THE BEHAVIOR OF THE UNBALANCED

I recently read your article, Load Transfer Basics, in the November issue of Racecar Engineering (Vol.13 No.11)[based on the August 2003 newsletter] and was very interested in the concepts you discussed. I am currently an engineering student in Australia and I am doing some work experience with a Porsche racing team out here. I would be very grateful if you would be able to help me with several questions I have concerning load transfer and suspension set up.

Firstly I was wondering if you could recommend any text books relating to load transfer and suspension set up so I can pursue some further research in this area.

Secondly during a race meeting held a month ago it was mentioned that the way to get faster lap times is to "balance" the car by getting the static load on the front left suspension divided by the load on the back right equal to the opposite diagonal. In other words getting the following ratios equal:

Front Left : Back Right = Front Right : Back Left

Our team thought that this sounded a reasonable proposition but we are unable to see exactly how this works. It seems to me that if this theory does in fact make the car more "balanced" it assumes that the left handed corners are similar to the right handed corners. Since this is often not the case surely the car must be preloaded to suit the important corners on the track, in which case the broad statement of balancing the car isnt helpful...?

I like the chapter in Milliken & Milliken's *Race Car Vehicle Dynamics* by Dave Segal on the subject (Chapter 18). The book is published by SAE, and available from their bookstore at www.sae.org. I would also of course not pass up the opportunity to plug my own video on the subject, mentioned at the beginning of this newsletter.

Another way of stating the relationship you were told would be to say that the rear percentage taken diagonally is the same for both diagonal wheel pairs.

For a car with 50% left, this works. You would have a car with 50% diagonal, the same rear percentage on both sides, and the same left percentage at both ends. Such a car should corner with similar balance in right and left turns.

However, as you note, sometimes even in road racing we want the car heavier on the side it turns toward the most, on a particular track. It may also happen that we get something other than 50% left unintentionally, due to packaging or rules constraints. We still will normally want the car to corner similarly in both right and left turns, even though it will be faster when turning toward its heavy side.

My recommendation for this is to start with what I call an unwedged car. My definition of this is a car with equal rear percentage on both sides, and equal left percentage at both ends. This is different than what your instructor taught you, and also different from 50% diagonal weight.

Let’s consider an example. Suppose we have a 2000 lb. Porsche with 60% rear, and we are racing on a road course with predominantly right turns. Suppose the car has right hand drive, and we have enough ballast so we can get 55% right. An unwedged car, as I define it, would have 55% of 60%, or 33%, on the right rear (660 lb.); 55% of 40%, or 22%, on the right front (440 lb.); 45% of 60%, or 27%, on the left rear (540 lb.); and 45% of 40%, or 18%, on the left front (360 lb.). The car has the same 45% left at both ends, and therefore LF/RF = LR/RR. Also, it has the same 60% rear on both sides: LF/LR = RF/RR. This setup imposes no static torsional load on the frame.

Note that we can get either of these equations from the other by algebraic manipulation. (Starting with the former, we multiply both sides by RF, and divide both sides by LR.) They are different forms of the same expression. However, there is no way we can manipulate either equation to obtain the form LF/RR = RF/LR. That is a different equation.

If we look at the ratios in your instructor’s equation, in the above example, we get: LF/RR = 18%/33% = 360/660 = .545; RF/LR = 22%/27% = 440/540 = .815. Not close at all.

Suppose we look at the above example with reference to the target more conventional wisdom would set, namely 50% diagonal. That would mean 2(RF+LR) = RF+RR+LF+LR. Our example doesn’t fit that rule either. Its diagonal percentage is 27% + 22% = 49%. Close, but not the same.

Diagonal weight would be 980 lb. To increase that to 1000 lb. and make the diagonal 50%, we would have to adjust the suspension to transfer 10 lb. left-to-right at the front and transfer 10 lb. right-to-left at the rear (assuming equal track widths, for simplicity). That would mean that the static left percentage at the front would be 350/800 = 43.8%; left percentage at the rear would be 550/1200 = 45.8%. If the car has equal roll resistance in both directions front and rear, we can see that the change we’ve made to the static load distribution will help the front wheels in a right turn and the rear wheels in a left turn. We would therefore expect more understeer in left turns than in right turns, which is probably not what we want.

Now, suppose we wanted to set this car up to your instructor’s rule. Could we even do it? Well, yes, we could, but it would take a pretty freakish setup to do it. We’d have to have 600 lb. on the LF, 900 lb. on the RR, 200 lb. on the RF, and 300 lb. on the LR. This is obviously not what we want. Our diagonal percentage would be 25%. Our left percentage would be 75% at the front, 33% at the rear. Yow! The car would push like a dump truck in right turns and spin if you sneeze in left turns.

Rather interestingly, if the car has 50% left, all three rules work. If the car has either 50% left or 50% rear, the 50% diagonal rule works. If the car has neither 50% left nor 50% rear, my rule works, or gets you closest.

Even with my rule, small adjustments may be needed. If there is more understeer turning right than turning left, add load to the RF and LR. If there is more understeer turning left than turning right, add load to the LF and RR.

INDEPENDENT OR BEAM AXLE FRONT SUSPENSION FOR PAVED OVAL?

We race primarily on quarter-mile asphalt ovals – no banking, no bumps. The car weighs 700 kilos (1500 lb.) and runs on 10-inch Hoosier slicks front and rear. Everyone uses A-arm front suspension, with a mandatory beam axle rear. Would there be any advantage running a beam type front end? If the steering rack is mounted on the axle, would the driver feel roll-induced steer?

The main advantage of a beam axle front end for an oval-track application is that it reduces overall weight, and also usually reduces the torsional stiffness needed from the frame. Even if you have a minimum weight requirement, reducing car weight is advantageous because you have more ballast to move as desired. This is especially beneficial if you have no limit on left percentage, or if an engine setback limit prevents you from getting as much rear percentage as you'd like.

A beam axle also eliminates camber change due to roll, which is good. However, on an oval, independent suspension can be set up with sufficient static camber so that you can also obtain desired camber when cornering with an independent system.

Back when I was doing consulting for fun and experience, I worked with a west coast super modified team. At that time, there were no rules on suspension design. Beam axles have now been made mandatory front and rear for these cars. (Interestingly, in the US, other classes prohibit beam axles and mandate independent front ends. There is no universal consensus as to which is better.) Back in the '80's, people ran both styles, and the usual practice was to mount the rack on the axle on the beam axle cars. This works fine. The driver does feel a little roll steer, but it's not bad, and it's in the right direction, i.e. roll understeer.

The main advantage of the independent setups in this class was that the car was more controllable at the point of inside front wheel lift. With a beam axle, especially with a high roll center and a soft wheel rate in roll, when the inside front wheel comes off the ground, the roll center rises as the axle rises, causing the wheel to rise further. The wheel tends to rise a lot then, causing unfavorable camber on the outside front, and therefore understeer. It is difficult for the driver to maintain an attitude where the inside front tire is carried just a little way off the ground. With independent, this is much easier. However, I believe that with a low roll center and a high wheel rate in roll, a beam axle could deliver similar controllability. And if your cars never reach the point of carrying a wheel, the whole issue is moot.