Tech Tip: Springs & Dampers, Part One

The Phantom Knowledge

By Matt Giaraffa
matt.giaraffa@optimumg.com

When out on the track, you can spend an entire weekend tuning spring rates and shock settings, and shave a second from lap times. But, how do you know if you are even in the right ballpark, and maybe a completely different spring and shock setup could cut two seconds per lap, especially when working with an unfamiliar car?

In the first of a series of tech tips on springs and dampers, we will explore how to develop baseline spring rates for ride to provide a solid foundation to tune from.

Ride and Single Wheel Bump

The first step in choosing spring stiffness is to choose your desired ride frequencies, front and rear. A ride frequency is the undamped natural frequency of the body in ride. The higher the frequency, the stiffer the ride. So, this parameter can be viewed as normalized ride stiffness. Based on the application, there are ballpark numbers to consider.

- 0.5 - 1.5 Hz for passenger cars
- 1.5 - 2.0 Hz for sedan racecars and moderate downforce formula cars
- 3.0 - 5.0+ Hz for high downforce racecars

Lower frequencies produce a softer suspension with more mechanical grip, however the response will be slower in transient (what drivers report as "lack of support"). Higher frequencies create less suspension travel for a given track, allowing lower ride heights, and in turn, lowering the center of gravity.

Ride frequencies front are rear are generally not the same, there are several theories to provide a baseline. Two examples below show exaggerated plots of what happens with unequal ride frequencies front and rear as the car hits a bump. In Figure 1, we can see the *undamped* vertical motion of the chassis with the front ride frequency higher than the rear. The first period is the most dominant on the car when looking at frequency phase, due to effects of damping to be explained later.

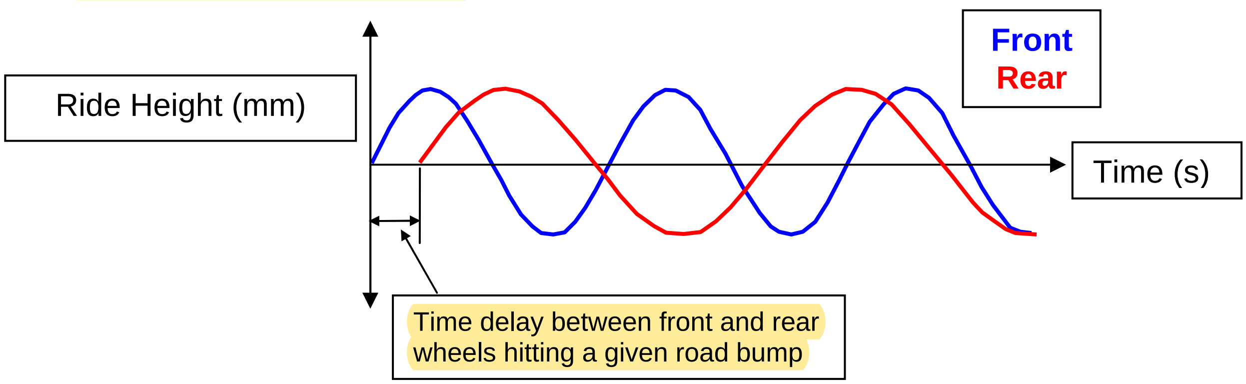


Figure 1. Higher Front Ride Frequency

The out of phase motion between front and rear vertical motion, caused by the time delay between when the front wheel and rear wheel hit the bump, is accentuated by the frequency difference. A result of the phase difference is pitching of the body. To reduce the pitch induced by hitting a bump, the rear needs to have a higher natural frequency to "catch up" with the front, as shown in Figure 2. This notion is called producing a "flat ride", meaning that the induced body pitch from road bumps is minimized.

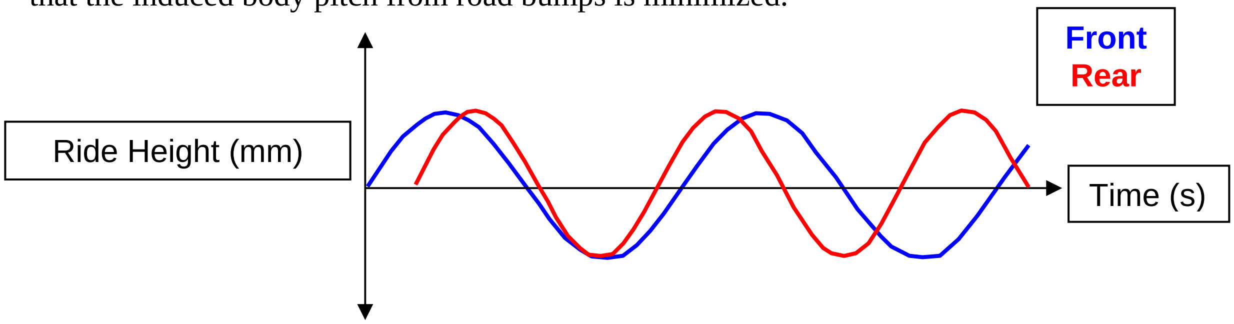


Figure 2. Higher Rear Ride Frequency

For a given wheelbase and speed, a frequency split front to rear can be calculated to minimize pitching of the body due to road bumps. A common split is 10 – 20% front to rear.

The above theory was originally developed for passenger cars, where comfort takes priority over performance, which leads to low damping ratios, and minimum pitching over bumps. Racecars in general run higher damping ratios, and have a much smaller concern for comfort, leading to some racecars using higher front ride frequencies. The higher damping ratios will reduce the amount of oscillation resultant from road bumps, in return reducing the need for a flat ride. Damping ratios will be explained in the next tech tip in detail. A higher front ride frequency in a racecar allows faster transient response at corner entry, less ride height variation on the front (the aerodynamics are usually more pitch sensitive on the front of the car) and allows for better rear wheel traction (for rear wheel drive cars) on corner exit. The ride frequency split should be chosen based on which is more important on the car you are racing, the track surface, the speed, pitch sensitivity, etc.

As an example of ride frequency split front to rear, Figures 3 and 4 shows a simple example of a single degree of freedom vehicle model over an impulse disturbance. The ride frequency difference is 10 percent, 70% critical damping, 100 km/h speed, and 1.75 m wheelbase.

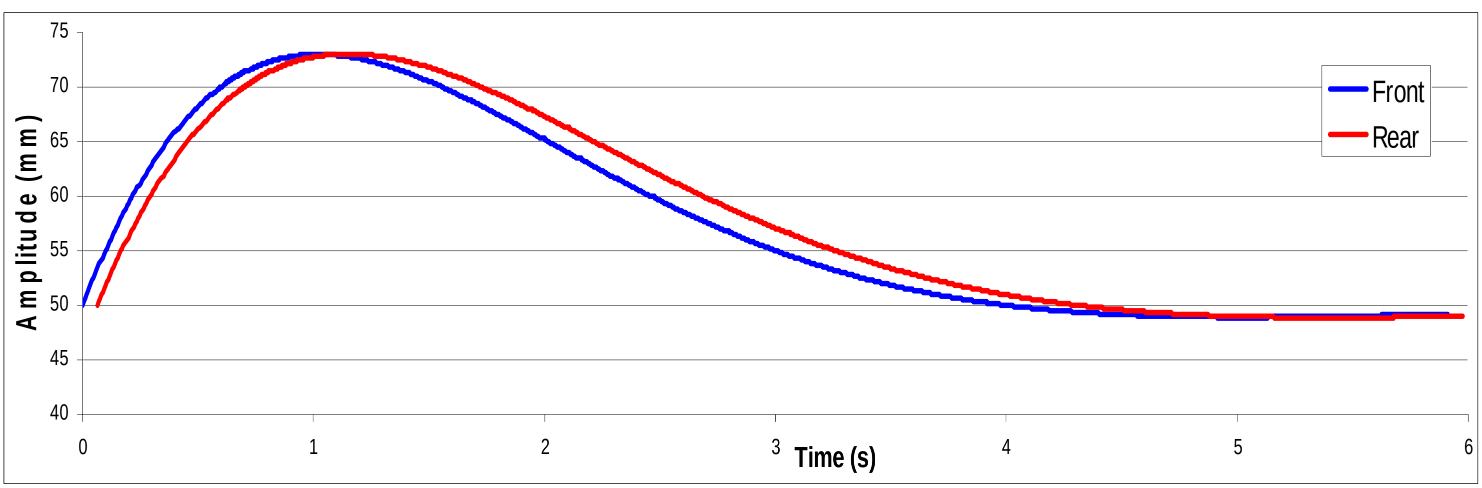


Figure 3. Front ride frequency 10% higher than rear

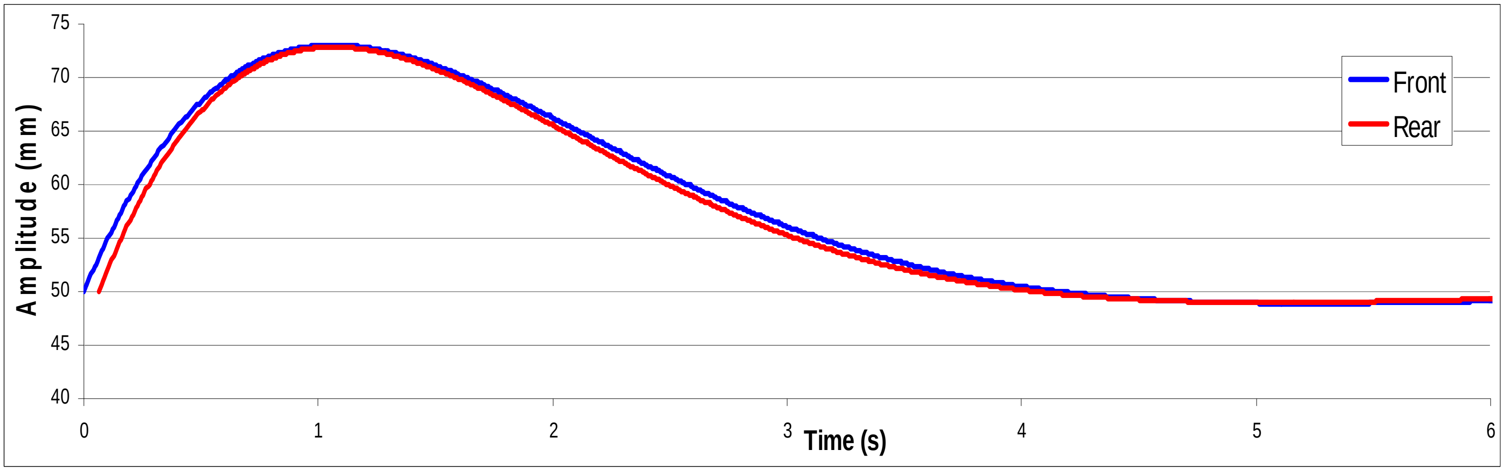


Figure 4. Rear ride frequency 10% higher than front

Once the ride frequencies are chosen, the spring rate needed can be determined from the motion ratio of the suspension, sprung mass supported by each wheel, and the desired ride frequency.

Starting with the basic equation from physics, relating natural frequency, spring rate, and mass:

$$f = 1/(2\pi)\sqrt{\frac{K}{M}}$$

f = Natural frequency (Hz)
K = Spring rate (N/m)
M = Mass (kg)

Solving for spring rate, and applying to a suspension to calculate spring rate from a chosen ride frequency, measured motion ratio, and mass:

$$K_s = 4\pi^2 f_r^{\,2} m_{sm} MR^2$$

K_s = Spring rate (N/m)
m_{sm} = Sprung mass (kg)
f_r = Ride frequency (Hz)
MR = Motion ratio (Wheel/Spring travel)

Shown below in Table 1 is an example using the above calculation. Table 1 shows spring rates needed for a traditional suspension with one ride spring per wheel, and a motion ratio of one. As the suspended mass per corner and ride frequency are changed, the corresponding spring requirement is shown.

Sprung Mass per corner

Ride Frequency (Hz)	100	300	500	700	900	1100	lbs.	Motion Ratio:
	45	136	227	318	409	500	kg	1
1.0	10.3	30.8	51.4	72.0	92.5	113.1	lb/in	
	1.8	5.4	9.0	12.6	16.2	19.7	N/mm	
1.5	23.1	69.4	115.7	161.9	208.2	254.5	lb/in	
	4.0	12.1	20.2	28.3	36.3	44.4	N/mm	
2.0	41.1	123.4	205.6	287.9	370.1	452.4	lb/in	
	7.2	21.5	35.9	50.2	64.6	79.0	N/mm	
2.5	64.3	192.8	321.3	449.8	578.3	706.8	lb/in	
	11.2	33.6	56.1	78.5	100.9	123.4	N/mm	
3.0	92.5	277.6	462.6	647.7	832.7	1017.8	lb/in	Spring Rates
	16.2	48.5	80.8	113.1	145.4	177.7	N/mm	
3.5	125.9	377.8	629.7	881.6	1133.5	1385.3	lb/in	
	22.0	65.9	109.9	153.9	197.8	241.8	N/mm	
4.0	164.5	493.5	822.5	1151.5	1480.4	1809.4	lb/in	
	28.7	86.1	143.6	201.0	258.4	315.8	N/mm	
4.5	208.2	624.6	1040.9	1457.3	1873.7	2290.1	lb/in	
	36.3	109.0	181.7	254.4	327.0	399.7	N/mm	
5.0	257.0	771.1	1285.1	1799.1	2313.2	2827.2	lb/in	
	44.9	134.6	224.3	314.0	403.8	493.5	N/mm	

Table 1. Spring Rates vs Sprung Mass and Ride Frequency

Now, you can create baseline spring rates for your car in ride motion. Next month, single wheel bump springs and ride damping will be explained; eventually over the series, spring rates and damping in ride, roll, and pitch will be covered.

Tech Tip: Springs & Dampers, Part Two

Attack of the Units

By Matt Giaraffa
matt.giaraffa@optimumg.com

After understanding last month's tech tip, you know how to pick ride frequencies for your racecar, and calculate the spring rate needed for the chose frequency. Now, what do you do about the anti-roll bars.

But first, we have had a couple emails asking about units and frequency calculations, so that will be explained further, and then we will explore how to develop baseline spring rates for anti-roll bars (ARB's).

If you don't watch your units, you can't have any pudding.

When calculating any parameters on the car, you must be sure the units are correct. Just because your physics teacher is not overseeing your calculations does not mean you can slack on the units.

One of the biggest mistakes people make is confusing weight and mass. Weight is a force, mass is an inertia- do not forget that. Once you have decided on a ride frequency and want to figure out what spring rate you need units errors will put you far off target. An example is shown below where weight is used in place of mass.

Equation for calculating spring rate:

$$K_s = 4\pi^2 f_r^{\,2} m_{sm} MR^2$$

K_s = Spring rate (N/m)
m_{sm} = Sprung mass (kg)
f_r = Ride frequency (Hz)
MR = Motion ratio (Wheel/Spring travel)

<u>SI Correct example:</u>

m_{sm} = 200 kg f_r = 2 Hz   MR = 1.25

$K_s = 4\pi^2(2\text{ Hz})^2(200\text{ kg})(1.25)^2$ = 49348 N/m = 49.3 N/mm

<u>SI Incorrect example:</u>

m_{sm} = 1960 N f_r = 2 Hz   MR = 1.25

$K_s = 4\pi^2(2\text{ Hz})^2(1960\text{ N})(1.25)^2$ = 483611 N/s^2 = Wrong and not useful

<u>English Correct example:</u>

m_{sm} = 440 lbm f_r = 2 Hz   MR = 1.25

$K_s = 4\pi^2(2\text{ Hz})^2/386.4*(440\text{ lbm})(1.25)^2$ = 282 lbf/in

<u>English Incorrect example:</u>

m_{sm} = 440 lbf f_r = 2 Hz   MR = 1.25

$K_s = 4\pi^2(2\text{ Hz})^2(440\text{ lbf})(1.25)^2$ = 108566 lbf/s^2 = Wrong and not useful

As you can see, with incorrect units, the results you get are useless. Be careful to use weight/force when needed and use mass when needed- the two are not interchangeable.

Roll

Similar to choosing ride frequencies for bump travel, a roll stiffness must be chosen next. The normalized roll stiffness number is the roll gradient, expressed in degrees of body roll per "g" of lateral acceleration. A lower roll gradient produces less body roll per degree of body roll, resulting in a stiffer vehicle in roll. Typical values are listed below:

- 0.2 – 0.7 deg/g for stiff higher downforce cars
- 1.0 – 1.8 deg/g for low downforce sedans

A stiffer roll gradient will produce a car that is faster responding in transient conditions, but at the expense of mechanical grip over bumps in a corner.

Once a roll gradient has been chosen, the roll gradient of the springs should be calculated, the anti-roll bar stiffness is used to increase the roll gradient to the chosen value. The roll gradient is usually not shared equally by the front and rear. At OptimumG, we call the roll gradient distribution the Magic Number (Milliken calls it Total Lateral Load Transfer Distribution). The Magic Number is expressed as the percentage of the roll gradient taken by the front suspension of the car.

As a baseline, use 5% higher Magic Number than the static front weight distribution.
Roll gradients are degrees of body roll per g of lateral acceleration.
Roll rates are Newton-meters of torque per degree of body roll or ARB twist. The following equations do no take into account roll due to the tires.

<u>Roll gradient of ride springs:</u>

$$\frac{\phi_r}{A_y} = \frac{-W \times H}{K_{\phi F} + K_{\phi R}}$$

H = Cg to Roll axis dist (m)

W = Vehicle weight (N)

ϕ_r/A_y = Roll gradient from ride springs (deg/g)

$$K_{\phi F} = \frac{\pi\,(t_f^2)K_{LF}K_{RF}}{180(K_{LF}+K_{RF})}$$

$K_{\phi F}$ = Front roll rate (Nm/deg roll)

t_f = Front track width (m)
K_{LF} = LF Wheel rate (N/m)
K_{RF} = RF Wheel rate (N/m)

Remember that wheel rate is spring rate/ MR^2; the effect of the spring at the wheel

$$K_{\phi R} = \frac{\pi\,(t_r^2)K_{LR}K_{RR}}{180(K_{LR}+K_{RR})}$$

$K_{\phi R}$ = Rear roll rate (Nm/deg roll)

t_r = Rear track width (m)
K_{LR} = LR Wheel rate (N/m)
K_{RR} = RR Wheel rate (N/m)

Total ARB roll rate needed to increase the roll stiffness of the vehicle to the desired roll gradient:

$$K_{\phi A} = \frac{\pi}{180}\left[\frac{K_{\phi DES}K_T(t^2/2)}{[K_T(t^2/2)\pi/180 - K_{\phi DES}]}\right] - \frac{\pi K_W(t^2/2)}{180}$$

$K_{\phi A}$ = Total ARB roll rate needed (Nm/deg roll)
$K_{\phi DES}$ = Desired total roll rate (Nm/deg roll)
K_W = Wheel rate (N/m)
K_T = Tire rate (N/m)
t = Average track width between front and rear (m)

$$K_{\phi DES} = WH/(\phi/A_y)$$

W = Weight of vehicle (N)
H = Vertical distance from roll center axis to Cg (m)
ϕ/A_y = Desired total roll gradient, chosen earlier (deg/g)

Front and Rear Anti-Roll Bar stiffness:

$$K_{\phi FA} = K_{\phi A}N_{mag}MR_{FA}^2/100$$

$K_{\phi FA}$ = FARB roll rate (Nm/deg twist)
$K_{\phi A}$ = Total roll rate (Nm/deg roll)
N_{mag} = Magic Number (%)
MR_{FA} = FARB Motion ratio

$$K_{\phi RA} = K_{\phi A}(100-N_{mag})MR_{RA}^2/100$$

$K_{\phi RA}$ = RARB roll rate (Nm/deg twist)
$K_{\phi A}$ = Total roll rate (Nm/deg roll)
N_{mag} = Magic Number (%)
MR_{RA} = RARB Motion ratio

Remember, the chassis acts as a torsional spring in roll. It is worth comparing the roll rate of your suspension to the roll rate of your chassis- if the chassis twists as much as the suspension, it could be a larger area of concern than the suspension. With steady state roll angles different front to rear, or different roll frequencies front to rear, chassis torsion will be induced- this should be kept in mind.

Try this on an excel spreadsheet and have fun. Email us for comment and questions.

Next time we will speak about the *3rd spring* and introduce damping.

Tech Tip: Springs & Dampers, Part Three

Revenge of the Damping Ratio

By Matt Giaraffa
matt.giaraffa@optimumg.com

After understanding the first two tech tips in the Spring & Damper series, you know how to choose ride frequencies for your racecar, calculate the spring rate needed for the chosen frequencies, choose a roll gradient, and calculate the stiffness required from the anti-roll bars to produce your desired roll gradient. Now, what is the deal with these "third" springs people and using, and how in the world do I know where to start when it comes to damping on the racecar? Pitch springs will be skipped for now, as damping baselines will be much more useful.

Single Wheel Bump

In the first Spring & Damper tech tip, you learned how to pick a ride frequency and calculate the needed spring rate for your car. What happens if you want a lower ride frequency for grip over bumps, but need a higher ride frequency to keep the car off the ground from aerodynamic load or banking? The "third" spring is a solution. In addition to the ride spring on each wheel, an additional spring can be added that operates in ride, but not single wheel bump. This allows a lower frequency in single wheel bump than overall ride.

For example, the front ride frequency could be set at 1.5 Hz, with each front wheel set at 1.0 Hz to provide a softer suspension for bumps, providing more mechanical grip, without sacrificing overall ride. Attaching a third spring to a T-bar anti-roll bar (ARB) is the most popular method of accomplishing this, shown in Figure 5. Lower single wheel bump frequencies are useful on bumpy tracks where more vertical stiffness is desired- reducing the compromise between vertical stiffness and mechanical grip over undulations.

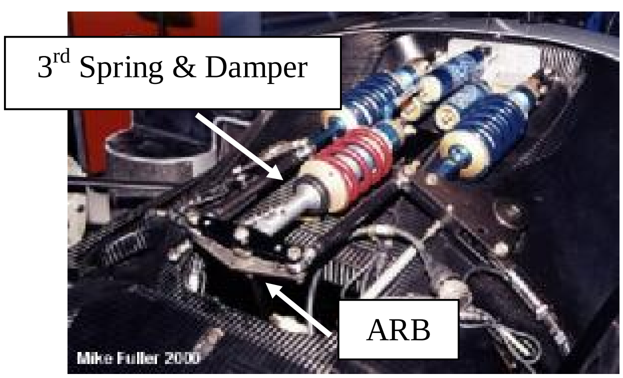


Figure 5. Third spring connected to T-Bar ARB

Once the ride frequencies are chosen, the spring rate needed can be determined from the motion ratio of the suspension, sprung mass supported by each wheel, and the desired ride frequency.

$$K_{ss} = 4\pi^2 f_s^{\,2} m_{sm} MR_s^{\,2}$$

K_{ss} = Single wheel spring rate (N/m)
m_{sm} = Sprung mass on that corner (kg)
f_s = Single wheel bump frequency (Hz)
MR_s = Motion ratio (Wheel/Single wheel spring travel)

Calculating the single wheel spring rates above allows you to calculate the spring rate of the third ride spring. The process below is shown for the front suspension; procedure for calculating the rear is the same.

$$K_{rs} = \left[4\pi^2 f_r^{\,2} m_{fsm} - K_{lss}/MR_{ls}^{\,2} - K_{rss}/MR_{rs}^{\,2}\right] MR_r^{\,2}$$

K_{rs} = "Third spring" rate, front (N/m)
f_r = Front ride frequency (Hz)
m_{fsm} = Front suspended mass (kg)
K_{lss} = LF single wheel spring rate (N/m)
MR_{ls} = Motion ratio of K_{lss}
K_{rss} = Right front single wheel spring rate (N/m)
MR_{rs} = Motion ratio of K_{rss}
MR_r = Motion ratio of third spring

<u>Third Spring Example:</u>

K_{rs} = "Third spring" rate, front (N/m)
f_r = 2.0 (Hz)
m_{fsm} = 300 (kg)
K_{lss} = 30000 (N/m)
MR_{ls} = 1.3
K_{rss} = 30200 (N/m)
MR_{rs} = 1.2
MR_r = 1.5

$$K_{rs} = (4\pi^2(2)^2 300 - 30000/1.3^2 - 30200/1.2^2)1.5^2 = 19.5\ \text{N/mm}$$

What damping should I start with?

Why a racecar needs 10 shocks for complete control….

As you can have four different spring rates- ride, single wheel bump, roll, and pitch, in an ideal situation, you will have four different damping ratios (ζ). The first step is to calculate the desired damping in ride, single wheel bump, roll, and finally pitch.

An undamped system will tend to eternally vibrate at its natural frequency. As the damping ratio is increased from zero, the oscillation trails off as the system approaches a steady state value. Eventually, critical damping is reached- the fastest response time without overshoot. Beyond critical damping, the system is slow responding. An important point to understand that will be useful when tuning the shocks on the car is that once any damping is present, the amount of damping does not change the steady state value- it only changes the amount of time to get there and the overshoot. Examples are shown below, and the effect of dampers on a sprung mass system is shown in Figure 6.

$$C_{cr} = 2\sqrt{K_w m_{sm}}$$

K_w = Wheelrate (N/m) m_{sm} = Sprung mass (kg)

$$\zeta = C/C_{cr}$$

C = Damping force (N) ζ = damping ratio

Wheelrate Example:

$$K_w = K_s/MR^2$$

Kw = Wheelrate (N/mm) Ks = Spring rate (N/mm)
MR = Motion Ratio (wheel/spring travel)

SI Example:

m_{sm} = 300 kg K_w = 90000 N/m

$$C_{cr} = 2\sqrt{90000 \text{ N/m} * 300 \text{ kg}} = 10392 \text{ N*s/m} = 10.4 \text{ N/(mm/s)}$$

English Example:

m_{sm} = 750 lbm K_w = 700 lbf/in

$$C_{cr} = 2\sqrt{700 \text{ lbf/in} * 750 \text{ lbm}/386.4 \text{ in/s}^2} = 73.7 \text{ lbf/(in/s)}$$

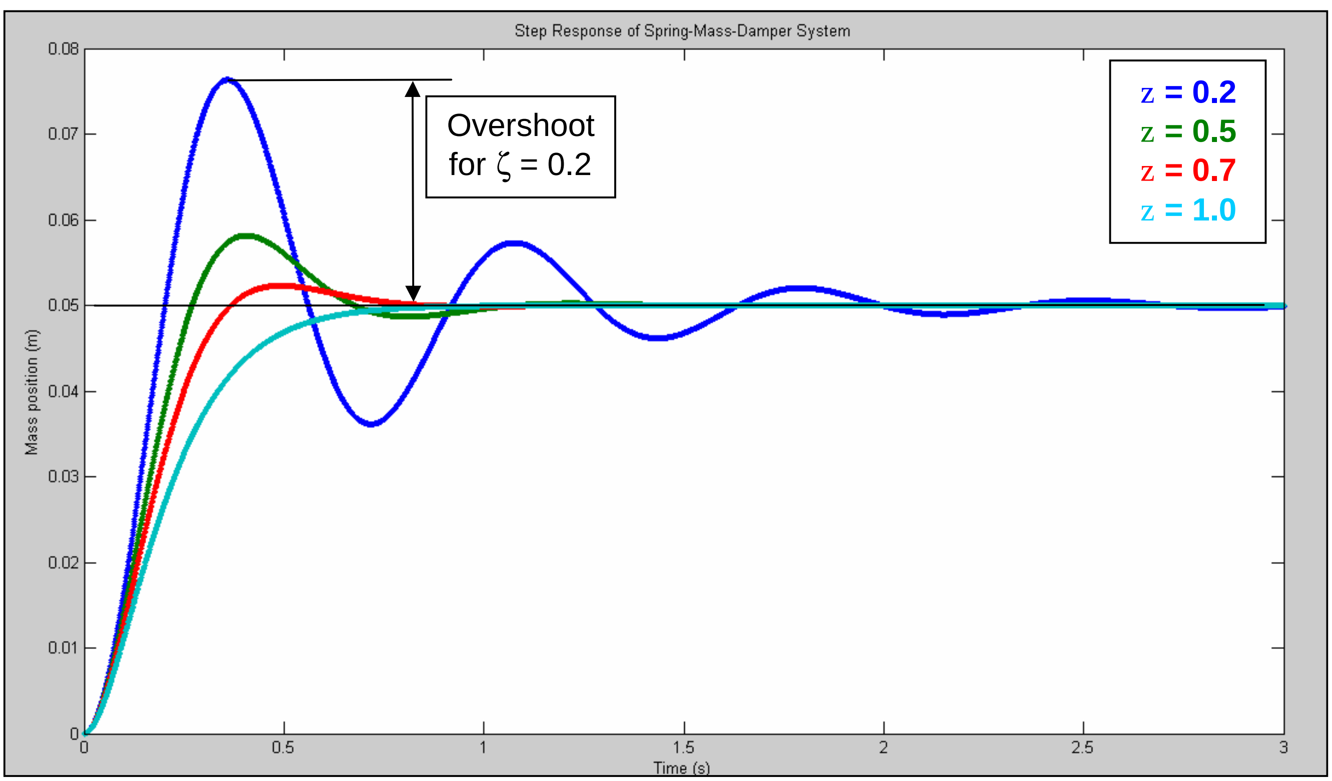


Figure 6. Effect of damping ratio to a sprung mass system

Ride and Single Wheel Bump Damping

The first place to start on damping is ride motion. Choosing a damping ratio is a tradeoff between response time and overshoot- you want the smallest of each. Passenger vehicles generally use a damping ratio of approximately 0.25 for maximizing ride comfort. In racecars, 0.65 to 0.70 is a good baseline; this provides much better body control than a passenger car (less overshoot), and faster response than critical damping. Some successful teams end up running damping ratios in ride greater than 1, this does not indicate that damping ratios in ride should be large, it shows that there is a compensation for a lack of damping in roll and pitch, as the dampers that control ride motion usually also control the roll and pitch motion.

Up to this point in the spring and damper tech tips, you should be able to choose ride frequencies, roll gradients, calculate needed spring and ARB stiffnesses, and now choose a damping ratio in ride.

Next time, the development of a baseline force versus velocity curve for the ride dampers will be explained.

Tech Tip: Spring & Dampers, Episode Four

A New Understanding

By Matt Giaraffa & Samuel Brisson
matt.giaraffa@optimumg.com
samuel.brisson@optimumg.com

Correction:

In Spring & Dampers, Part Two this equation is incorrect:

$K_{\phi DES} = WH/(\phi/A_y) + K_{\phi R} + K_{\phi F}$

It should read:

$K_{\phi DES} = WH/(\phi/A_y)$

Moving on to this month's topic: understanding the basics of damping in a racecar and developing a baseline ride damper curve.

Transmissibility

Before going into the details of damping force versus damper shaft velocity, the concept of transmissibility should be understood.

From driving street cars, we all know that if you hit a speed bump going very slow the body of the car (sprung mass) moves vertically almost as much as the wheels. Hitting the same bump going fast (you know you have done it, especially in a rental car) the body of the car does not move nearly as much. The size of the bump was the same, but the body motions were different depending on the speed at which you hit it. The cause of this is that response of the system (the car sitting on the suspension) is dictated by the frequency and amplitude of the input. Hitting the speed bump faster increases the frequency of the disturbance, producing a different response. To quantify this reality we use the concept of transmissibility.

The transmissibility (TR) is the ratio between output and input amplitude. In the above case, the input amplitude is the height of the speed bump, with output amplitude being vertical movement of the body.

$$TR = \frac{output\ \ amplitude}{input\ \ amplitude}$$

Rearranging the equation above gives a method to calculate vertical body movement from input disturbance amplitude and the transmissibility, which you can calculate from the mass, spring rate, and damping ratio.

$$output\ \ amplitude = TR \times input\ \ amplitude$$

In our case the input is a displacement of the wheel caused by the speed bump. For example, let's say the speed bump is four inches tall- moving the wheel four inches up, and four inches back down. The input is the wheel movement and the input amplitude is four inches. The distance the mass of the car will move up and down is the output amplitude. The time it takes for the wheel to complete the up-down cycle is the frequency divided by two. As you increase the speed, you increase the frequency- and for sprung mass systems the transmissibility changes with frequency. Figure 7 shows the transmissibility for a spring-mass-damper system with a fixed damping ratio of 0.5- a simple model of the car hitting the speed bump.

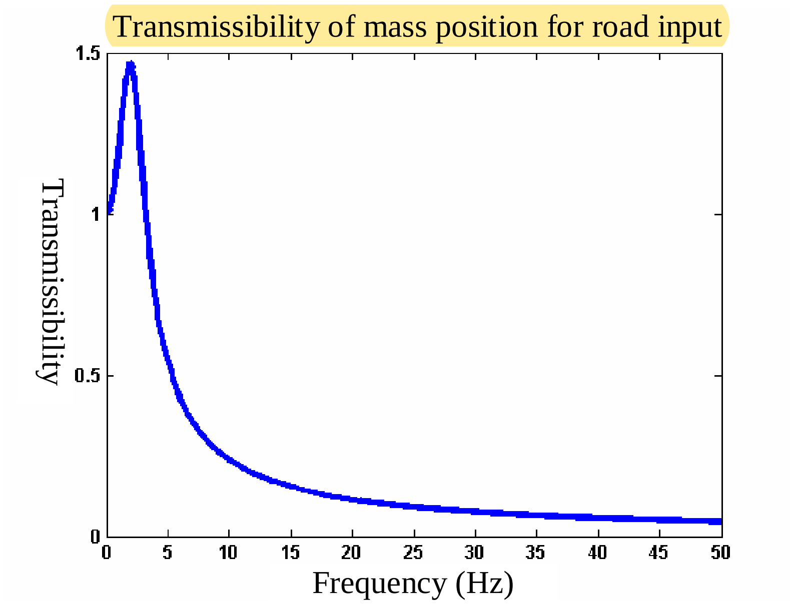


Figure 7. Transmissibility of a spring-mass-damper system

Very Low frequency example

TR = 1 Input amplitude = 4 in

Output amplitude = 1 x 4in = 4 in

The output has the same amplitude as the input, this means the body moves the same as the wheels. This makes sense since you go really slow over the speed bump with nearly no spring deflection.

As you go faster, the frequency increase and you will reach a frequency where the body movement reaches a maximum, this is the resonant frequency. At this frequency the transmissibility is maximum and higher than one (for the interested the resonant frequency is equal to the ride frequency calculated in Spring & Dampers Tech Tip Part 1). In our case the resonant frequency is around two hertz and the transmissibility is 1.45.

Resonant frequency example

TR = 1.45 Input amplitude = 4 in

Output amplitude = 1.45 x 4in = 5.8 in

This means that at this frequency the car movement is greater than the input. The driver will feel a harsh ride, and it will feel like the body of the car is catapulted off the speed bump.

High frequency example

TR = 0.1 Input amplitude = 4in

Output amplitude = 0.1 x 4in = 0.4 in

This means that at this frequency the car movement is reduced and the suspension absorbs the bumps. The driver will feel a smooth ride.

Now that we defined what is transmissibility we can use it to find damping values. The first step is to examine the transmissibility for different damping ratios, as shown on Figure 8.

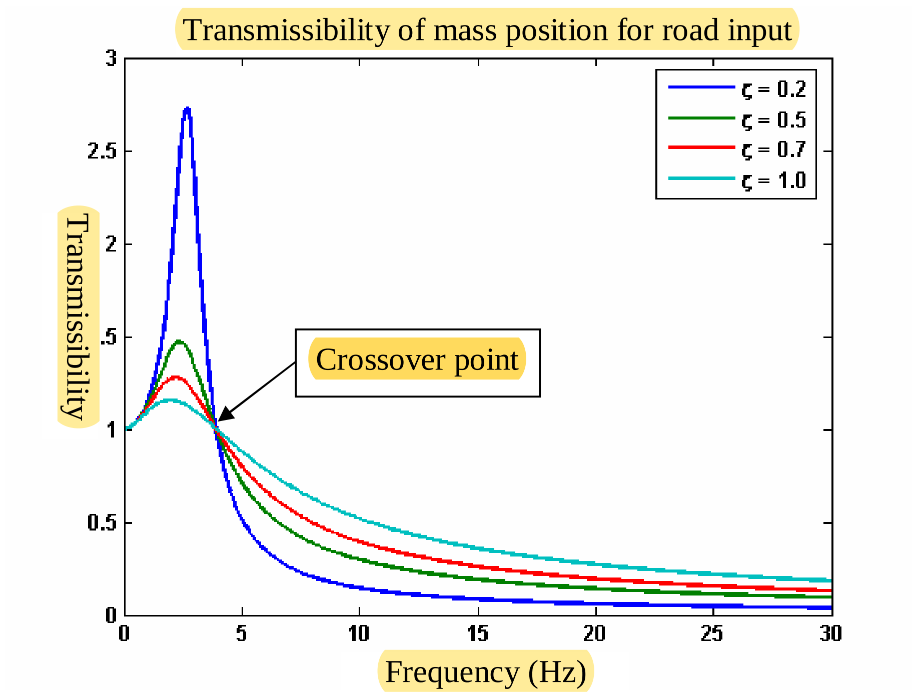


Figure 8. Transmissibility for different damping ratios

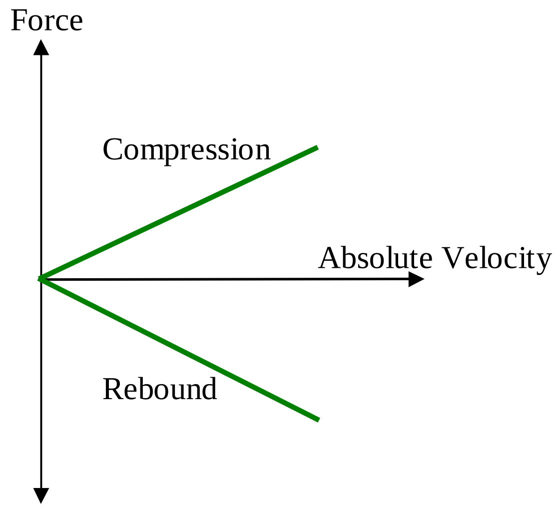
In order to tune for maximum grip, you want the lowest transmissibility possible- as the body is bouncing around, the forces on the springs are changing, decreasing the grip. As you can see in Figure 8, there is a crossover point, this is at $\sqrt{2}$ * Resonant Frequency. At low frequencies, if we increase the damping we reduce the maximum transmissibility, which is good, making a higher damping ratio at low frequencies desirable. On the other side of the crossover point, low damping ratios give a lower transmissibility, meaning low damping ratios are desirable at high frequency. Since low frequencies generally correspond to low damper velocity, and high frequencies generally correspond to high damper velocity, you can see now that you want a higher damping ratio at low damper velocity than high damper velocity. The next section shows how to take the above theoretical explanation and apply it to come up with a baseline damper curve.

Baseline Ride Damping Curve

A damper curve is the famous (or infamous) *force vs velocity* curve of a shock absorber. Calculating a baseline damper curve in ride is explained below in several steps. If you have no idea where to start on dampers, or want to know if what you are using is in the theoretically correct ballpark, this should be a big help. To begin, you need the suspended mass supported by each damper (this is the suspended mass per corner on a normal car with one spring and damper

per corner), the ride frequencies, and damping ratios chosen earlier. For more explanation on these three, see the first three Spring & Damper Tech Tips.

The first step is linear for the entire velocity range later to be modified in steps two and three. Calculating the slope of the initial damping curve for step one is shown below in Figure 8.

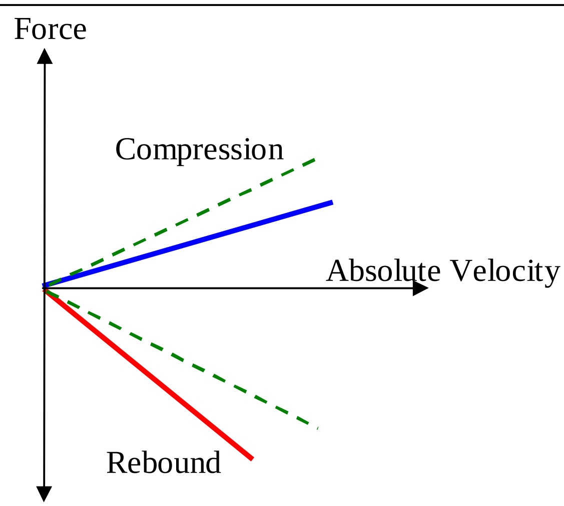


Initial Slope = $(4\pi\zeta_{ride}\omega_{ride}m_{sm})$ N/(m/s)

ζ_{ride} = Damping ratio in ride
ω_{ride} = Ride frequency (Hz)
m_{sm} = Sprung mass supported by damper (kg)

Figure 9. Initial Damping Curve

The basic curve above has the same damping ratio in compression and rebound. In reality, it is more desirable to have a lower damping ratio in compression and higher damping ratio in rebound. A guideline for modifying the above plot is shown in Figure 9. These produce an average damping ratio the same as above, but the damping forces produced in rebound travel being twice that of compression damping forces. When the suspension is compressed, energy is being stored in the spring, and during rebound energy is being released from the spring. Since the job of a damper is to absorb energy for the purpose of controlling resonance, less damping force is required by the damper during compression due to the energy going into the spring. Similarly, more damping force is required by the damper during rebound, as it has to control resonance and the energy being released by the spring.



Compression Slope = 2/3*Initial

Rebound Slope = 3/2*Initial

Figure 10. Modified shock curve

Unfortunately, the new shock curve is still not ideal- it will cause harshness over small amplitude, high frequency road disturbances. It is desirable to reduce the damping ratio at high shock speed (above the Low/High speed split velocity) to reduce this effect. Referring back to the transmissibility example above, remember you want the lowest transmissibility for mechanical grip. Lower frequencies on the transmissibility plot usually translate to lower damper velocities, where a damping ratio of approximately 0.7 is ideal. However, at higher frequencies that usually translate to higher damper velocities, lower damping ratios around 0.2 produce lower transmissibility. The split between low and high damper velocity should be initially set to isolate body motions (low velocity) and track bumps (high velocity), or if you're feeling adventurous, correlate the crossover point on the transmissibility graph to a damper velocity as a split point to start from.

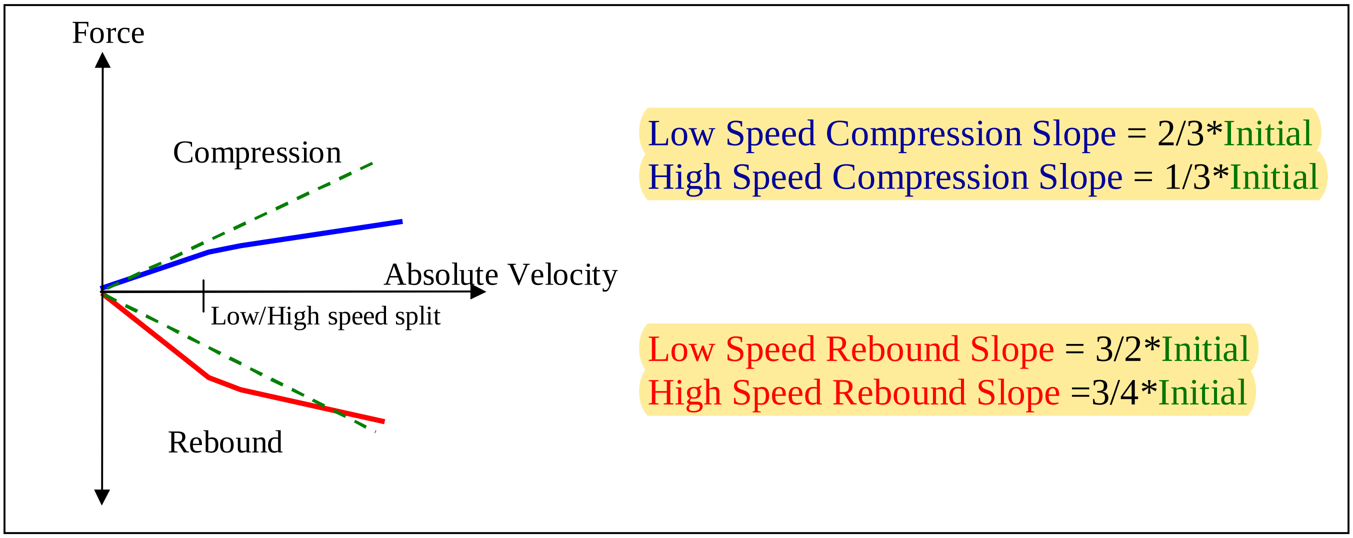


Figure 11. Modified high speed shock curve

Now you have a baseline force versus velocity curve for the shocks in ride. Next month, calculating a baseline curve for roll and pitch damping will be explained. On a car where rules allow, and you choose to do so, a suspension system can be designed where damping adjustments can be made that isolate ride, roll, and pitch. Finding an ideal baseline for roll and pitch damping are discussed below, however, most cars due to their suspension design are forced to make a compromise between the three (this explains the point mentioned above with successful cars using non-ideal damping ratios in ride).

With one ride damper for each wheel, and one on each "third spring"….6 shocks so far.

As always, for more in-depth knowledge, OptimumG offers 3-Day seminars around the world, in-house seminars, a 12-Day Workshop, simulation, and consulting services.

Tech Tip: Spring & Dampers, Episode Five

The Damping Ratio Strikes Back

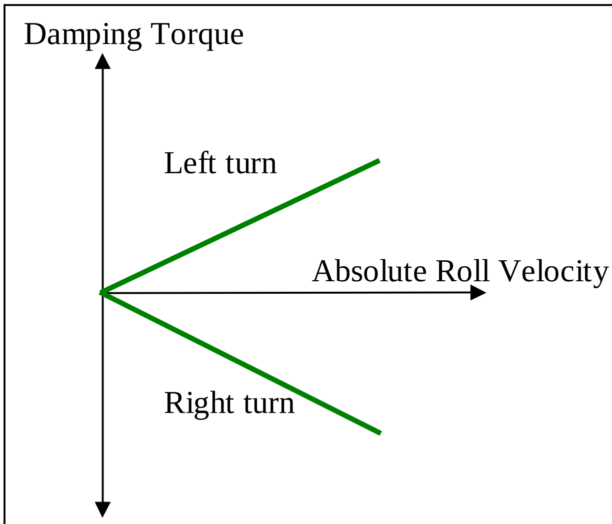
By Matt Giaraffa
matt.giaraffa@optimumg.com

Roll Damping

Often an overlooked aspect on racecars is the damping in roll. Most of the time, cars have a damper for each single-wheel spring, but not a damper for the anti-roll bars. The goal of this tech tip is to explain the benefits of using a roll damper. Where this setup is not allowed in the rules, the knowledge will help in understanding the setup and compromises in the lack of a roll damper.

As with ride and single wheel motion, roll can be seen as a mass oscillating on a damped spring. In this case, however, instead of linear motion derived from **F** = m**a** and the ride frequency, roll is rotation around an axis derived from **T** = Ia (Torque = Moment of inertia * Angular acceleration) and the natural roll frequency.. Applied to the roll of a racecar, "T" is the roll torque, "I" is the roll inertia of the sprung mass, and "α" is the roll acceleration.

Similar to ride and single wheel damping, you want to start off by choosing a damping ratio. The same compromise of response time and overshoot applies here- making 0.65 – 0.7 a good baseline damping ratio in roll for mechanical grip. Shown below in Figure 12 is developing a baseline roll damping plot for damping torque versus vehicle roll velocity.



Initial Slope = $\pi^2\zeta_{roll}\omega_{roll}I_{roll}/45$ Nm/(deg/s)

ζ_{roll} = Damping ratio in roll
ω_{roll} = Roll frequency (Hz)
I_{roll} = Roll inertia of sprung mass

Figure 12. Initial Roll Damping Curve

Unlike ride and single wheel damping where you next modify the slopes to make rebound damping higher, roll damping should start off symmetrical. The only modification necessary is similar to high speed roll-off used for ride and single wheel damping. Since the car has higher frequency roll vibration happening as the body rolls at a much lower frequency into a corner, the lower damping ratio at higher roll velocity tends to even the load on the tires as the car rolls.

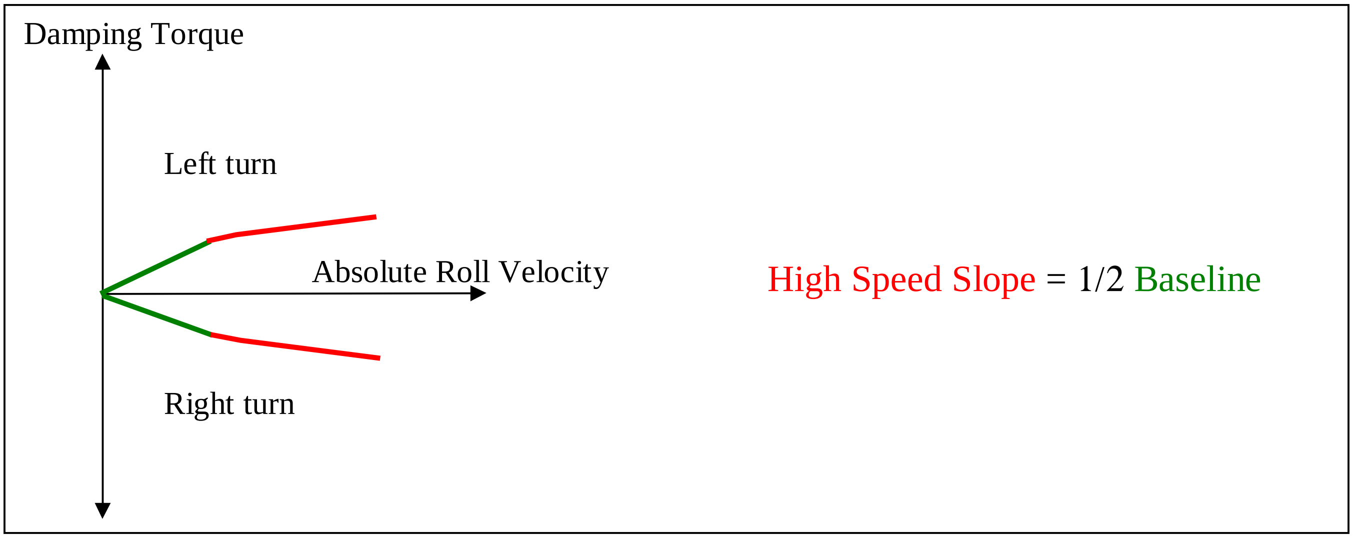


Figure 13. Modified roll damper curve

The damping curve calculated above is a good baseline for most cars- in some situations the roll damping can deviate significantly, such as Indy cars that run extremely high damping coefficients in roll for improved stability. However, for most racecars the above baseline is a good place to begin. Once you calculate the baseline for damping torque vs vehicle roll velocity, use the below equation to calculate the damper curve for the roll damper itself.

$F_{\text{roll damper}} = F_{\text{wheel}} / MR^2$

$F_{\text{roll damper}}$ = Damping force of roll damper (Nm)
F_{wheel} = Roll damping force at the wheel from Figure 13. (Nm)
MR = Motion ratio of roll damper- body roll/roll damper angular displacement

Most cars do not use a dedicated roll damper, necessitating a compromise in damping between ride and roll. However, for complete control of the car handling one roll damper is needed for the front and rear suspension, adding in the previously counted six dampers, brings the count so far to eight dampers on the car for complete control.

Next month for the final episode of the Spring & Damper tech tip series, damping in pitch will be explained.

As always, for more in-depth knowledge, OptimumG offers 3-Day seminars around the world, in-house seminars, a 12-Day Workshop, simulation, track support, and consulting services.

Tech Tip: Spring & Dampers, Episode Six

Return of the Race Engineer

By Matt Giaraffa
matt.giaraffa@optimumg.com

After examining damping in roll last month, we will now look at springs and damping in pitch of the car. As explained last time with roll damping acting in a polar coordinate system, pitch is very similar- except the sprung mass is rotating about the pitch axis instead of the roll axis. Roll damping is a usually overlooked aspect in a suspension system, and pitch damping is overlooked even more so.

Most race series prevent the use of dedicated roll and pitch dampers, but even that being the case, understanding how the whole system works will increase your engineering.

Pitch Springs

Before going into detail, some common confusion about what a pitch spring is needs to be cleared up- you do not want to analyze pitch damping and apply it to the wrong spring. As shown in the third Spring & Damper tech tip, below in Figure 14 is a picture of a car with two single wheel springs, one roll spring (the anti-roll bar), and one RIDE spring. The ride spring in the middle is commonly incorrectly referred to as a pitch spring. It controls ride stiffness on the front of the car- but coincidentally also operates in pitch. A pitch spring controls the relative ride height between the front and rear of the car- and they are rare.

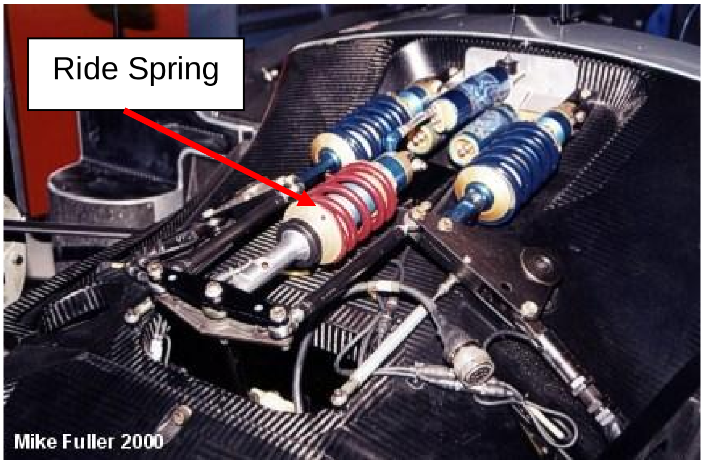


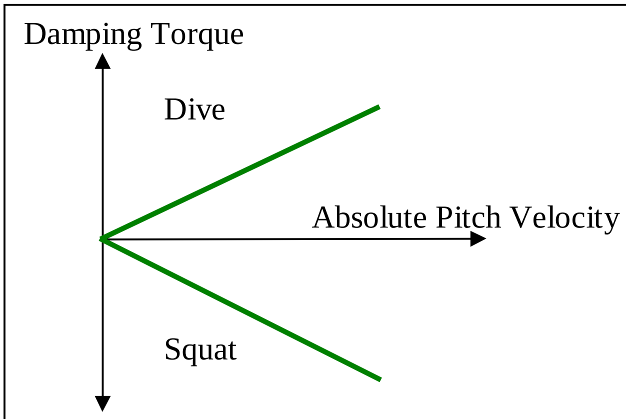
Figure 14. Pitch Spring Misconception

A pitch spring installed on a car, most of the time, is a torsion bar linking the front and rear control arms or rockers together. With this configuration, the spring provides no resistance in ride, or roll, but when the front and rear suspensions move in opposite directions (pitch), the torsion pitch spring resists the motion.

Pitch Damping

As with roll, pitch can be seen as a mass oscillating on a damped spring- derived from **T** = Iα (Torque = Moment of inertia * Angular acceleration) and the natural pitch frequency. Applied to the pitch of a racecar, "T" is the pitch torque, "I" is the pitch inertia of the sprung mass, and "α" is the pitch acceleration.

Similar to all other modes of damping, you want to start off by choosing a damping ratio. The same compromise of response time and overshoot applies here- making 0.65 – 0.7 a good baseline damping ratio in pitch for mechanical grip. Shown below in Figure 15 is developing a baseline pitch damping plot for damping torque versus vehicle pitch velocity.

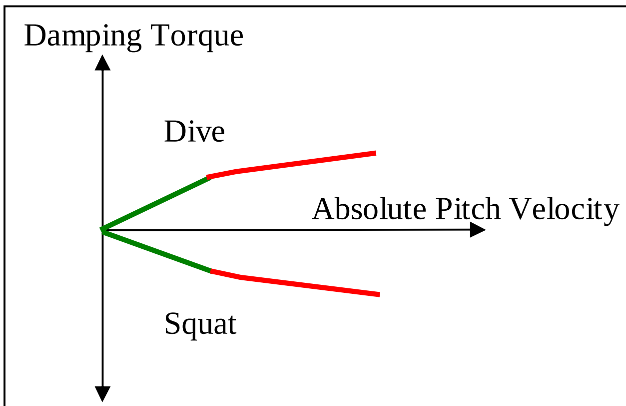


Initial Slope = $\pi^2\zeta_{pitch}\omega_{pitch}I_{pitch}/45$ Nm/(deg/s)

ζ_{pitch} = Damping ratio in pitch
ω_{pitch} = Pitch frequency (Hz)
I_{pitch} = Pitch inertia of sprung mass

Figure 15. Initial Pitch Damping Curve

Similar to roll damping, pitch damping should start off symmetrical. The only modification necessary is similar to high speed roll-off used for ride and single wheel damping. Since the car has higher frequency pitch vibration happening as the body pitches at a much lower frequency into a braking zone or acceleration, the lower damping ratio at higher pitch velocity tends to even the load on the tires as the car pitches.



High Speed Slope = 1/2 Baseline

Figure 16. Modified pitch damper curve

The damping curve calculated above is a good baseline for most cars- in some situations the pitch damping can deviate, for example, a high damping coefficient in pitch could be useful on a very high downforce car due to aerodynamic pitch sensitivity- stiff pitch damping will slow down the pitch, and hence the aero balance change. However, for most racecars the above baseline is a good place to begin. Once you calculate the baseline for damping torque vs vehicle pitch velocity, use the below equation to calculate the damper curve for the pitch damper itself.

$$F_{pitch\ damper} = F_{wheel} / MR^2$$

$F_{pitch\ damper}$ = Damping torque of pitch damper (Nm)
F_{wheel} = Pitch damping torque at the wheel from Figure 16. (Nm)
MR = Motion ratio of pitch damper- body pitch/pitch damper angular displacement

Before calculating your desired pitch damping baseline plot, you need your pitch frequency. Using the pitch stiffness ($K_{\theta des}$) calculated similarly to roll stiffness used in the Spring & Damper Tech Tip Part 5.

$$\omega_{pitch} = 1/(2\pi)\sqrt{\frac{180*K_{\theta des}}{\pi*I_{pitch}}}$$

Example

SI
m_{sm} = 1500 kg ω_{pitch} = 2.8 Hz ζ_{pitch} = 0.7 I_{pitch} = 410 kg m^2

Initial Slope = π^2 * 0.7 * 2.8 Hz * 410 kg m^2 / 45 = 176 Nm/(deg/s)

English
m_{sm} = 3300 lbm ω_{pitch} = 2.8 Hz ζ_{pitch} = 0.7 I_{pitch} = 9704 lbm ft^2

Initial Slope = π^2 * 0.7 * 2.8 Hz * 9704 lbm ft^2 / 45 = 108 lb-ft/(deg/s)

Most cars do not use a dedicated pitch damper, necessitating a compromise in damping between ride and pitch. However, for complete control of the car handling one pitch damper is needed for the left and right suspension, adding in the previously counted eight dampers, brings the count in total to ten dampers on the car for complete control.

This concludes the series of Spring & Damper Tech Tips. We hope they have been useful in helping people understand suspension systems more, and keep posted for a new tech tip topic next month.

As always, for more in-depth knowledge, OptimumG offers 3-Day seminars around the world, in-house seminars, a 12-Day Workshop, simulation, track support, and consulting services.