

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.

April/May/June 2003

My apologies to anybody inconvenienced by the lateness of the April and May issues, and my thanks to those who have kept me so busy lately, especially those who have done so on a paid basis!

WELCOME

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SHOCK AND SPRING FORCES

With the increased emphasis on tuning using shocks, could you explain how the shock absorber forces are fed through the chassis to affect the tire loads? How do these forces differ from the forces that are transmitted through the springs and sway bars?

To discuss any subject, we need a vocabulary. So first, let's define some terms.

Car wheels move in three dimensions, but we can simplify and think of the suspension as mainly just letting the wheel go up and down. Viewed this way, the suspension for a particular wheel can move in two directions: compression and extension. We also sometimes use the terms bump and droop or rebound for these. As with any motion, the system can be said to have a position, a velocity, an acceleration, and a jerk at any instant we may choose to look at.

Position, or displacement, is the inches (or millimeters) of compression or extension from some designated reference or zero point. Usually, we take the static position as this zero – the position the suspension is in when we set the car up on the scales.

Note that suspension position or displacement (for one wheel) can be expressed as a single number. (Position or displacement of the sprung mass requires six numbers to completely express it: three for linear position along three axes, and three for angular position about those axes. Rotation about a longitudinal axis is roll; rotation about a transverse axis is pitch; rotation about a vertical axis is yaw.)

The suspension's position may be one-dimensional, but it is still a *vector* quantity: it has a magnitude, and a direction – so many inches or millimeters, compression or extension.

The position or displacement can change over time. This change of position with respect to time is called *velocity*. It likewise has a magnitude and a direction – so many inches or millimeters per second, compression or extension.

The velocity can change over time. The change of velocity with respect to time is called *acceleration*. Acceleration has a magnitude and a direction – so many inches or millimeters per second per second (in/sec^2 or mm/sec^2), compression or extension.

The acceleration can change over time. The change of acceleration with respect to time is called *jerk*. Once again, this is a vector quantity – so many inches or millimeters per second per second per second (in/sec^3 or mm/sec^3), compression or extension.

Readers who've had calculus will recognize that we are taking a series of derivatives here. Velocity is the first derivative of displacement, with respect to time. Acceleration is the second derivative of displacement and the first derivative of velocity. Jerk is the third derivative of displacement, the second derivative of velocity, and the first derivative of acceleration. We could go on taking derivatives indefinitely (the next one is called *quirk* – no, I'm not making this up), but the usefulness of doing so is doubtful.

It definitely is useful to look at longitudinal and lateral acceleration and jerk. As we will see, the car's accelerations determine the suspension's displacements, and the changes in the car's accelerations (its jerk values) determine the individual suspensions' velocities. And since the suspensions' velocities are not constant, the suspensions likewise have non-zero acceleration and jerk values.

Those with engineering backgrounds may feel I'm belaboring the obvious in this discussion. However, I have recently been recruiting volunteers to collaborate in a project to test sensitivity of dampers to suspension acceleration and jerk, and I have found that many readers and clients have not understood what I meant by acceleration or jerk as applied to a shock or a suspension system.

We can express the direction of any of these quantities with a mathematical sign – positive or negative. Which way should we do this? Should positive be compression or extension? It is customary to use positive for compression displacement in data acquisition, so maybe that's the way to go. Calling compression positive also more or less agrees with the conventional tire axis system, in which normal or road-vertical force is considered positive. In general, we think of increased suspension compression displacement and increased normal force as going together, although this is not always so. Then again, compression displacement generally implies extension force within the suspension, so if extension is positive, then positive force in the suspension corresponds to positive force at the tire, at least as a crude generalization.

In shock dynamometer testing, the dyno manufacturers have to establish sign conventions within their own software. In the software for Roehrig dynos, the most popular make in the US, compression strokes have positive force (that's extension force, resisting the compression motion) and negative velocity. This agrees with the extension-positive reasoning above. However, it disagrees with the conventions generally used in data acquisition. No matter what we do, we will either disagree with the shock testing convention or the data acquisition one, since they disagree with each other.

If this were not confusing enough, even these sign conventions are not universal, as we will see shortly.

One might think we could avoid all confusion by dispensing with signs and simply stating direction with the word “compression” or “extension”. This works fairly well when we are talking about displacement, velocity, acceleration, and jerk direction. However, some confusion arises when we discuss damping force. It is customary to speak of compression (or bump) damping force as the force occurring when the shock is compressing, and extension (or rebound) damping force as the force occurring when the shock is extending. But in fact these forces ordinarily are opposite in direction to the shock's velocity: compression damping force acts in the extension direction; extension damping force acts in the compression direction. So substituting words for mathematical symbols is no refuge. Either way, we have to keep in mind the actual physical phenomena we're trying to describe, and apply some common sense, to avoid confusion.

When a number or quantity has a positive or negative sign, we may speak of its *absolute value*. A quantity's absolute value is the greater of the quantity and its opposite. The absolute value of 4 is 4. The absolute value of -4 is 4. ($|4| = 4$; $|-4| = 4$)

Correspondingly, when we have a unidimensional vector quantity such as suspension displacement, velocity, acceleration, or jerk, whose direction can be expressed by a positive or negative sign, we may speak of the quantity's absolute value. This means the quantity's magnitude, irrespective of direction. So, for example, when we speak of large **absolute velocities**, that means large compression or extension velocities. When we speak of large **velocities**, on the other hand, that means large velocities in whatever direction we call positive.

In casual conversation, these distinctions are often disregarded, so again we face the need to apply common sense, and understand people's words (and also their math symbols) in context.

A familiar synonym for absolute velocity is *speed*.

The most common type of shock dyno plot is force (vertical axis of the graph, forces resisting compression positive, forces resisting extension negative) versus absolute velocity (horizontal axis, all values positive). Also available is force versus velocity. Here it is customary to show velocity as negative in compression, and force resisting compression as positive. The trace is generally S-shaped, and lies mainly in the second and fourth quadrants. These sign conventions are opposite to the compression-positive convention used in data acquisition, but they do show a realization that the velocities and the most common forces should have opposite signs.

That's with most dyno software I've seen. A correspondent in Australia recently sent me force-versus-velocity traces from an SPA dyno, in which the compression stroke has both velocity and force positive, and the extension stroke has both velocity and force negative. The trace then lies mainly in the first and third quadrants of the graph. With this choice of sign convention, velocity

agrees with the data acquisition, but forces acting opposite to velocity are shown with the same sign as velocity.

Note that I refer to the **most common** forces associated with a particular velocity direction, rather than all the forces. A true damping force acts in opposition to motion – otherwise it wouldn't suppress motion. However, not all the forces our dampers generate are actually damping forces in this sense.

If you examine shock dyno plots, you will see that sometimes shocks generate forces in the same direction as velocity. There are at least three known phenomena at work here, and perhaps additional ones. The first known phenomenon is gas spring effect. In gas pressure shocks, the gas compartment acts as a rising-rate spring. The smaller the gas volume, and the higher the pressure, the greater the gas spring rate. The gas spring force always acts in the extension direction. So when the shock is moving very slowly in extension, it exerts a net extension force.

If we look at a force vs. absolute velocity plot of the full stroke from a crank dyno, there will be two *noses* or points at zero velocity, representing the extended and compressed ends of the stroke. In a gas-pressure shock, the compressed end of the stroke will show a higher force reading (meaning more extension force) than the extended end. If the dyno is cycled very slowly, and the shock has

very soft low-speed valving – especially if it has bidirectional bleed – the difference between the two noses will be almost entirely from gas spring effect.

It is customary to zero the dyno, and omit gas spring force from the force reading, at some point in the cycle – typically the extended end of the stroke, although mid-stroke and full-compression

zeroing are also common. Even when this is done, the force reading will be higher at full compression than at full extension. Thus, the shock will either show an extension force early in the

extension stroke, or a compression force early in the compression stroke, or both, just from gas spring effect.

There is a second known effect that will cause the noses to spread further apart as low-speed valving is stiffened, and as the shock is cycled at higher frequencies or longer strokes. This effect is fluid compressibility.

Suppose we have a shock with the body sprung, mounted body-up. As the shock nears the end of the extension stroke, the fluid below the piston is under substantially greater pressure than the fluid above the piston. If it is not allowed to bleed off very rapidly, it will still be under pressure as the piston comes to rest and starts to move upward. Consequently, the shock will not resist compression until it is some distance into the compression stroke. This effect is sometimes called *lag*. If the fluid were perfectly incompressible, this couldn't happen. Pressure would equalize instantaneously as soon as velocity reached zero. But shock fluid has substantial compressibility, despite our efforts to reduce this.

Thus, the fluid itself will act partially as a spring rather than a damping medium. And until there is greater pressure above the piston than below it, there will be no flow downward through the piston, and therefore there will be no extension force damping the compression. Indeed, as long as pressure is greater below the piston, fluid will try to flow upward through the piston.

Lag occurs at both ends of the stroke.

Lag is somewhat distinct from acceleration sensitivity, but it does relate to accelerations, especially when the velocity is changing sign.

A third known effect comes from the masses and inertias of the valving elements. With deflective-disc valving, these effects are generally small. Where a valving element of considerable mass acts against a spring, the effects can be large. Shocks made by Ricor and sold under the Edelbrock name

make deliberate use of this effect and advertise it as a selling point. These shocks use a weighted element on top of the piston, which softens the valving when acceleration is in the extension direction, i.e. during the more compressed half of the stroke, or the

compression closing/rebound opening (cc/ro) portion. I have also seen a patent description for a shock with a similar weighted element under the piston, to soften the rebound closing/compression opening (rc/co) portion.

Intentional acceleration sensitivity usually reduces the forces generated by the shock – although it could be made to increase them – when acceleration is in a particular direction, and sometimes only when a particular combination of acceleration and velocity directions is present. Since lag is related to valving stiffness when the velocity is changing sign, acceleration sensitivity affects lag.

Acceleration sensitivity affects force whenever relevant accelerations are present, not just near velocity reversals. Thus, although acceleration sensitivity affects lag, it is a distinct phenomenon.

Acceleration sensitivity is not necessarily bad for car behavior, and may in fact be beneficial when intelligently applied, but it's a complication in terms of modeling or understanding.

So shock forces are complex. Sometimes our dampers create spring forces. Sometimes we can't predict their behavior just by knowing their velocity.

Spring forces can also be complex. In leaf springs especially, there is damping in the spring, mainly from inter-leaf friction. In some large vehicles, with many leaves in the springs, this effect provides all the damping; there are no shocks. Even coils and torsion bars have some internal hysteresis. They will heat up as they flex, and they will come to rest after a number of oscillations, even in the absence of external damping.

There is also friction in all the pivots in the suspension and steering, and there is friction in the sliding contacts in the shocks.

So we get some damping forces from our springs, and from other components in the system.

I am not trying to confuse matters here. I merely wish to point out that the remarks which follow are based on simplifying assumptions, rather than comprehensive models of spring and shock behavior.

In understanding how shocks and springs affect wheel loads, we think of springs as being exclusively sensitive to position or displacement. We estimate their forces on the basis of their displacement. We think of shocks as being entirely velocity-sensitive. We assume that they always make compression forces when they are extending, and extension forces when they are compressing. We assume that if the shaft speed is greater, the absolute force is greater, in some predictable relationship, though usually not a linear one.

An anti-roll bar is an interconnective spring. It generates forces based on its displacement, but its displacement depends on the relative displacement of the two wheels it connects, rather than their individual displacements. It generates equal and opposite forces in the two suspensions it connects.

It is useful to divide suspension forces affecting wheel loads into the forces present at static condition (as the car stands on the scales at the conclusion of static setup) and the forces that add to or subtract from these static forces as the car runs. At static condition, all suspension displacements from static are zero, suspension velocities are all zero, suspension accelerations are zero, and suspension jerks are zero. The anti-roll bar or bars may have preload or may not. In a road racing car they usually will not.

The springs, shocks, and anti-roll bars act on the suspension **in parallel**. Although these elements may act through different motion ratios, each of them can be thought of as exerting a particular force at the wheel at any given instant, and these forces can simply be summed (with proper attention to sign) to arrive at the resultant effect. A 400 pound extension force from the spring, with no force from the a/r bar and the shock, is equivalent to a 600 pound extension force from the spring, countered by a 100 pound compression force from the a/r bar and a 100 pound compression force from the shock (all as measured at the wheel). Or, either of these is equivalent to a 200 pound extension force from the spring, and a 200 pound extension force from the shock, and no a/r bar present. The tire doesn't know the difference. It only responds to the total force spreading the wheel away from the sprung mass.

With springs and a/r bars, we have a *wheel rate*. That's the rate of the spring, or the bar at the lever arm end, times the **square** of the spring-to-wheel, or arm-end-to-wheel, motion ratio. The wheel rate defines a simple relationship between force and displacement. Using the wheel rate, we can calculate the spring and bar forces at the wheel when we know displacement at the wheel.

A shock doesn't have a wheel rate in the sense that a spring does, because it is not a displacement-sensitive device. To find shock force at the wheel, we need to know either velocity at the wheel, or velocity at the shock. If we are working from data acquisition outputs, often the sensor will be set up to read shock motion one-to-one, or as nearly so as practicable. If we are calculating from an assumed or predicted suspension motion, or from photographic data, we may be working from wheel motion. To calculate shock force at the wheel from velocity at the shock, we first estimate the force

the shock generates at that velocity, based on dyno testing, then multiply this by the **first power** – not the square – of the shock-to-wheel motion ratio. To calculate shock force at the wheel from velocity at the wheel, we first find shock velocity by multiplying wheel velocity by the shock-to-wheel motion ratio, then proceed as before: estimate shock force, multiply by motion ratio. So we do multiply by the motion ratio two times in this process, but in between, we estimate the shock force.

A complete dissertation on all possible wheel loading effects from springs and dampers is beyond our scope here, but let's consider a simple case: a turn on a level, smooth surface. We will assume that the road has no small-scale or large-scale irregularities – billiard-table flat, no hills, no crests, no dips, no banking. We will also ignore aerodynamic effects. This means that the sum of our four wheel loads is the same as we'd see in the shop while doing our setup on the scales. It also means that any change in the distribution of those wheel loads is entirely the result of the way the suspension transfers weight or wheel loading in response to horizontal forces generated by the tires. This lets us isolate these effects and look at them.

We will also assume that the suspension generates no extension or compression forces due to linkage geometry: no anti-roll or pro-roll, no anti-dive or pro-dive, no anti-squat or pro-squat, no anti-lift or pro-lift. This is actually impossible to achieve for all conditions of suspension motion, and it wouldn't be desirable, but we can imagine it, and it is not too far from the actual properties of current four-wheel-independent chassis. This simplifying assumption lets us focus on wheel load changes from the springs and dampers.

As when the car is on the scales, **an increase in positive (meaning extension) force at one corner of the car adds wheel loading at that corner and the diagonally opposite one, and correspondingly reduces loading on the other two corners of the car. And a negative (compression) force reduces loading at that corner and the diagonally opposite one, and adds load at the other two corners.** This is true regardless of whether the force is generated by the damper, the spring, or the a/r bar. The tire doesn't know which part does what. It only behaves according to the resultant loading generated by the suspension elements acting together.

Unlike the static condition, front, rear, left, and right percentages do change. However, the suspension does not control these changes in this simplified case; the wheelbase, track width, and CG height – not the suspension – control how much load transfers at a particular longitudinal and lateral acceleration. The springs and shocks control how the diagonal percentage varies as all this is going on, and thereby influence the car's cornering balance. More diagonal percentage (meaning outside front wheel load plus inside rear, as a percentage of total) at any point in the cornering process adds understeer, or tightens the car. Less diagonal percentage adds oversteer, or loosens the car.

The July 2001 newsletter contained a troubleshooting chart based on five parts of a turn, with complete explanations of what the five parts were. For the convenience of readers receiving the newsletter by e-mail, I am sending that back issue with this one, as a reference. Readers seeing this issue as hardcopy can order back issues from me.

Part One, or early entry – braking increasing while turning in: This may or may not happen at all. In oval track racing, it is very common. In road racing, braking force more commonly reaches its maximum while the car is still running straight.

The car as a whole is accelerating rearward at an increasing rate, and accelerating laterally in the direction of the turn at an increasing rate. Angularly, it is pitching forward and rolling out of the turn.

Its roll displacement is outward. Its pitch displacement is forward. Its roll velocity is outward. Its pitch velocity is forward.

Therefore, the outside front suspension has a compression displacement, and a compression velocity. The inside rear has an extension displacement, and an extension velocity. Without more information, it is hard to say exactly what the displacements and velocities at the inside front and outside rear are, but they are relatively small, because the effects of roll and pitch are subtractive at those corners.

Consequently, spring and damper changes at the outside front and inside rear will have the greatest and most certain effects on the car.

Taking springs first, the important principle is that a stiffer spring creates more load **change** with displacement change – not necessarily more load. So a stiffer outside front spring increases load at that corner (negative displacement, positive load change), and at the inside rear, and correspondingly unloads the inside front and outside rear. This adds diagonal percentage, which tightens the car, or adds understeer.

A stiffer spring on the inside rear creates a bigger load **decrease** with displacement change. That translates to less diagonal percentage, and a looser car (more oversteer or less understeer).

A stiffer front anti-roll bar creates a positive (extension) force at the outside front, and an equal and opposite negative (compression) force at the inside front. This also creates equal and opposite load changes at the rear – more load at the inside rear, less at the outside rear. Result: more diagonal percentage, tighter car (more understeer). A stiffer rear bar does the opposite, and loosens the car.

As for the dampers, if we stiffen the outside front low-speed compression valving, that adds a positive (extension) force at the outside front, adding diagonal percentage and tightening the car (adding understeer). If we stiffen the inside rear low-speed extension valving, we are creating a negative (compression) force at the inside rear. This reduces diagonal percentage and loosens the car.

Important things to note regarding the role of the dampers:

- 1) When the suspension velocity and the suspension displacement are in the same direction, stiffening the damper and stiffening the spring have qualitatively similar effects on oversteer/understeer balance.
- 2) Contrary to a very common misconception, stiffening the dampers does not slow down or momentarily reduce the load changes at the outside front or inside rear – these load changes are sped up, or are momentarily **increased**. Spring loads are momentarily decreased at the outside front and increased at the inside rear – in other words, spring

load changes are momentarily decreased by the shocks – but the effect on tire loads is the opposite.

3) If the low-speed valving is soft and the velocities are small, the damper forces may be relatively insignificant.

Also, note that:

1) We are assuming that the road is smooth. As long as this is true, the shock movements will be low-speed (less than 2 in/sec) and will be caused by sprung mass motion. When the surface is bumpy, bumps become the main factor in shock motion and none of what we've been saying about load transfer effects from the dampers applies. There still are sprung-mass-motion components to the shock motion, superimposed on the motions from the bumps. When looking at track data we can, at least to some degree, separate these components, but the shocks can't do that. They only know their actual motion at a particular instant.

2) We are assuming that the brake bias is such that the front wheels do at least half of the braking. If the car is slowed primarily by the rear wheels, the effects of diagonal percentage may reverse. This is due to the distribution of rearward force at the rear tires, and not to any fundamental difference in tire properties during entry.

3) Contrary to the contentions of some writers, tire load sensitivity (the decrease in coefficient of friction with increasing load, which is responsible for the car getting tighter with increasing diagonal percentage) does not reverse or work backwards during entry or with cold tires.

Part two, or late entry – braking decreasing, cornering force increasing: This may be the first or only phase of entry if the driver reaches peak braking while traveling straight. It is also possible for a period of “semi-steady-state” braking and cornering to exist between early (increasing) and late (decreasing) braking, particularly on ovals. More on this later.

In terms of spring and damper behavior, the difference between part one and part two is the direction of pitch velocity. In part one, the car has a forward pitch displacement and a forward pitch velocity. In part two, the car has a forward pitch displacement and rearward pitch velocity. We may say it's de-pitching; it has a forward tilt, but a decreasing one.

Roll displacement and velocity are both outward, same as in part one. However, roll displacement is increasing at a decreasing rate, which reaches zero at the conclusion of part two. So roll velocity is outward and decreasing, and roll acceleration is inward.

At the beginning of part two, the car has a combination of outward roll displacement and forward pitch displacement. At the conclusion of part two, the car has near-zero pitch displacement and increased outward roll displacement.

The biggest individual suspension displacement changes from the conclusion of part one, and the greatest individual suspension velocities, are at the inside front and outside rear.

These are the wheels where the effects of rearward pitch velocity and outward roll velocity are additive.

At the beginning of part two, the displacements of the inside front and outside rear will be small. At the conclusion of part two, the inside front will have an extension displacement, and the outside rear will have a compression displacement. The velocities, therefore, are in the extension direction at the inside front, and in the compression direction at the outside rear.

Consequently, the low-speed extension (rebound) valving on the inside front and the low-speed compression valving on the outside rear are in a position to influence wheel loads. Stiffening inside front extension introduces a negative (compression) force and increases diagonal percentage, tightening the car (adding understeer). Stiffening outside rear compression introduces a positive (extension) force, which decreases diagonal percentage and loosens the car (reduces understeer).

Inside front and outside rear spring rates will be of little importance at the beginning of part two, but will be as significant as outside front and inside rear rates at the conclusion of part two. Stiffening the inside front spring will tighten the car (add understeer), and stiffening the outside rear will loosen the car (reduce understeer). As in part one, velocities and displacements at the most influential wheels have the same sign, and stiffer springing and stiffer damping have qualitatively similar effects on balance.

Part three, or mid-turn – steady-state cornering: Most turns, with most drivers, will include some interval of approximately steady-state cornering. This means that the driver applies just enough power to maintain or slightly increase speed, and most of the tires' traction is used in the car-lateral direction. The car will be traveling in a nearly constant-radius path. In a street-intersection turn on a street circuit, this phase may be so brief as to be negligible. In a carrousel-type turn or long sweeper, or on a high-speed oval, the car may experience approximately steady-state cornering for as long as five seconds.

In this situation, the car has pitch and roll velocities very close to zero. Pitch displacement is also close to zero. Roll displacement is substantial, and outward. Suspension displacements are compression on the outside wheels, and extension on the inside wheels. Suspension velocities are close to zero. Therefore, damper forces will likewise be close to zero.

This means that the car will be sensitive to springs and anti-roll bars, and insensitive to dampers. Stiffening either front spring, or the front anti-roll bar, will tighten the car (add understeer). Stiffening either rear spring, or the rear anti-roll bar, will loosen the car (add oversteer).

Remember we are assuming that the turn is completely unbanked. In banked turns, the inside suspensions may compress. With soft springs and stiff bars, this may happen at

surprisingly shallow banking angles. In such cases, effect of outside spring and anti-roll bar changes are the same as in a flat turn, but effects of inside spring changes reverse. A stiffer inside front spring will loosen the car (add oversteer). A stiffer inside rear spring will tighten the car (add understeer). With a beam axle, we may have moderate compression of the inside spring even though we have moderate extension at the outside tire, because the spring will be inboard of the tire. In this situation, there will be a *node*,

or a point where there is neither compression nor extension, somewhere between the inside tire and the inside spring.

I am digressing from our simplified flat-turn example here to remind the reader that our example is simple, but the real world is complex. As we proceed through our hypothetical flat turn, it is

important for the reader to pay attention to how things work and why, rather than treating these simplified dynamics as a universally applicable troubleshooting guide.

Part four, or early exit – car-forward acceleration present and increasing, but less important than car-lateral acceleration: The driver now begins to apply greater throttle than required to merely sustain constant speed, and begins to release the car in terms of cornering. The car's lateral acceleration is diminishing, and its forward acceleration is increasing. In this phase, lateral acceleration still dominates the car's behavior in terms of suspension displacements. The car has an outward roll displacement, but this is decreasing, so the car has an inward roll velocity. The car has a rearward pitch displacement, and a rearward pitch velocity.

The effects of roll and pitch velocities are additive at the outside front and inside rear corners, so these will be the most influential wheels in terms of damper tuning. The outside front suspension will have a compressed displacement, but this will be diminishing, so the velocity will be in the extension direction. The inside rear suspension will have an extended displacement, but this will again be diminishing, so the velocity will be in the compression direction.

The effects of roll and pitch are subtractive at the inside front and outside rear. We cannot generalize about the net velocities at these corners, except to say that they will be smaller than at the outside front and inside rear. Therefore, the car will be relatively insensitive to damping changes at these corners.

Note that we now have at least two corners where the displacement and the velocity are opposite in direction. This means that stiffening the spring and stiffening the damper have opposite effects on wheel load, diagonal percentage, and oversteer/understeer balance. Spring and anti-roll bar effects are as in earlier parts of the turn: stiffer front tightens (adds understeer); stiffer rear loosens (adds oversteer). As part four progresses, the outside front and inside rear suspensions approach their static positions, and the influence of spring rates at these corners correspondingly diminishes. So, especially toward the end

of part four, the corners where the shocks matter most are the corners where the springs matter least.

At the outside front, stiffening the low-speed extension damping adds a negative, or compressive force. This reduces wheel loading, reduces diagonal percentage, and loosens the car (adds oversteer) – an opposite effect from stiffening the spring or the bar, as long as the spring is compressed compared to static. At the inside rear, stiffening the low-speed compression damping adds a positive, or extension force. This increases wheel loading, increases diagonal percentage, and tightens the car (adds understeer). Again, this is opposite to the effect of stiffening the spring or bar, as long as the spring is extended compared to static.

Note that I am contradicting the much-repeated advice to soften inside rear compression damping to hasten loading of the inside rear and tighten exit. In fact, softening the inside rear compression damping momentarily diminishes total extension force, and therefore momentarily diminishes inside rear tire loading and diagonal percentage, compared to stiffer inside rear compression damping.

Part five, or late exit – combined forward and lateral acceleration as in part four, but with forward acceleration dominant: The difference between part five and part four is the displacement direction at the outside front and inside rear. Forward acceleration is now large enough, and lateral acceleration is small enough, so that the outside front is extended relative to static (though less than the inside front), and the inside rear is compressed relative to static (though less than the outside rear). Suspension velocities are similar to part four: greatest at the outside front and inside rear; extension on the outside front; compression on the inside rear.

So for shock tuning purposes, exit can be treated as a single phase of the cornering process, and does not need to be broken down into two parts. However, part five is distinct from part four for spring tuning, because the car's response to spring changes at the outside front and inside rear reverses. Stiffening the outside front spring loosens the car (adds oversteer), and stiffening the inside rear tightens the car (adds understeer).

It may be worth clarifying what basis of comparison I'm using when I speak of a change tightening or loosening the car (adding oversteer or understeer). In the above remarks, we are referring to the car's behavior compared to the same part of the turn, before the change in question. It is also possible to consider how a change affects a given part of the turn relative to the previous part of the turn, or some other part, with the same change, as opposed to the same part of the turn, before the change. Both of these modes of comparison are useful. We do have to be mindful of which mode we are using, however.

For example, in a flat turn, stiffening the inside rear spring loosens the car during part four of the turn – but less than it does in part three or part two, especially toward the end of part four. So we might reasonably say that a stiffer inside rear spring tightens the car in

part four, **relative to its condition in the preceding portions of the turn**. Changing the choice of baseline for a comparison can change the outcome of the comparison.

We have given much attention to the distinctions between inside and outside wheels. It will of course be obvious that when the car has to turn both right and left, any given wheel will be an inside wheel in some turns and an outside wheel in others. For most road racing applications, we can condense spring, bar, and shock tuning to a surprisingly simple set of rules:

- 1) To tighten the car (add understeer) overall, add spring and/or bar to the front and/or take spring and/or bar out of the rear. To loosen the car (add oversteer), do the opposite: add spring and/or bar to the rear, and/or take spring and/or bar out of the front.
- 2) To loosen the car (add oversteer) on entry and tighten it (add understeer) on exit, add rear damping and/or take out front damping. For opposite effect, do the reverse.

On ovals, suspension tuning in general is considerably more complex, because we can use asymmetries of many kinds, in addition to everything we use in road racing.

One other nuance mainly relating to ovals is that there may exist what might be called semi-steady-state cornering conditions (my own terminology) between parts one and two and parts three and four. In steady-state cornering, longitudinal acceleration is zero, or near zero. In semi-steady-state cornering, longitudinal acceleration is substantial, but not changing.

Such a state can occur during entry if the driver applies the brakes and then holds braking force roughly constant for a time before releasing the brakes. Assuming the driver is using the tires' full capability, lateral acceleration will also be close to constant. This will place the car on a path whose instantaneous radius is steadily decreasing, even though the car's vector-sum acceleration is not changing. The car will have an outward roll displacement and a forward pitch displacement. These will be substantially constant, and therefore all suspension displacements will be nearly constant and suspension velocities will be close to zero. That means that damping forces will be negligible, and the car will be unresponsive to damper tuning.

Semi-steady-state cornering can also occur during exit if the driver holds forward acceleration roughly constant with the throttle, while using the tires' full capability. Again, neither speed nor instantaneous path radius is constant, but vector-sum acceleration of the car is constant. And again, the suspension velocities will be close to zero, damping forces will be negligible, and the car will not be affected by damper tuning.

Regarding whether to add or reduce damping on compression or extension, and at high velocities or low, some widely repeated advice would have us set compression damping to control sprung mass motion, and set extension damping to control unsprung mass motion. In my opinion this is incorrect. At some time it may have served as simple advice to racers faced with setting the earliest double-adjustable shocks, but now we have

revalveable and four-way adjustable shocks, and reasonably good shock dynos. My advice nowadays is:

- 1) Use low-speed damping, in both extension and compression, to manage transient weight transfer and sprung mass motion. Do not expect this to work unless the surface is smooth enough so that sprung mass motion is the main cause of suspension movement. Use the springs and bars as your main means of managing weight transfer.
- 2) Use damping properties at velocities above 2 in/sec to manage sprung and unsprung mass behavior over road irregularities. Again, both compression and extension matter.
- 3) Keep compression and extension damping in reasonable proportion to each other. At most absolute velocities, extension damping should be at least a little stiffer than compression damping, but usually not more than twice as stiff and never more than three times as stiff unless you are deliberately trying to make the car jack down.

This relatively brief discussion will inevitably not have covered all possible situations, but hopefully it has covered the main principles, and illustrated a useful way to think systematically about springs, anti-roll bars, and shocks. Evaluate effects of a change in terms of whether it adds to or diminishes extension force at the corner you're changing, then imagine the car on the scales and imagine you are adjusting the extension force the same direction with the spring seat or jacking screw, and you can predict the change's effect on car behavior.