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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.

SHOCKS WANTED FOR RESEARCH

In previous newsletters, notably August 2000 and December 2001, I have discussed acceleration sensitivity of shock absorbers. (Acceleration is the rate and direction of change in velocity.) I noted that just looking at the difference between the extended end of the stroke and the compressed end of the stroke in a standard sinusoidal shock dyno test will give you a crude indication of a damper's acceleration sensitivity. If the two ends of the stroke look substantially different, that suggests a high degree of acceleration sensitivity.

It is reasonable to suppose that differences in acceleration sensitivity are a big part of the reason why shocks that generate similar traces in the most common dyno test (sinusoidal motion produced by a crank, 2" stroke, 100 rpm) can act so different on a car.

I have for some time been interested in investigating the matter more systematically. I would like to come up with test procedures that will give us a way to measure and quantify sensitivity to acceleration, and also investigate the importance of jerk sensitivity. (Jerk is the rate and direction of change in acceleration.)

It now looks like I'll get the chance. At least one, and quite possibly two, shock dyno manufacturers are interested in working with me on this. At least one up and coming damper manufacturer has expressed interest in building shocks for test. At least one racing team has expressed interest in working with us. That's enough to get started.

What I'm looking for now is additional teams actually running cars, who are interested in collaborating on this. As things stand, teams won't pay us, and we won't pay them. Teams will furnish shocks they already run, and/or experimental ones, for dyno testing, and provide us with feedback regarding how the various shocks affect their car. We are particularly interested in obtaining shocks that are reported to dyno similarly but act different. The team gets free shock dyno testing and a better understanding of how shocks work. We get free shocks to test, and a better understanding of how shocks work. At some point, perhaps I will write a feature article in *Racecar Engineering*, and get some publicity for the consulting business. Teams interested in being on the inside of this cutting-edge effort are invited to contact me at the address, e-mail, or phone at the top of this page.

TIRE WARMERS FOUND

Back when the forum on www.RacecarTech.com was running, somebody asked me where to get tire warmers for their dirt Late Model. At that time, I pursued several leads, but they all ultimately came up dry.

Finally, at the PRI show last month in Indianapolis, I found a US source. They are Chicken Hawk Racing, at 249 Hapeman Hill Rd., Red Hook, NY 12571. Their phone is 866-HOT-TIRE (468-8473). They have a website at www.chickenhawkracing.com.

For those unfamiliar with tire warmers, they are basically high-temperature electric blankets that wrap around a tire's tread and heat it up. Apart from the obvious advantage of giving you sticky rubber right from the green, tire warmers also allow you to heat and cool the tires gently, and keep them hot between runs. This reduces the effects of heat cycling, keeping the rubber soft longer. Additionally, they allow you to set your "cold" pressure at a controlled temperature, rather than ambient. This temperature can be high enough to assure that the tire won't have any significant liquid water in it. Regular readers may recall that water in a tire does not cause any unusual pressure rise if it's in the vapor state when you set the pressure.

Why wouldn't you use them? First of all, many sanctioning bodies and tracks have outlawed them as a cost-containment measure. And they aren't cheap. Chicken Hawk sells two models, one for around $1500 and one around $2000. That's each, and you need at least four for a car (they make them for motorcycles too). The less expensive model has a pre-set thermostat, ordinarily 175deg F (80deg C). The more expensive model has an adjustable thermostat, and a digital thermometer so you can see if the tire's up to temperature yet.

Whether the performance gain is worth the money depends on your personal situation, but the performance gain is real.

SPRING PLACEMENT ON TRIANGULATED 4-LINK

I have a question on rear spring placement on a stock metric 4-link suspension. I have built several chassis and have been mounting the rear spring centerline forward 2½ inches of the centerline of the
axle. I've started on a new chassis and thought I would go back to mounting the spring directly on the axle centerline. Since the housing does not rotate under power I don't feel I'm gaining anything. Does mounting the spring forward of centerline affect the static rear percentage or in any way change the motion ratio of the spring?

US oval track racers will need no introduction to this type of suspension. For readers unfamiliar with it, this is what is sometimes called a triangulated 4-link, or Chevelle-style 4-link. It has been used on various GM cars, including the “metric” series referred to here, and also recent Mustangs. It is illustrated on p.648 of Milliken and on p.260 of Gillespie. It uses four angled trailing links to locate a beam axle both longitudinally and laterally, with no Panhard bar, Watt linkage, or other purely lateral locating device.

In most such layouts, the side-view geometry gives a substantial amount of anti-squat. The axle does rotate with ride motion, nose-down in bump and nose-up in droop. However, the only rotational *compliance* with drive torque comes from flexure of the parts, mainly the bushings. When the questioner here says the axle does not rotate, he means that there is no highly compliant torque absorbing device such as a torque arm or pull bar incorporating a spring.

In roll, there is little or no axle housing rotation.

The location of the springs has no effect on static rear percentage, except that the mass of the springs is positioned slightly further forward or back. Spring location fore-and-aft does affect motion ratio a little bit in ride. Moving the spring forward makes the spring-to-wheel motion ratio slightly less than 1:1 in ride. In roll, the motion ratio is the same as it would otherwise be, assuming the lateral spring spacing is unchanged. Note that this motion ratio in roll is always less than 1:1 for any beam axle, which means that any beam axle without an anti-roll bar has a substantially softer wheel rate in roll than in ride.

So on a stock metric suspension, moving the springs forward softens the wheel rate in ride somewhat, without softening it appreciably in roll. This makes the ride and roll wheel rates less unequal. However, if the spring is moved forward only 2½ inches, that will have only a small effect.

Note that we are speaking here of springs (on buckets, on coilovers, or on sliders) mounting directly to the axle, not to a link or a birdcage.

Even in cars with compliant torque arms or pull bars, mounting the springs forward of the axle does not add a lot of rear jacking, and rear jacking only adds total rear wheel loading due to the overall vehicle CG being slightly higher when accelerating forward. Such effects tend to be small.

Remember that jacking up both rear corners does not increase rear percentage, in and of itself. Remember also that jacking one rear corner up more than the other also doesn’t significantly change rear percentage, but it does change diagonal percentage.

Correspondingly, fairly significant effects in torque-compliant axles can result when the fore-and-aft spring offset differs on the right and left, as when the left spring is ahead and the right spring is behind. Then there can be a meaningful change in instantaneous diagonal percentage as power is applied. This in turn will affect the car’s cornering balance under power.

ROLL CENTER WITH A J-BAR

Many books, forum posts, and websites go into great detail on on the front roll center and only touch on the rear. I run an IMCA modified with a j-bar [short, off-center Panhard bar, bent into a J shape to clear the pinion snout – usually mounted to the left side of the frame and the right side of the pinion snout, with the left pivot somewhat higher than the right]. I would like to determine where my rear roll center is.

This is actually a fairly complex question. First let's discuss **what** a roll center is, and isn't.

A roll center isn't a real thing. It's a modeling construct – an invented idea that helps us think and talk about the suspension's behavior. It's a way of representing the geometric roll resistance of a front or rear wheel-pair suspension system, to simplify prediction of wheel loads when cornering. In the simplest method of modeling wheel load changes due to lateral acceleration, the suspension is imagined as a beam axle (which yours actually is), and the roll center describes a **height** at which **lateral** force is transmitted between the axle and the sprung mass.

It is vital to recognize that we are not talking about a point the car actually rotates around, or a point whose lateral location determines how vertical forces react. The roll center is best thought of as a point in a side view of the car, that has no defined lateral location at all, or perhaps as a point in the same longitudinal plane as the sprung mass CG. In other words, we should imagine the roll center as the height of a pin in a vertical slot, or the height of a horizontal Panhard bar, not as a pin joint. It is a notional device that transmits **horizontal force only**.

Okay, now with a Panhard bar that's curved, offset, and inclined, how do we assign that imaginary point to get the best wheel load prediction? There are two answers to this, depending on how much work we want to do, and how accurate we want our model to be. In both methods, we disregard the bend in the bar, and think of it as a straight link connecting its two pivot points.

In the simpler method, we find the point where the centerline of this imaginary straight bar intersects the longitudinal CG plane, and take that point's height as the roll center height. With this method, we disregard effects due to the off-center, inclined Panhard bar jacking the rear of the car up or down.

In the more rigorous method, we take the midpoint height of the imaginary straight bar as the roll center height. We then must also take into account the vertical forces resulting from bar inclination. We likewise consider these as acting at the bar's midpoint. When the left pivot is higher than the right pivot, in a left turn the jacking force tries to raise the sprung mass. When the bar centerpoint is left of the sprung mass CG, this effect tries to roll the car rightward, reducing effective roll resistance at the rear. So we have a higher roll center, suggesting more roll resistance, but also a pro-roll moment from the jacking. Net result will be similar to the load transfer predicted from the lower roll center in the simpler method, though not exactly the same. (As described here, both methods have some inaccuracy due to the bar being forward of the axle. Correction for this is possible, but beyond our scope here.)