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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.

UPCOMING SEMINAR

The Racing Drivers Club in San Francisco has invited me to present a one-day seminar consisting of a lecture on basics of vehicle dynamics, followed by a question-and-answer session. Location is the Sheraton Hotel near the SF International Airport. Time is 8:00am on Saturday, August 10. Fee is $250 for club members, or $300 for non-members. Since dues are only $30, I think the club stands to recruit some members.

For more information please contact Arthur Muncheryan at rose1art@earthlink.net.

STEERING GEOMETRY VARIABLES

In response to numerous requests for information on steering geometry, I am going to attempt a reasonably complete explanation of the various parameters and their effects.

The *steering axis* is a line about which the wheel steers, usually through the two ball joint centers of rotation in an independent suspension, or the kingpin axis in a beam axle. This line can be defined by the point where it intersects the ground and by its angular orientation. These are commonly described in terms of the X and Y coordinates of the ground intercept, with respect to a local origin at the contact patch center, and the transverse and longitudinal angles relative to ground plane horizontal.

The front view distance from ground intercept to contact patch center, or local Y, is called *scrub radius*, or *steering offset*. It would make more sense to call the top view distance from ground intercept to contact patch center the scrub radius, but most people use the term to mean the Y or transverse component of this. This quantity is generally considered positive when the contact patch center is outboard of the ground intercept.

The side view distance from ground intercept to contact patch center, or local X, is called *trail*, or sometimes *caster trail* or *mechanical trail*. It is positive when the ground intercept is forward of the contact patch center.

The front view angle of the steering axis from ground vertical is called *steering axis inclination* (SAI), or sometimes *kingpin inclination* (KPI). It is positive when the steering axis tilts inboard at the top, which is almost always the case.

The side view angle of the steering axis from ground vertical is called *caster*. It is positive when the steering axis slopes rearward at the top.

These parameters are controlled partly by the design and adjustment of the control arms, and partly by the design of the spindle, or spindle/upright assembly, together with the hub and wheel.

The term *spindle* can mean either the stub axle, or pin, that carries the bearings, or the assembly including this pin and the upright, especially when these are one piece.

The spindle or spindle/upright determines two important parameters: *spindle inclination* and *pin lead* or *pin trail*.

Spindle inclination is the front-view inclination of the steering axis, relative to pin or wheel vertical, as opposed to ground vertical. Spindle inclination approximately equals SAI minus camber. Spindle inclination is almost exactly identical to SAI when camber is zero. It is exactly identical when both camber and caster are zero.

The steering axis and the wheel axis do not have to intersect, unless we want the right and left uprights to be identical parts, with bolt-on steering arms and caliper brackets. The steering axis can pass behind the wheel axis, as it does on a bicycle. The perpendicular distance between the two axes is called *pin lead*. This is equivalent to the dimension we call fork rake on a bicycle. If the steering axis passes in front of the wheel axis, that's *pin trail*. So pin trail is negative pin lead, and vice versa.

Effective pin length is the distance, along the wheel axis, in front view, from the steering axis to the wheel centerplane. This distance depends on the wheel and hub as well as the spindle/upright.

We now have sufficient vocabulary to describe and discuss basic steering and spindle geometry. If we can specify all the quantities above, we have enough data to construct a stick model of the basic steering geometry.

We may want to add steering arms. For purposes of spindle/upright design, we can define the position of the outer tie rod end with respect to the pin and the steering axis. We may define a height from pin axis to tie rod end center of rotation. To do this in a manner appropriate for drawing the upright, or inspecting it when removed from the car, this should be the vertical dimension in side

view, assuming zero caster and camber -- in other words, we are projecting to the wheel plane, and taking the steering axis in side view as our local vertical.

In such a side view, we may construct a horizontal line from tie rod end to steering axis. This is our *side view steering arm length*.

We may project a top view from the side view, and locate the lateral position of the tie rod end. If we have a longitudinal line corresponding to the side view steering arm described above, we may construct a transverse line from it to the tie rod end, and measure that distance. This we may call *steering arm offset.* It will usually be outboard for a front-steer layout, and inboard for a rear-steer layout. I don't know what sign conventions other people use, but I generally call outboard positive for front steer and inboard positive for rear steer. Thus positive offset is the direction that gives us positive Ackermann.

In terms of coordinates, we are establishing a local origin where the side view steering arm meets the steering axis. The side view steering arm length is our local X, and the steering arm offset is our local Y.

This doesn't mean there's anything wrong with assigning global or front-suspension coordinates to the tie rod ends when doing an overall front end layout. I'm just pointing out that at some point you will have to deal with the spindle/upright/steering arm unit as a sub-assembly, off the car, and it helps to be able to measure and discuss it that way too.

Now we have a fairly complete vocabulary to describe steering geometry, so we can discuss what effects these parameters have.

Trail causes *lateral* forces at the contact patch to produce a torque about the steering axis. This causes the steering to seek a gravitational/inertial center. The driver feels lateral cornering force through the steering. He also feels the lateral force that the tires must generate to make the car run straight on a laterally sloping, or cambered, road surface. It is worth noting that this is only one component of the self-centering forces the driver feels. Another is the tire's own *self-aligning torque*, which is present whenever the tire runs at any slip angle. This will provide some feedback of cornering force even in the total absence of trail. This effect is sometimes described as mimicking trail. The amount of tire self-aligning torque, divided by lateral force, is sometimes called *pneumatic trail*. Note that this is a calculated value which depends on tire properties, and not an actual steering geometry parameter.

One important distinction between the forces from trail and tire self-aligning torque is that tire self-aligning torque is not a linear function of lateral force. It builds at a decreasing rate as lateral force increases, and at a point a bit short of maximum lateral force it actually begins to decrease. This means that if our car has little or no trail, the steering will start to go light a bit *before* the point of tire breakaway. Some argue that this is a good thing, especially for a passenger car, because it gives

the driver a signal to ease up short of the point of actual loss of control. In a race car, this type of steering feel requires that the driver be accustomed to driving just a controlled increment beyond the point where the steering wheel tells him/her that the limit of adhesion has been reached. If the driver is used to having more trail, he/she will often find this very difficult.

Trail also causes a small lateral movement of the front of the car with steer, in the direction of steer. We might call this *steer yaw*. It can rationally be argued that this improves turn-in, both by yawing the car promptly and by causing the rear wheels to develop a slip angle promptly.

Scrub radius or steering offset causes *longitudinal* forces at the contact patch to generate a torque about the steering axis. If right and left scrub radii are equal and longitudinal forces at the right and left wheels are equal, no net torque at the steering wheel results. The driver feels the *difference* between the longitudinal forces at the front wheels. The driver feels one-wheel bumps, brake pulsations, and crash impacts where one wheel hits something, in direct proportion to scrub radius.

A car with a lot of scrub radius is sensitive to wheel imbalance and tire and brake imperfections, has a lot of "wheel fight", and has greater tendency to injure the driver's hands in one-wheel crash impacts or curb or pothole impacts. A car with very little scrub radius is less subject to these problems, but the steering will tend to be numb and uncommunicative.

A car with large scrub radius *may* steer more easily at parking speeds, depending on other parameters, provided the brakes are not applied. This is because the wheels can roll as they steer rather than purely scuffing. With the brakes applied and the car stationary, a car with a small scrub radius steers more easily.

Caster causes the front wheels to lean in the direction of steer. With a given spindle/upright geometry, more caster implies more trail.

Caster combined with trail causes s*teer drop* or *steer dive*. The front of the vehicle drops as the wheels steer away from center, if caster is equal on right and left. This tends to cause an anti-centering force at the steering wheel. It is the reason why the front wheels of a dragster at rest tend to flop to one side or the other.

Caster combined with scrub radius causes the car to drop as the wheel steers forward (toes in), and lift as the wheel steers rearward (toes out). When this occurs on the right and left wheels as one steers forward and the other steers rearward, the result is *steer roll.* The car leans away from the direction of steer. The wheel loads also change. The car de-wedges: the inside front and outside rear gain load; the outside front and inside rear lose load. This effect can help the car turn in slow corners, especially with a spool or limited-slip differential. In excess, it can create low-speed oversteer and over-sensitivity to steering angle. In general, cars running on lower-speed tracks need more steer roll, and cars on fast ovals should have very little.

The camber change associated with caster is favorable, particularly for road racing cars, which usually cannot get favorable camber on both front wheels any other way. We can have too much of this good thing, but that's extremely uncommon.

Steering axis inclination (SAI) causes both front wheels to gain positive camber as they steer away from center.

SAI combined with scrub radius causes *steer lift*. The front of the vehicle rises as the wheels steer away from center. This induces a self-centering force in the steering which seeks vehicle center rather than inertial/gravitational center. This is particularly useful in passenger cars because it reduces the car's tendency to follow road camber, and therefore reduces the need for the driver to pay close attention in casual driving on roads with varying slope. The centering force also tends to suppress steering shimmy.

In race cars, the camber change associated with SAI is unfavorable on the outside wheel. The self-centering force increases steering effort, which is a factor for any vehicle without power steering. It

also creates what could be considered a false message to the driver about the lateral forces present at the contact patches. There is therefore a rational case for using more caster and less SAI in a race car.

With the packaging constraints we usually face, more SAI generally implies less scrub radius. The main limitation will often be how far outboard we can place the lower ball joint without having it too close to the brake disc. If the wheel has generous negative offset, we may instead be limited by the wheel rim hitting the control arms in some combinations of suspension motion and steer. Either way, we often cannot place the entire steering axis as far outboard as we would theoretically like to. Using SAI allows us to at least get the ground intercept further outboard in such cases. With MacPherson strut front ends, large amounts of SAI are necessary if we are to obtain any camber recovery in roll.

Consequently, in many cars we see SAI used for reasons not directly related to SAI's own dynamic effects.

A full discussion of **Ackermann effect** (increase of toe-out with steer) is beyond our scope here, but we can at least say that in low speed turns with the wheels steered into the turn, the car generally needs toe-out on the front wheels. For high-speed sweepers or ovals, the front wheels generally need toe-in instead. The key determining factor is whether the *turn center* -- the instantaneous center of curvature of the car's path -- is ahead of or behind the front axle line. Other determining factors include the tendency of the loaded wheel to want a larger slip angle than the unloaded one, and what yaw moments we wish to create with the tire drag forces.

The attitude of the front wheels at any given instant depends on both the static toe setting and the change in wheel-to-wheel toe with steer. This means that optimum Ackermann depends on static toe setting.

It should be clear, then, that there is no such thing as perfect Ackermann properties. But we can at least say some definite things about what geometric parameters will do to Ackermann. In particular, increasing steering arm offset increases Ackermann effect.

Ackermann for oval track cars is often asymmetrical. The side view steering arm length is less on the left wheel than on the right. This produces more Ackermann when steering left than when steering right.

We should mention that if we are willing to tolerate a bit of additional complexity, there are ways around some of the tradeoffs in steering geometry. For example, it is possible to create a self-centering force by springing the steering system. This can mimick the self-centering that we get from SAI, without the adverse effects on camber. We can also damp the steering to reduce kick and shimmy.

We can get small SAI and small scrub radius at the same time by using compound control arms (two single links replacing the usual wishbone or A-frame) and dual ball joints. This gives us an instantaneous virtual ball joint outboard of the linkage itself. We can adopt this arrangement at the upper end of the upright, or the lower end, or both.

IMPORTANCE OF STEERING RACK PLACEMENT

I have read that one should either place the steering rack above and behind the wheel axis, or below it and ahead of it, due to deflection steer considerations. How important is this really in race cars, where there is no rubber in the suspension or the steering mounts?

The stiffer everything is, the less this matters. Most formula cars nowadays violate the rule you mention; the steering is ahead of the wheel axis and above it.

Of course it is still true that all cars have some deflection steer, and we would prefer that this be deflection understeer rather than deflection oversteer.

Actually, what determines the critical height for this is not necessarily the height of the wheel axis. Rather, it is the point of zero lateral deflection at the upright, which varies with the combination of loads on the wheel, the geometry of the system, and the distribution of rigidities in the components.

Another consideration is that we may prefer the tie rod on the loaded wheel to be in tension rather than compression, on the logic that the tie rod tends to be the slenderest element in the system. This concern argues for front steer in all cases.

BUMP STEER AS ACKERMANN MODIFICATION

I have a Pantera that has very little bump steer in compression from static position, but has quite a lot of toe-in in droop. It also has a fair amount of Ackermann. Could it be that this is intentional, to provide less Ackermann in hard driving when there is more roll, yet have adequate Ackermann in tight turns at low speed? Have you ever heard of a car being designed that way on purpose?

I haven't ever worked with a Pantera, but I don't really think the combination was thought out on the logic you describe. It is quite common, though, for production cars to exhibit the type of bump steer you describe. The cause is usually that the inboard pivot axes of the lower control arms are splayed out toward the rear in plan view, while the inboard pivot axes of the upper arms are not so much so, or even angle outward toward the front.

This makes for a side view virtual swing arm that shortens dramatically in droop, causing caster to diminish at a rapidly increasing rate in droop. That unavoidably causes some toe-in in droop if the geometry is laid out to avoid major toe change in bump.

The usual reason for designing the lower control arm this way is to get more front view control arm length, while still making room for the engine or the footwell.

THE OLD TORQUE/HORSEPOWER CONTROVERSY

More horsepower makes a car accelerate quicker – true or false?

The above question was recently included in a quiz in a magazine, written by a reputable author who gets this newsletter. To my surprise, he said the correct answer was "false". He explained that, assuming all else identical, including tire size, more acceleration implies more axle torque, not necessarily more horsepower.

What this reasoning misses is that since power is the product of torque and rpm, more torque at the same rpm *is* more power, at the axle.

Now, if we have more axle torque at a given axle rpm, what can we say about conditions at the engine flywheel? Do we have more torque at the flywheel? Not necessarily. The author himself points out that we could have identical torque, and more torque multiplication due to lower (numerically higher as customarily expressed) gearing.

Okay. So if there is identical torque at the flywheel, identical speed at the axle, and shorter gearing, what do we know about engine rpm? It must be higher, in direct proportion to the gear. Again, power is the product of torque and rpm. Therefore, if torque is identical and rpm is greater, power is greater, at the engine.

If the axle has more torque at a given speed, that alone does not tell us anything about torque at the flywheel. But we can say with certainty that both the product of engine torque and gear, and the product of engine torque and engine rpm, are greater. This means we know that both power at the engine and power at the axle are greater.

Looking at it another way, power is the rate of energy flow. To make the engine add kinetic energy to the car at a greater rate, energy must flow from the engine at a greater rate.

Note that this is a somewhat different question from whether to build a "horsepower motor" or a "torque motor". What counts at a particular instant is horsepower *at that instant*, not at the peak of the power curve. In many cases, especially when running a short oval without shifting, a car will exit a turn at an rpm closer to its torque peak than its power peak. In such a case, making the engine strong at the torque peak may make for better lap times – but we are still talking about a *power* increase (achieved through more torque at comparable rpm – or alternatively through comparable torque and more rpm and gear, or some combination of both approaches), even though the increase is at a point in the rpm range that is below the engine's power peak.

So a broad power band is worthwhile, and may be worth more than maximum peak horsepower. But at a particular instant, at a particular speed, more axle torque definitely implies more engine horsepower, and by itself implies nothing about engine torque.