December 2002

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.

ACKERMANN RECOMMENDATION

I am modifying a road racing Formula Ford for SCCA Solo 2 [American autocross]. I am considering adding more Ackermann effect to make the car work better in tight turns. Is this a sound idea, and if so, what do you suggest for geometry?

Without writing a really long piece on Ackermann, yes you will probably help the car. I don't know what geometry you have now, but as a general rule a car needs more Ackermann for events with tight turns, e.g. autocross or hillclimbs.

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%.

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.

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.

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).

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.

Having entered these abundant caveats, I will now make some general-purpose recommendations for autocross and hillclimb applications:

1. In plan view, at zero steer (straight-ahead position) the steering arms should converge to a point somewhere between the rear axle line and the midpoint of the wheelbase. In traditional parlance, that's somewhere between 100% and 200% Ackermann. The tighter the turns, the higher the percentage.
2. At all steering positions, the rack or relay rod should be either slightly behind the outer tie rod ends or even with them. This applies to both front steer and rear steer cars. With rack and pinion steering, it means that at zero steer, the rack should be a bit behind the outer tie rod ends on a front steer car, and about even with the outer tie rod ends on a rear steer car. Purpose of this is to assure that tie rod angularity adds Ackermann at large steer angles, rather than subtracting.
3. Angle between any arm and any link in the system should never be less than 30 degrees or greater than 150 degrees. This helps to assure that the mechanism cannot snap over-center due to deflections of the components. Alternatively, over-centering can also be prevented by provision of stops at appropriate points in the mechanism.

TORQUE, RPM, AND POWER DISTRIBUTION IN DIFFERENTIALS

I would like some clarification on the issue of torque distribution between the front and rear axles on 4wd vehicles. I find the matter fairly easy to understand when you have wheels spinning, and a limited-slip differential, but I find it more confusing when I read statements that a vehicle has a permanent torque distribution of, say, 32% front and 68% rear.

To me, torque and revolutions go hand in hand: reduce rpm and you increase torque, as in a ring and pinion. Doesn't that mean that if you want different torque at the front and rear axles, they have to turn at different speeds?

I know that in vehicles with viscous coupling drive to one axle, one can have a different overall drive ratio at each end, and this is often deliberately employed just to load the system in normal driving, and make it respond quicker to traction loss. But how does a rigid system, with a planetary differential for example, split torque unequally?

When we are dealing with one input torque, from one gear or shaft, and one output torque on a single shaft or other member, the relationship you describe between torque and speed does hold. Neglecting friction, power in equals power out. If rpm is changed, torque must change too, in inverse proportion, for the product of the two (power) to remain constant.

However, when the output power is divided between two shafts by a differential, things change a bit. Total power in still equals total power out (again neglecting friction), but power at each of the two output shafts is not necessarily equal to power at the other shaft. Any non-locking differential maintains a fixed distribution of torque between the two output shafts, while letting their relative speeds vary freely. In a conventional differential, the torque split is 50/50. In a planetary differential with one planetary gearset, the torque split is unequal but still fixed, while the shafts can turn at different speeds.

Usually the differential carrier or planet carrier is driven by a gear, which receives power from another gear driven by the input shaft. At the carrier, the simple inverse relationship between speed and torque applies. Torque at the carrier is input torque times rpm reduction factor. The sum of the output torques equals the carrier torque. The average of the output speeds equals the carrier speed. Power at each individual output shaft can be any value at all. It is even possible to have negative power (retardation) at one output shaft if that shaft is being forced to turn backward (opposite to torque). But the sum of the two power outputs must equal the power input. (That's the sum of their signed values, not their absolute values.)

It is helpful to think of each spider or planet gear as being similar to a beam, with a load applied at its midpoint, and reaction or support forces at two points equidistant from the load. The load is the drive force applied at the spider or planet gear's shaft. The reaction forces are the output shaft resistances to vehicle motion, acting at the points of mesh between spider and side gears, or between planet and sun and planet and annulus. Since the spider or planet shaft is always at the gear's center, the forces at the mesh points are always equal. This is true regardless of the rotational speeds of the various elements.

In a conventional differential, the side gears are equal diameter, so the equal forces at the mesh points act on equal moment arms, and produce equal torques. In a planetary, the annulus is larger than the sun, so the output torque at the annulus is greater than the output torque at the sun. The ratio of the output torques is the ratio of the pitch diameters of the annulus and sun. So the bigger the planet gears are in comparison to the sun, the more unequal the torque split becomes. Usually, the annulus drives the rear axle and the sun drives the front axle.

We can, in fact, regard the conventional differential as a unique version of the planetary, cleverly reconfigured by the use of bevel gears to allow the sun and annulus to be the same size.

All of this determines the torques at the front and rear drive shafts. Usually, the main rpm reduction and torque multiplication (after the transmission) happens at the axle, not at the transfer case. It is possible to use different ring and pinion ratios at the front and rear axles, and/or different tire sizes front and rear, and further alter the drive force distribution at the tire contact patches. At the axles, the usual rpm/torque inverse proportionality applies. To get more front torque and less rear by using dissimilar axle ratios, the front drive shaft must turn faster than the rear. That will increase wear at the center diff, rather like traveling a long distance with unequal size tires on an axle. Actually, the least wear at the center diff comes with slightly less torque multiplication at the front axle than at the rear – say a 4.10:1 ring and pinion at the front and a 4.11 at the rear. This is because even on a straight road, the car doesn't quite go perfectly straight, and in most turns the front wheels will track outside the rears. Consequently, the front wheels travel a few more revolutions per mile more than the rears, even if the effective radii of the tires are equal.

A spool or completely locked differential drives both output shafts at the same rpm, and does not split the torque in any fixed proportion. This is opposite to an open differential, which controls relative torque at the output shafts but not relative speed. With a spool, torque distribution depends on relative resistance at the two output shafts. It is quite possible for one output shaft to have negative resistance (wheel dragging and trying to drive the axle), while the other output shaft has a torque greater than the sum of the two (wheel driving the car plus overcoming drag from the other wheel). The former condition exists on the outside wheel, and the latter on the inside wheel, when making a turn with a spool and no tire stagger.

A partially locking or limited-slip differential is midway between. It allows some difference in speed, but adds torque to the slower output shaft and takes that torque from the faster output shaft.

A viscous coupling transmits torque according to the amount of slippage at the coupling. The faster the input shaft turns relative to the output, the greater the torque at the output shaft. Unlike a gear set, however, the relationship is usually not a simple linear function of the rpm ratio.

Note that none of these alternatives split power equally. No known passive mechanical device does that.