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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.

FINER POINTS OF ANTI-ROLL BARS ON STOCK CARS

I race primarily on two ovals, one a 1/3 mile with 7 degree banking and short straights, and one a 5/8 mile with 15 degree banking and long straights. I am not clear on some of the finer points of setup and adjustment of front anti-roll bars for a specific track, especially for a car running at two tracks such as these.

Most cars in my area run a solid link to the anti-roll bar on the right, and a slider link on the left, where the left front A frame in droop would cause tension on the link, but the left front in bump would remove any tension. I have run many cars with this setup, and a few with solid links on both sides. I don't really understand why there would be a preference for one over the other, or in what situation the solid links on both sides would be preferred over the slider setup.

I have found from home-made data acquisition that once a car is up to speed on the 1/3 mile oval, the left side never returns to ride height, as the car always has considerable g's throughout the lap. Many competitors run play in their anti-roll bars, as much as ¼ to 5/8". While it is easier to duplicate a setting if there is a specific gap to measure rather than a preload when making last-minute changes in tire size or wedge adjustments, it seems to me it might be better if there is less spring in the front of the car and less or zero gap in the bar.

For most bar play settings, the gap is gone as soon as the car comes up to speed, and the gap doesn't return until the cool-down lap at the 1/3 mile oval. Running play in the bar at a track with long straights, where the chassis returns to zero roll angle on the straights, seems like it would create a variable roll rate at the front, as the initial roll resistance would be only on the RF spring. Then the anti-roll bar and the resistance from the extension of the LF spring would kick in, greatly increasing the front roll resistance.

The car running the 7 degree banked turn would be in true roll, with the right frame rail down and the left rail up, while at the 15 degree track, the right rail would be down, but the left rail would be at approximately static ride height in the turn. It would seem that the bar would be half as stiff dynamically for the same right side travel at this track.

Another concern I have is that I'm not always sure in which situation it would be best to change the bar rate as opposed to the front spring rate. I realize the bar controls only roll, but in many situations where a rate change is desired, I have a hard time deciding which would be best, bar or spring.

Concerning small adjustments in bar play made at the track, consider a situation where a car running at the 1/3 mile, 7 degree track has a bar rated at 800 lbs. per inch of movement at the right front, at the location where the bar link is mounted, and ½" of static play. Suppose an adjustment is made to the bar reducing play to 3/8", or 1/8" less than before. This would presumably increase roll resistance by 100 lbs. For a particular right front travel, but would not increase the roll ***rate****. How would this compare to changing to a bar that rates at 900 lb/in, with the original ½" play retained?*

It has long been common on oval track stock cars to use some type of "soft link" on the front anti-roll bar. These may take the form of a slider, a chain, or a pad on the anti-roll bar arm that bears against a pad on the lower control arm. The last of these is the form commonly seen in the upper divisions of stock car racing. Chains and sliders are common in "street stock" type cars, where the suspension components actually come from production passenger cars.

All these variations do the same thing: they create a connection that transmits force in only one direction. The anti-roll bar resists rightward roll only. The bar may be preloaded, in which case it will resist leftward roll up to the point where the preload is relieved. (Some sanctioning bodies prohibit preloading the bar. The reason for this is a mystery to me.)

The reason the soft link is there at all is to make the front suspension more compliant when the driver gets the left front wheel on the flat apron of a banked track. In this situation, the car tends to go loose (develop oversteer) and spin. This is partly because of the leftward yaw moment created when the left front encounters increased resistance, and partly because the car de-wedges (load increases on the left front and right rear, and decreases on the right front and left rear). If the suspension is more compliant, both effects are reduced, especially the de-wedging.

However, even in this situation the front suspension only acts softer beyond the point where the left front suspension compresses enough to put slack in the bar. If the bar has no play and no preload, the front suspension must go into leftward roll (left front compressed more than right front) before the soft link has any effect. And the soft link will make the car looser if the driver needs to turn right to avoid a wreck. So the soft link is a mixed blessing.

Wheel rates and arm end rates of anti-roll bars are a confusing subject for many. The main cause of this confusion is that there is no agreed convention as to what an inch of arm end or wheel motion means. Does it mean an inch of motion at just one wheel? An inch of motion in opposite directions at both wheels? An inch of difference between the two wheels?

I find it convenient to express the rates of all springing devices, including interconnective ones such as anti-roll bars, in terms of pounds per inch per wheel. This agrees with the way we always express the rate of non-interconnective springing devices, i.e. the main springs or ride springs. We then have numbers that easily tell us what change of anti-roll bar rate equates to what change of spring rate.

However, it has become customary to rate anti-roll bars by testing them in devices originally intended for testing non-interconnective torsion bars. The bar is placed in a fixture that holds one end of the bar stationary, and the arm on the other end of the bar is moved one inch. The force required to do this is measured, and that is taken as the bar's rate in pounds per inch.

This isn't wrong. But it can create confusion when we try to translate it into pounds per inch per wheel, because what we have from the test is pounds per inch per wheel *pair*. An inch of roll per wheel pair is only half an inch of oppositional motion per wheel. Therefore the number from the testing machine is only half the bar's equivalency to wheel springs, if the bar and the springs acted at equal motion ratios.

On the other hand, it is common for anti-roll bar manufacturers to express the rate of their bars the way I like to see it, as pounds per inch per arm end, not pounds per inch per arm end pair. This sometimes leads to acrimonious exchanges between bar manufacturers and customers who test a bar they just bought and find that it "rates" at only half the advertised value.

Often, we encounter situations where we do not have "pure" roll motion. Pure roll, for a front or rear wheel pair, would mean equal amounts of suspension motion at each wheel, in opposite directions. Pure ride would mean equal amounts of motion at each wheel, in the same direction. Ride and roll are the two modes of motion for a front or rear wheel pair. Any possible motion of that wheel pair can be resolved into some amount of ride and some amount of roll.

Applying this to the questioner's example of a situation where a car in a banked left turn rolls purely by compressing the right front suspension, and the left front neither compresses nor extends, that is a condition of equal ride and roll. If the right front compresses an inch and the left front doesn't move, we have half an inch per wheel of rightward roll, plus half an inch per wheel of compressive ride. On the left wheel, the effects are subtractive and exactly cancel. On the right wheel, the effects are additive, and we have twice the half-inch per mode, or one inch of compression.

If we compare this to a pure roll situation (zero ride motion) which creates an inch of compression at the right front, yes, the bar creates twice as much force. But the bar's rate is the same. The difference is that there is twice as much roll. If the roll were equal to the previous example, the right front compression would be only half an inch, the difference between right front and left front would still be an inch, and the force generated by the bar would be the same as in the first example.

Once we learn to separate the ride and roll components of the motion, and remember that the bar acts only in response to the roll component, things get much simpler.

Regarding how much of the front wheel rate in roll should come from the springs and how much from the bar, it mainly comes down to how much we want the front of the car to drop in the turns due to the banking. This in turn relates to both suspension geometry and aerodynamics.

It is normal in all stock car racing classes to have a minimum static ground clearance requirement. The car has to pass over a barrier of a certain height to get through tech inspection. Yet we would like the valance to run just off the track, and have as much forward rake in the car as possible, through the turns. This gets us the greatest aerodynamic downforce available, within the bodywork and ground clearance rules.

Additionally, with passenger car front suspension, we usually have insufficient camber recovery in roll for racing. The control arms are close to parallel at static, causing the wheel to change camber very little in ride, but a lot in roll. This is good for tire wear in gentle driving, but not good for hard cornering. In some cases, the rules will allow us to use extended upper ball joints to improve the geometry for racing. Sometimes this may be enough, but in most cases the geometry will benefit if we run the car lower. To do this, within the ride height rule, we need to soften the springs and stiffen the bar.

In the upper divisions of stock car racing, we can get as much camber recovery as we want, so only the aerodynamic factor argues for soft front springs. In general, with soft front springs and a big bar, we'll want the front view instant centers a bit higher and further from the wheels at static than we'd want with stiffer springs and a softer bar.

In either case, a flatter track calls for more of a soft-spring/big-bar approach than a steeply-banked track.

Regarding rate of a bar with slack in it: the bar has a rate of zero until the slack takes up. Then it has the same rate it would have without slack.

We can speak of the bar's rate as an instantaneous value, at any given point in roll travel, or we may speak of its average rate, over a specified interval of travel. When there is slack in the system for a portion of the interval, but not at the end of the interval, the bar has the same instantaneous rate at the end of the interval whether there is slack or not, but it has less average rate over the interval if there is slack.

Relating this to the questioner's example comparing an 800 lb/in rate with 3/8" of slack (let's call this Case A) to a 900 lb/in rate with ½" of slack (Case B), we may say that these two systems have the same instantaneous rate (namely zero) at any displacement less than 3/8"; the same average rate (450 lb/in) over the displacement interval 0" to 1"; the same force (zero) over the displacement interval 0" to 3/8"; and the same force (450 lb.) at 1" displacement.

Case A has less rate than Case B when the slack is absent; more force at displacements between 3/8" and 1"; and less force at displacements greater than 1". At ½" displacement, Case A generates 100 lb. more force than Case B. From there, the forces converge to a point of equality at 1", then diverge again, with Case B making more force, beyond 1" displacement. At 1½" displacement, Case B generates 100 lb. more than Case A.

In terms of car behavior, if all other setup parameters are equal we would expect Case A to have more understeer (or less oversteer) from 3/8" displacement to slightly less than 1" displacement; both setups to have similar understeer around 1" displacement; and Case B to have more understeer at greater displacements. We would expect Case A to have a lesser tendency to get looser (or a greater tendency to get tighter) as grip diminishes.