

## Understanding your Dampers: A guide from Jim Kasprzak

### Introduction

I believe dampers are the most misunderstood components on a race car. In fact in the U.S. they are often called shock absorbers, even though they really don't absorb shock! Even the well know race car engineer Carroll Smith remarked, "Sometimes I think that I would have enjoyed racing more in the days of friction shocks. Since you couldn't do anything much to them or with them, I would have spent a lot less time being confused." It is thought by many that dampers are a black art, and that there is some "magic" to developing and tuning dampers. I am here to dispel those myths, and provide you information and tools to develop dampers for FSAE cars. There is art to damper development and tuning. The art is using engineering knowledge and tools to optimize your car's dampers.

Since you are college students, you probably don't have time to read this whole section of the book. However, there is a lot of valuable information contained in each part. If you decide to skim this chapter, be sure to read the following:

- Gas pressure
- Proper Suspension Design
- Damping force calculations
- Damper Fit Check
- Handling
- The Care and Feeding of your Dampers

### A Shock Absorber is a Damper

A shock absorber damps the motion of both the sprung and unsprung masses of the car. The sprung mass is the body and chassis of the car, and everything supported by the springs. The unsprung mass is composed of all the components not supported by the springs. This includes the suspension upright and all components attached to it; the brake caliper, brake disc, wheel, tire and a portion of the suspension arms.

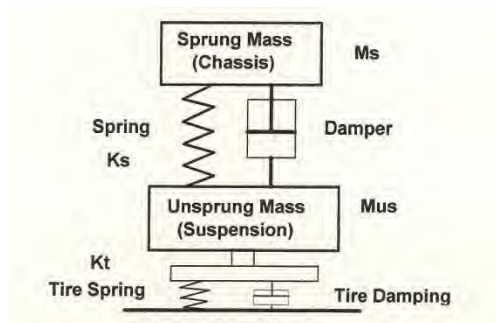


Figure 1

Although there is a small amount of damping in the tire (approximately 2%) the suspension damper must do the majority of the damping for this whole system. That means damping the sprung mass, unsprung mass and the tire! While springs and sway bars produce force based on their displacement or deflection, dampers produce force based on how fast you move them, or their velocity. Therefore, the amount of damping produced is proportional to velocity. This means the damper works like a dynamic spring; it produces force only when it is moving.

The primary function of the damper on the vehicle is to damp the sprung and unsprung masses at their resonant frequencies. For the sprung mass, this includes the pitch, heave and roll resonant frequencies. A secondary function is to control the rate of weight transfer during transients such as braking, corner entry and acceleration. In the best case, dampers are “The frosting on the cake.” If everything else in the suspension design and component selection is correct they simply damp the vibrations at resonant frequencies, control the rate of weight transfer and enhance vehicle comfort and performance. In the worst case dampers “Hold the cake together!” They are asked to compensate for structural deficiencies, control heave, pitch or roll imbalances, and compensate for design deficiencies like lack of suspension travel.

### **How Much Damping?**

The suspension on a FSAE car is two spring/mass/damper systems in series (see Figure 1). The first consists of the suspension spring, body/chassis mass (sprung mass) and the damper. The second consists of the tire (as the spring), suspension parts (unsprung mass) and the little bit of tire damping. Figure 1 shows the system for a single corner of the car. Remember, there are FOUR of these systems on your car. This means the dampers must damp the 16+ degrees of freedom of the car!!! However, for simplicity, we will deal with the primary vibrations of the vehicle; heave, pitch and roll of the sprung mass, and unsprung resonance. First and foremost you must get these under control. Once you do that, the rest is tuning to optimize performance.

So how much damping do you want? Just enough! Any more reduces grip, increases tire force variation and tire wear, and makes the ride worse. What we want is the correct amount of damping. Overdamping or underdamping increases the variation of tire force to the track, reducing tire adhesion or grip.

Most text books state the proper damping ratios are 0.2-0.3. This is appropriate for passenger cars, but not enough for FSAE and other race vehicles with higher spring and tire rates, and thus, higher natural frequencies. As a rule of thumb, a FSAE car will require a damping ratio of 0.5-0.7 to control the heave, pitch and roll resonances of the sprung mass, and a damping ratio of 0.3-0.5 to control the unsprung mass. But this is NOT the entire answer, so keep reading!

### **Damper Construction**

Before we get into damping force calculations let's examine the basic elements of a damper, compare the three basic types of damper construction and look at the advantages and disadvantages of each.

Elements of a damper (See Figure 2)

- Main piston: In all dampers the main piston contains the primary valving components and produces the majority of the damping forces. In all three constructions, all the rebound force is produced by the Main Piston.
- Compression Piston: The Compression Piston has several functions:
  - Produces compression force based on the rod displacement through the Compression Piston.
    - This results in lower compression pressures for the same damping force, typically resulting in less tire force variation and better grip.
  - Provides a pressure balance to the main piston during the compression stroke to prevent cavitation.
    - This enables dampers with Compression Pistons to operate with lower gas pressure.

Note that the Monotube damper does not have a Compression Piston.

- Gas Separator Piston: Keeps the gas separated from the oil.
- Main Piston Tube: This is the tube where the Main Piston operates. Note in the Monotube damper it is also the outer tube.
- Reservoir Tube: The Reservoir Tube is the outer tube on a Twin Tube damper and creates the area for extra oil and the gas pressure in a Twin Tube shock.

# Damper Constructions

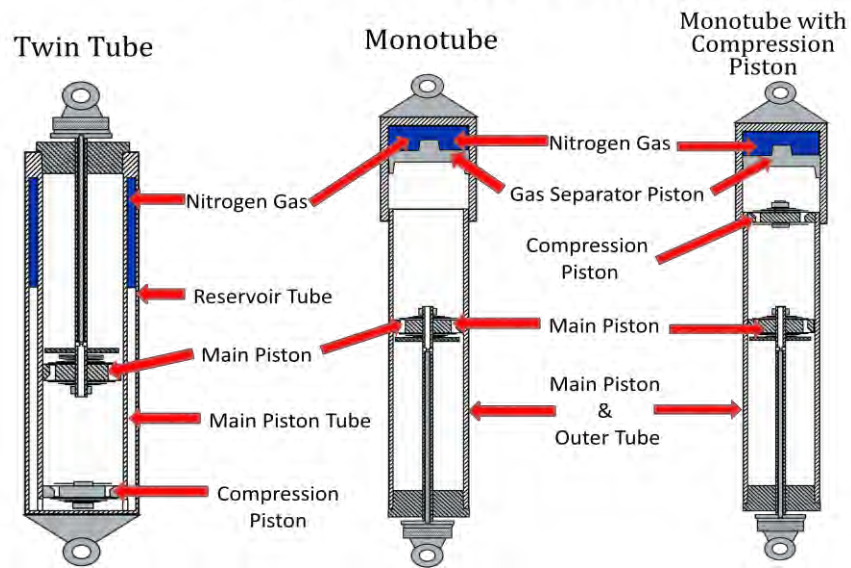


Figure 2

## Advantages and Disadvantages of Each Construction

### Twin Tube

- Advantages
  - Inexpensive
  - Compression Piston. This means lower compression pressures and typically lower gas pressure can be used.
- Disadvantages
  - Has to be mounted rod up to avoid cavitation
  - No Gas Separator Piston to separate the gas from the oil. This can result in the gas and the oil mixing, reducing damping forces.
  - Smaller piston diameter for a given outer tube diameter. The result is higher operating pressures and more fade.

### Monotube (no Compression Piston)

- Advantages
  - Oil is separated from the gas.
  - Higher quality construction than most Twin Tube shocks
- Disadvantages
  - No Compression Piston. This means higher gas pressure to prevent cavitation.
  - Nitrogen gas chamber in damper may make damper longer. This can be eliminated with a piggyback gas canister construction.

### Monotube with Compression Piston

- Advantages
  - Oil is separated from the gas.
  - Compression Piston. This means lower compression pressures and typically lower gas pressure can be used.
- Disadvantages
  - Nitrogen gas chamber and Compression Piston may make damper longer. This can be eliminated with a piggyback gas canister construction.

### Basic Damping Force Equations

In order to properly use a damper we must first understand how it works. Let's begin with the simplest to understand, the Monotube without Compression Piston. The rebound and compression force equations are shown in Figure 3 below.

## Monotube Damping Force Calculations

$$\text{Area} = \frac{\pi * \text{Diameter}^2}{4}$$
$$\text{Force} = \text{Pressure} * \text{Area}$$
  
$$A_{\text{Rebound}} = A_{\text{Tube}} - A_{\text{Rod}}$$
$$F_{\text{Rebound}} = (P_1 - P_2) * A_{\text{Rebound}}$$
  
$$A_{\text{Compression}} = A_{\text{Tube}} - A_{\text{Rod}}$$
$$F_{\text{Compression}} = (P_2 - P_1) * A_{\text{Compression}}$$
  
$$P_1 = \text{Rebound Pressure}$$
$$P_2 = \text{Compression Pressure}$$
$$P_3 = \text{Gas Pressure}$$

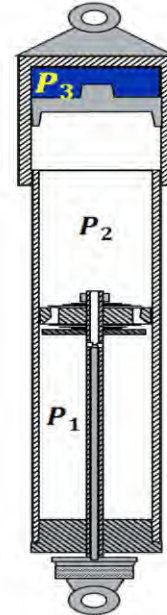


Figure 3

Note that the damping forces are generated by the pressure drop across the piston. In the Monotube without Compression Piston all the damping forces in compression and rebound are generated by pressure drops across the piston. Typically a series of discs on both sides of the piston is used to generate the pressure drops. We will discuss this further later in this chapter.

The damping equations for the Twin Tube damper and Monotube with Compression Piston are the same. As we saw in Figure 2, both have compression pistons. These generate additional compression force during the compression stroke due to the pressure drop across the compression piston. The compression force generated by the compression piston is proportional to the area of the rod. This is because the volume of fluid that passes through the compression piston is equal to the area of the rod times the stroke. The damping force equations for the Twin Tube and Monotube with Compression Piston are shown below in Figure 4.

## Twin Tube & Monotube with Compression Piston Damping Force Calculations

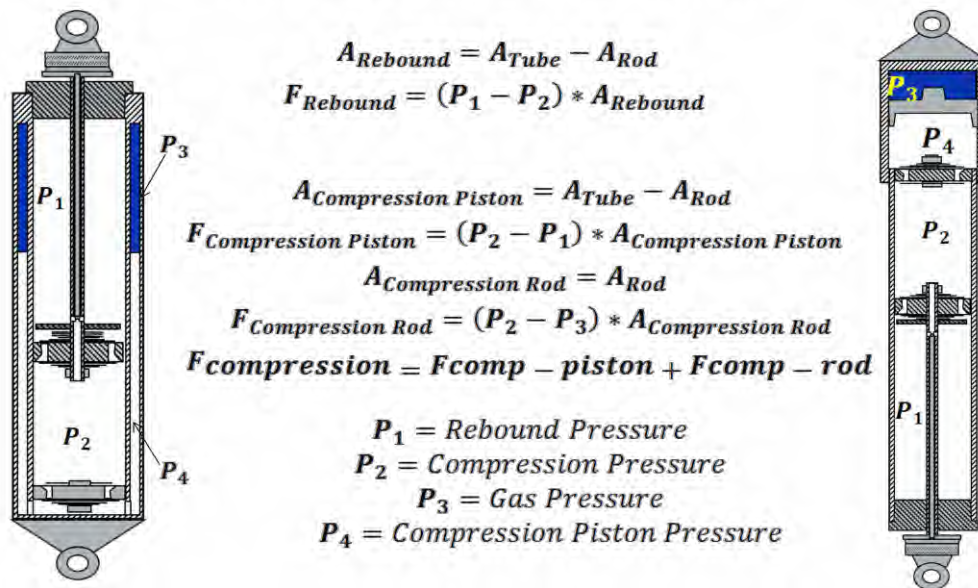


Figure 4

Please note the addition of the force from the compression rod. This is the additional force generated by the Compression Piston. This is combined with the compression force generated by the piston to yield the total compression force.

### Damper Valve Operation

As we look farther inside the damper, let's examine how the actual valving works. There are three stages to the valving operation; low speed or damper velocities zero to 50.8 mm/sec (0-2 in/sec), mid-speed or damper velocities 50.8-203.2 mm/sec (2 to 8 in/sec) and high speed, above 203.2 mm/sec (8 in/sec). The rebound and compression valves (piston and compression piston) all work the same. Below are descriptions of what controls each valving stage, followed by valving illustrations and force/absolute velocity graphs.

The Monotube with Compression Piston is used for illustration. The piston valving in all three damper constructions work the same. The compression piston valving in a Twin Tube damper and Monotube with Compression Piston work exactly the same.

#### Low Speed Control

- Low speed is zero to 50.8 mm/sec (0 to 2 in/sec)
- In the valving, this is controlled by:
  - Bleed holes or bleed discs
  - The by-pass jet or adjustment needle
  - Leakage around the piston

See Figures 5 and 6

# Rebound – Low Speed

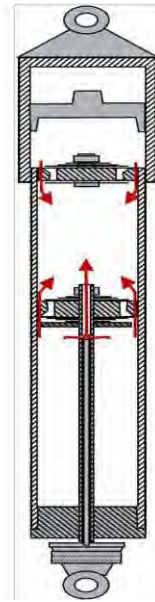
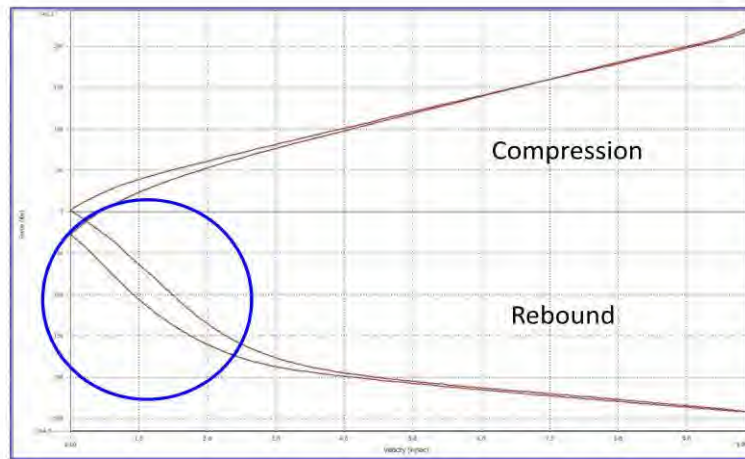


Figure 5

# Compression – Low Speed

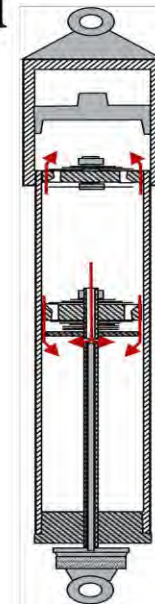
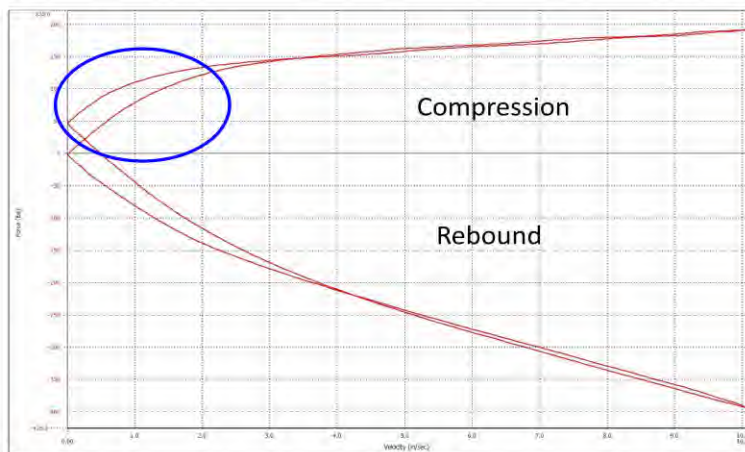


Figure 6

## Mid-Speed Control

- Mid speed is 50.8 to 203.2 mm/sec (2 to 8 in/sec)
- In the valving, this is controlled by:
  - Disc stack preload
  - Thickness of discs in disc stack

See Figures 7 and 8

## Rebound – Mid Speed

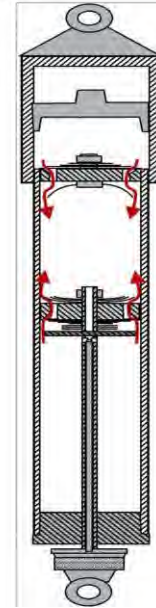
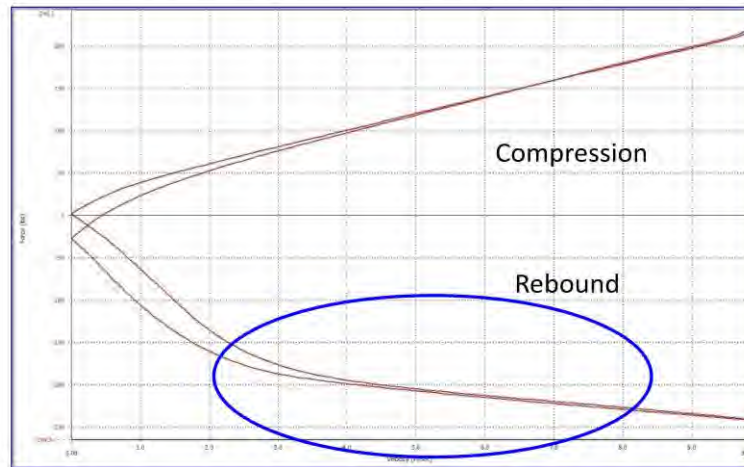


Figure 7

## Compression – Mid Speed

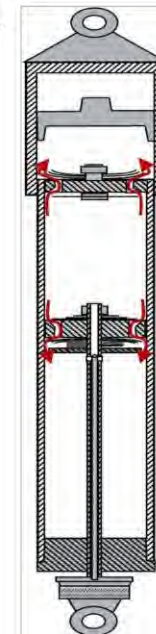
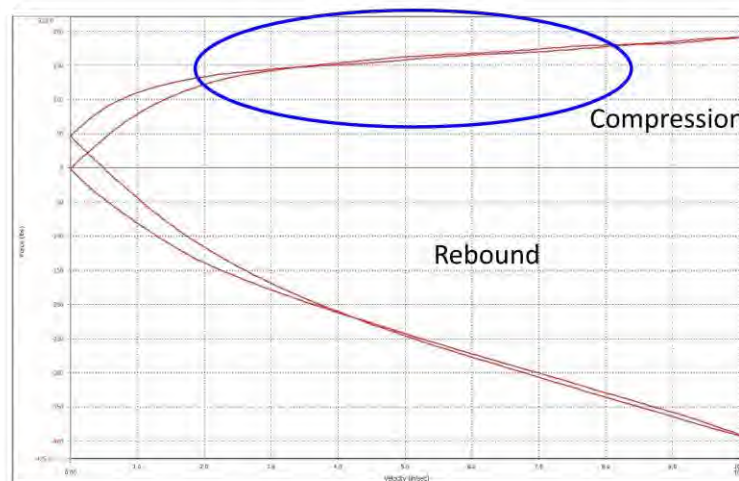


Figure 8

## High Speed Control

- High speed is greater than 203.2 mm/sec (8 inches/second)
- In the valving, this is controlled by:
  - High speed portion of disc stack
  - Piston port size
  - Restriction of disc stack travel

See Figures 9 and 10

## Rebound – High Speed

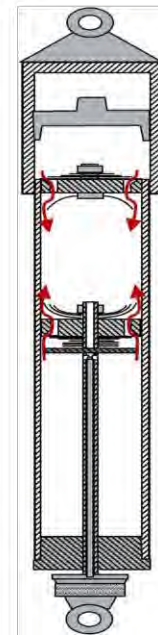
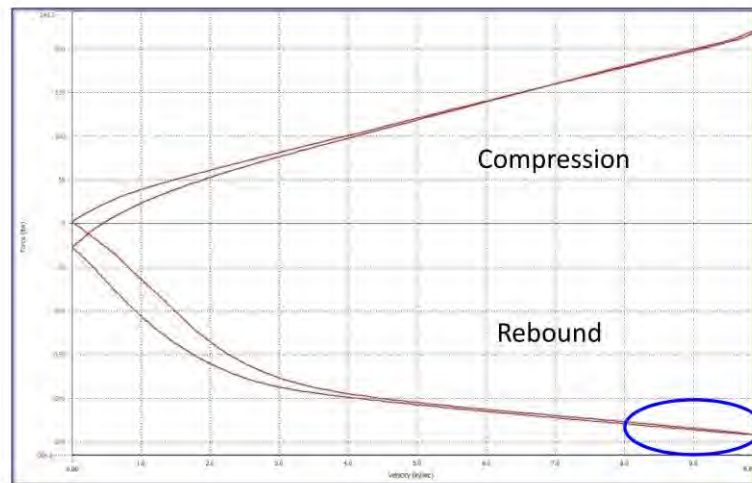


Figure 9

## Compression – High Speed

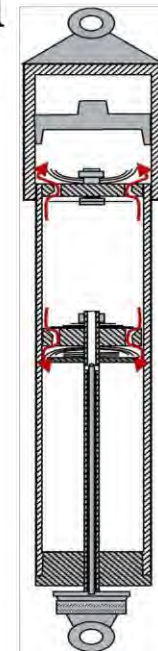
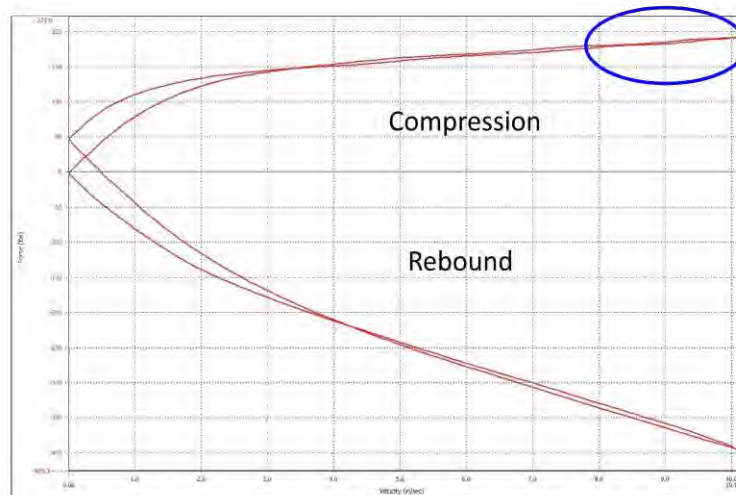


Figure 10



## Gas Pressure

Gas pressure improves the performance of both twin tube and monotube dampers. The gas pressurizes the oil to compress the air bubbles trapped in the oil. This reduces the compressibility of the oil, reducing the hysteresis. Figure 11 illustrates a proper damping curve with no lags or incongruities and little to no hysteresis

Proper Damping Curve

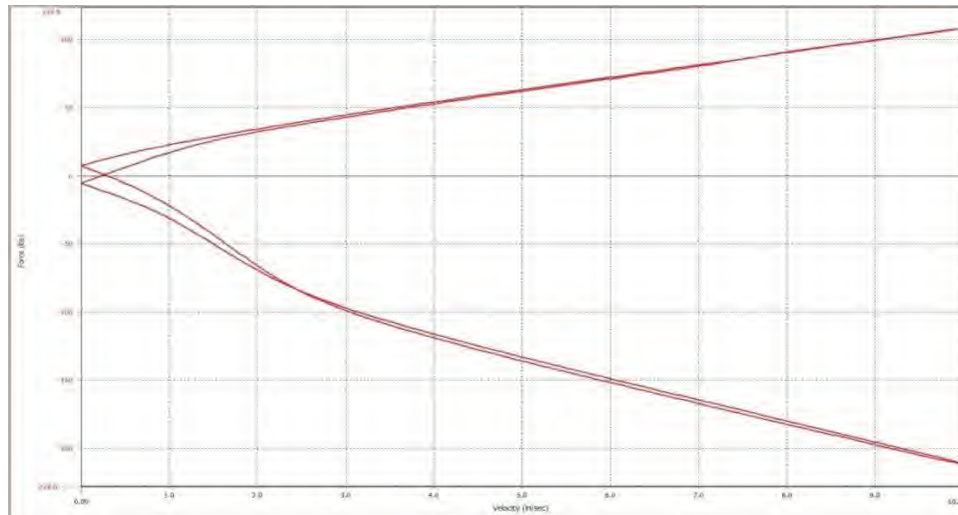


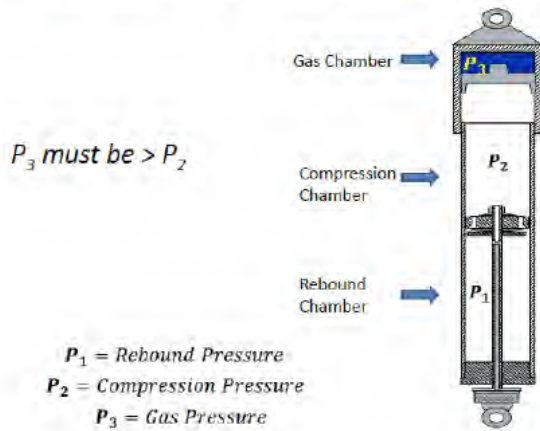
Figure 11

Proper gas pressure is essential to the proper performance of a monotube damper. In a monotube, the gas pressure must be high enough to assure the oil ALWAYS flows through the main piston during the compression stroke. In this respect, the pressure in the gas chamber (P3) must always be higher than the pressure in the compression chamber (P2). See Figure 12. If this is not true, the piston valve closes during the compression stroke, causing a negative pressure in the rebound chamber (P1). This produces cavitation in the rebound chamber and results in lag.

The equation is even more complex in a monotube with compression piston. The pressure drop across the compression piston must be greater than the pressure drop across the main piston, and the pressure in the gas chamber must be greater than the pressure in the compression piston chamber.

## Gas Pressure Balance

### Monotube Gas Pressure Balance



### Monotube with Compression Piston Gas Pressure Balance

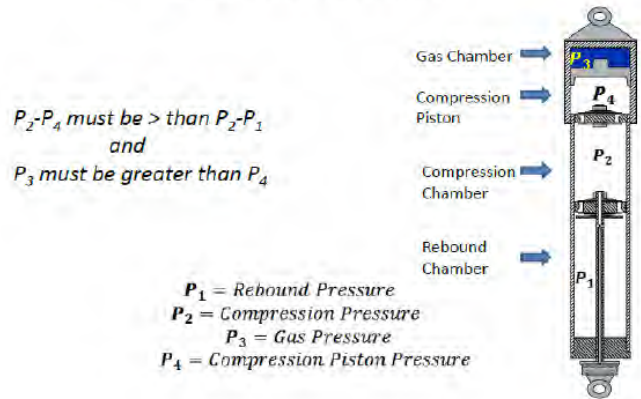


Figure 12

### Proper Suspension Design

Before considering how much damping you need for the car, you must design the suspension correctly to make the best use of the damper, and then properly specify the damper for your suspension design. Of course you want the lightest damper that has the maximum stroke and requires the least amount of space. But one must consider many factors when packaging the damper into the suspension system. These include required wheel travel, jounce bumper travel, desired wheel rates, strength requirements and packaging constraints. Probably the most important is motion ratio.

Quite simply, the motion ratio is the ratio of the spring or damper displacement to the wheel displacement. The equations are shown below. Since almost all FSAE cars use spring over dampers, the ratios are usually the same.

Motion Ratios

$$\text{Spring Motion Ratio} = \frac{\text{Spring Displacement}}{\text{Wheel Displacement}}$$

$$\text{Damper Motion Ratio} = \frac{\text{Damper Displacement}}{\text{Wheel Displacement}}$$

Please note that everyone does not define motion ratios the same! Often times the motion ratio is defined as the ratio of the wheel displacement to the spring or damper displacement. So, when discussing motion ratios be sure you agree on the definition!

For FSAE cars I recommend a motion ratio at or near 1.0. This means to meet the FSAE requirement of 50.8 mm (2 inches) of wheel travel, you will want to specify a damper with at least 63.5mm (2.5 inches) of travel. At a 1.0 motion ratio, this will allow for 50.8mm (2 inches) of wheel travel and 12.7mm (0.5 inches) of jounce bumper travel.

There are several reasons for using high motion ratios. The first is higher motion ratios require lower spring rates for the same wheel rates. Lower spring rates are also lighter, and result in less spring and shock friction as well as lower component loads. The other reason is greater damper travel and higher shock velocities. Since dampers perform better

at higher velocities, and the wheel displacements are quite small on a FSAE car, higher motion ratios produce better shock performance. Figure 13 shows a chart comparing travels, velocities and spring preload for various motion ratios.

Motion Ratio Comparison

Motion Ratio	1:1	0.75:1	0.5:1
Wheel Travel	25 mm	25 mm	25 mm
Shock/Spring Travel	25 mm	18.8 mm	12.5 mm
Wheel Velocity	50 mm/sec	50 mm/sec	50 mm/sec
Shock Velocity	50 mm/sec	37.5 mm/sec	25 mm/sec
Wheel Rate	25 N/mm	25 N/mm	25 N/mm
Spring Rate Required	25 N/mm	44 N/mm	100N/mm
Spring Preload	50 kg	67 kg	100 kg

Figure 13

### Ride Frequencies

The frequency at which a sprung system will resonate is call the resonant frequency. As we discussed earlier, a suspension system is a series two of spring mass damper systems. Thus, there are two undamped resonant frequencies, or what are called the sprung mass and the unsprung mass natural frequencies. The equations are shown below.

Sprung Mass

$$\omega_n(s) = \frac{1}{2\pi} \sqrt{\frac{(K_W * K_T)/(K_W + K_T)}{M_S}}$$

$\omega_n(s)$  = Sprung mass Natural Frequency (Hz)

$K_W$  = Wheel Rate

$K_T$  = Tire Spring Rate

$M_S$  = Sprung Mass

Unsprung Mass

$$\omega_n(us) = \frac{1}{2\pi} \sqrt{\frac{K_W + K_T}{M_{US}}}$$

$\omega_n(us)$  = Unsprung mass Natural Frequency (Hz)

$K_W$  = Wheel Rate

$K_T$  = Tire Spring Rate

$M_{US}$  = Unsprung Mass

Typical natural frequencies of sprung and unsprung mass are listed below:

#### Sprung Mass Natural Frequencies

- FSAE vehicle 2.5 – 3.5 Hz
- Passenger car 1 – 2 Hz
- NASCAR Cup car 1.5 – 4 Hz
  - All 4 corners are different!
- Indy Car 5 – 7 Hz

#### Unsprung Mass Natural Frequencies

- FSAE vehicle 15 – 19 Hz
- Passenger car 10 – 12 Hz
- NASCAR Cup car 15 – 17 Hz
- Indy car 23 – 27 Hz

The undamped frequency at which the sprung mass will resonate or bounce is often called the ride frequency. This is the same as the sprung natural frequency. Since the front and rear will resonate or bounce at different frequencies, we typically reference a front and rear ride frequency. The reason the front and rear have different ride frequencies is to reduce the pitch of the vehicle over bumps. The rear ride frequency is typically higher than the front, so that after encountering a bump, the rear will “catch up” with the front, and the front and rear will move in phase. See the illustration in Figure 14.

Different Front and Rear Ride Frequencies

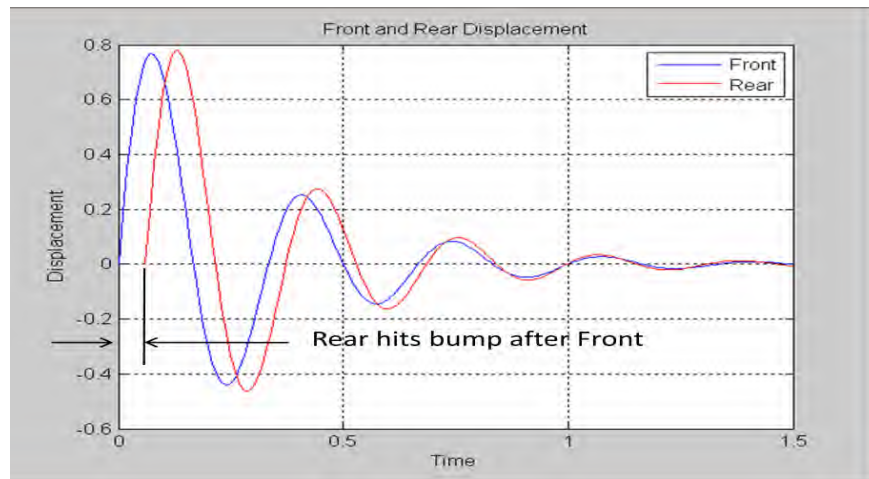


Figure 14

Typically the ride frequencies for a FSAE vehicle are between 2.5 and 3.5 Hz, with the rear 0.2-0.4 Hz higher than the front. What are the “right” ride frequencies? The ones that make YOUR car go the fastest! Don’t get hung up on ride frequency numbers. They are for reference. Of course it is useful to calculate the front and rear ride frequency to determine initial ride balance and starting spring rates. But ultimate spring rates and damping ratios that make the car go fast are much more important than the theoretical numbers.

#### Damping Force Calculations

Most text books reference damping coefficients when discussing the amount of damping required for a vehicle. In the real world, damper engineers talk in terms of damping force in compression and rebound at a damper velocity. You need both. Once you derive your theoretical damping coefficients, you will need the actual damping force numbers to

talk to your damper supplier and arrive at a valve code to put in the shock. Below is a “Damping Calculation Roadmap” to guide you from vehicle information through damping coefficient calculation to actual damping force numbers. See Figure 15

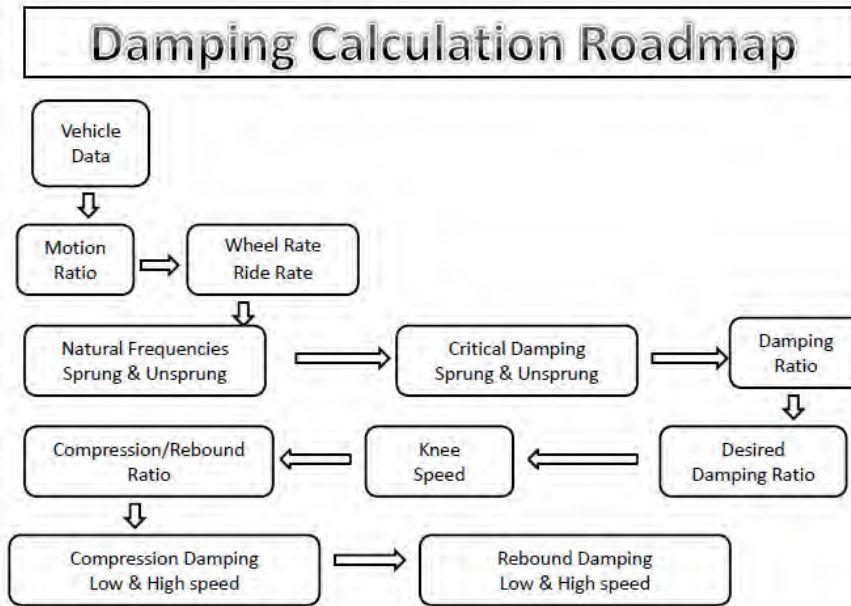


Figure 15

The Damping Calculation Roadmap is part of my damping calculation seminar. Due to space limitations, I will not include the entire calculation sequence in this book. Please go the following link to download the complete seminar presentation. <http://www.kaztechnologies.com/downloads/seminar-downloads/> Click on the download link for the 2011 FSAE Michigan Damping Calculation Seminar.

There are several elements of damping calculations that need to be discussed before moving on to tuning dampers on the vehicle. First, critical damping, which is defined as the level of damping that allows a mass to return to steady state most quickly with no overshoot. The equation is shown below. After you gather your vehicle information and calculate the sprung and unsprung natural frequencies, calculating the critical damping coefficient is next. As the definition implies, this is the amount of damping required to stop movement as quickly as possible.

Critical Damping Coefficient

$$C_{cr} = 2\sqrt{K_S * M}$$

$C_{cr}$  = Critical Damping Coefficient (N-s/m)

$K_S$  = System Spring Rate (N/m)

$M$  = System Mass (kg)

The next calculation is damping ratio. This is the ratio of the damping coefficient to the coefficient at critical damping. Think of this as a damping rate. The equation is shown below.

## Damping Ratio

$$\xi = \frac{C}{C_{cr}}$$

$\xi$  = Damping Ratio (N-s/m)

$C$  = Damping coefficient (N-s/m)

$C_{cr}$  = Critical damping coefficient (N-s/m)

Figure 16 below illustrates vehicle motion for various damping ratios. Note that at a zero damping ratio, or no damping, the vehicle will continue to resonate. This is only true if there is no friction. At a damping ratio of 1, or critical damping, the vehicle does not resonate at all. This condition is overdamped and will result in increased tire force variation and loss of tire adhesion. At a damping ratio of 0.2 the vehicle exhibits motion, but the motion damps in 1 ½ to 2 cycles. This level of damping lets the vehicle move, but controls the sprung mass.

Figure 17 illustrates typical damping ratios for various vehicles. Note a passenger car typically has a damping ratio of 0.2. This ratio is chosen for a combination of comfort and stability. Non-aero racecars, or those without ground effects, typically use damping ratios between 0.5 and 0.7, while racecars using ground effects aerodynamics typically have ride damping ratios near 1.0, and roll ratios as high at 9.0!

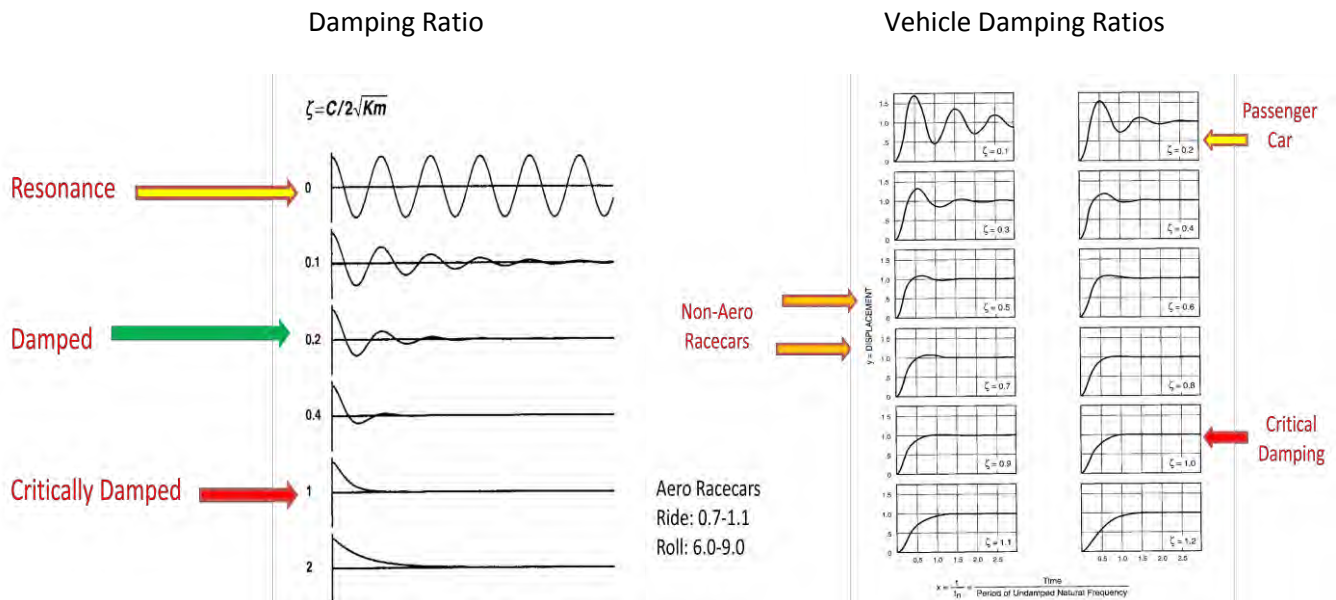


Figure 16

Figure 17

What the above examples illustrate is there is no one correct damping ratio. Damping ratio is just a number, like spring rate, ride frequency and tire pressure. The “correct” damping ratio is the one that makes your race car go fast.

In reality, the damping curve of an actual damper is composed of at least four damping ratios or damping rates. These are typically low speed compression, low speed rebound, high speed compression and high speed rebound, as shown in Figure 18 below. The low speed ratios control the low frequency motions of the sprung mass (heave, pitch and roll), as well as the rate of weight transfer in transient maneuvers (braking, turning and acceleration). The high speed ratios

damp the high frequency inputs and keep the suspension in control over bumps. A key point to remember is that dampers do not control weight transfer. They control the RATE of weight transfer.

### Low and High Speed Damping Ratios

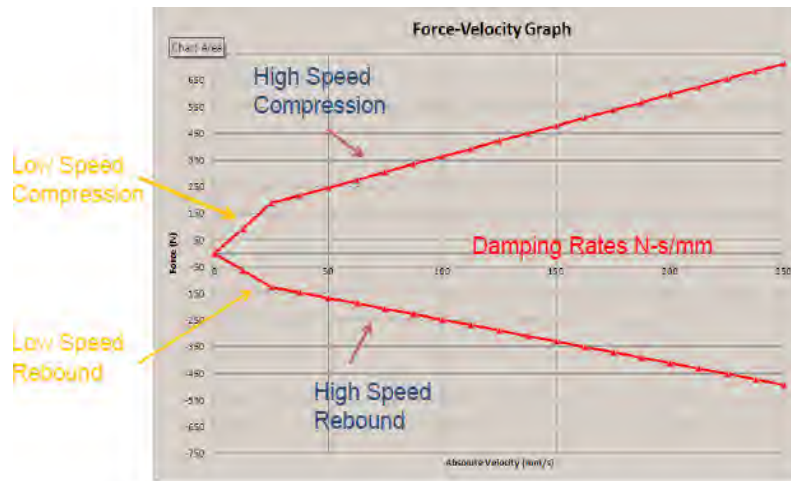


Figure 18

The other ratio to consider is the ratio of compression to rebound damping. Most text books recommend a ratio of three to 1, rebound to compression. This is the typical ratio seen on most passenger vehicles. This is due to the fact that the sprung mass is significantly heavier than the unsprung mass, and primarily controlled by the rebound, while the initial movements of the unsprung mass are controlled by the compression. See Figure 19 below.

### Compression to Rebound Ratio

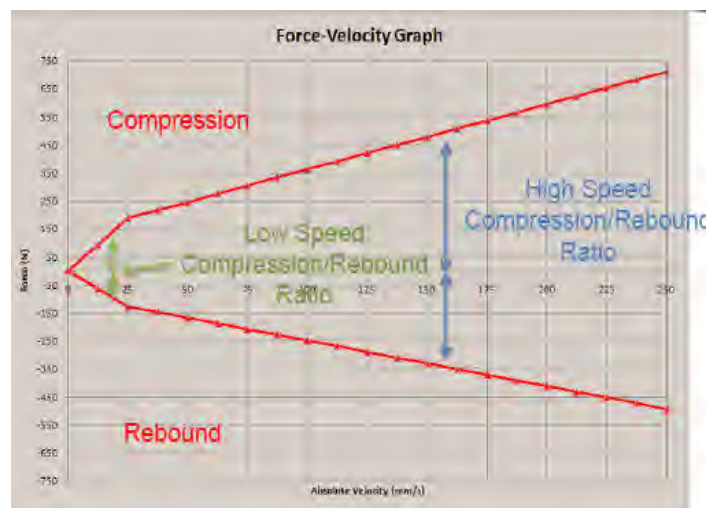


Figure 19

However, the use of dampers in race cars is much different than in passenger vehicles where dampers are primarily used to damp the sprung and unsprung masses. In race cars dampers are a key component of suspension tuning. As mentioned above, dampers control the RATE of weight transfer in transient maneuvers. During these events we are making use of the unique characteristic of the damper, the fact that it develops force proportional to velocity, not displacement like springs. In this respect, the damper acts like a dynamic spring, developing force when in motion during transient maneuvers. Thus, you can think of the damping ratios as dynamic spring rates!

## Typical Damping

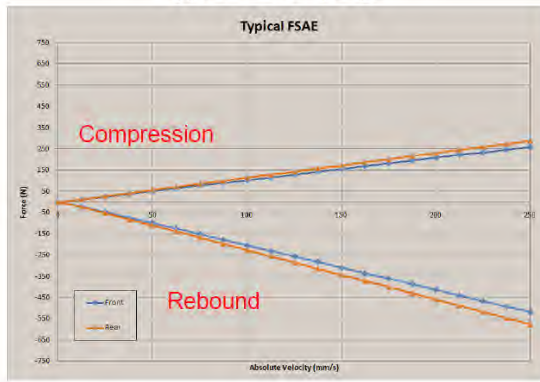


Figure 20

## Compression Biased

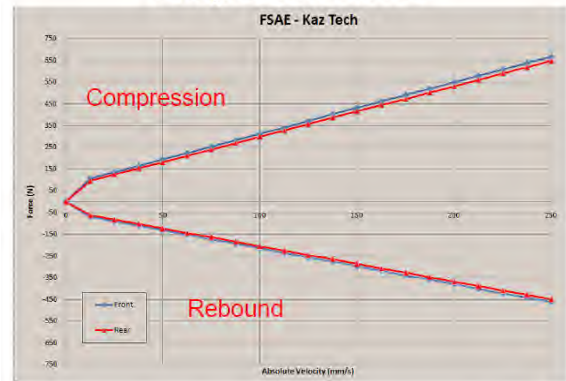


Figure 21

Figure 20 shows a graph of “traditional” damping ratios with a rebound to compression ratio of three to one. Figure 21 shows a graph of a compression biased damping curve. Which is “correct”? Again, it depends on your car, the suspension design and setup.

For FSAE cars I prefer the compression biased damping on both the front and rear. This damping characteristic makes use of the compression damping as a dynamic spring to quickly load the outside tire during turn-in and quick slalom maneuvers. With this damping the car feels nimble and very responsive. However, this is only one approach, and must be properly married to the suspension geometry, springs, anti-roll bar rates and other suspension adjustments.

### Damper Testing Prior to Vehicle Testing

Before you begin to tune the dampers on the vehicle you must make sure they are working properly. This is done by testing the dampers on a shock dyno. Figure 22 illustrates a proper force-velocity graph, showing both the acceleration and deceleration portions of the stroke. You will note the curve is very smooth, showing no incongruities, lag or cavitation. The fact that the acceleration and deceleration portions of the curve are very close together indicates very little hysteresis. If someone tests the damper for you, ask for a force-velocity graph showing the acceleration and deceleration stroke. Otherwise you will not be able to see the issues shown below.

### Proper Damping Curve

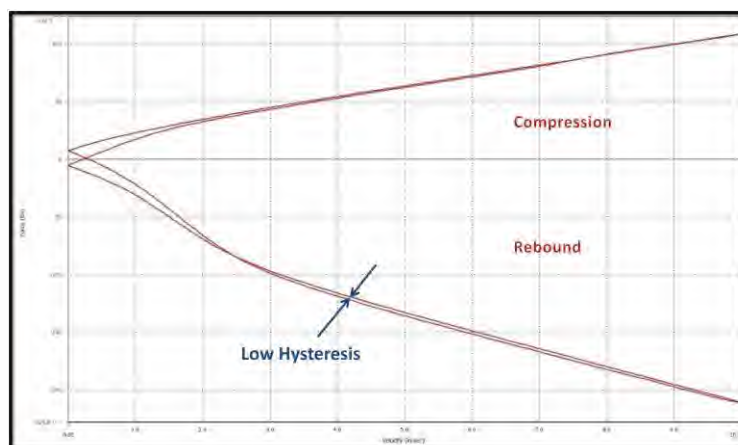


Figure 22



Figures 23 and 24 illustrate issues with the damper. Figure 23 shows incongruities in the force-velocity curve. This could be caused by a bent rod or a problem with the valving. Figure 24 illustrates cavitation and lag. Note the extreme hysteresis and loss of damping force in rebound and compression. This could be caused by low gas pressure, loss of oil or a problem with the valving. Both these issues require disassembly and proper servicing of the dampers.

### Incongruity

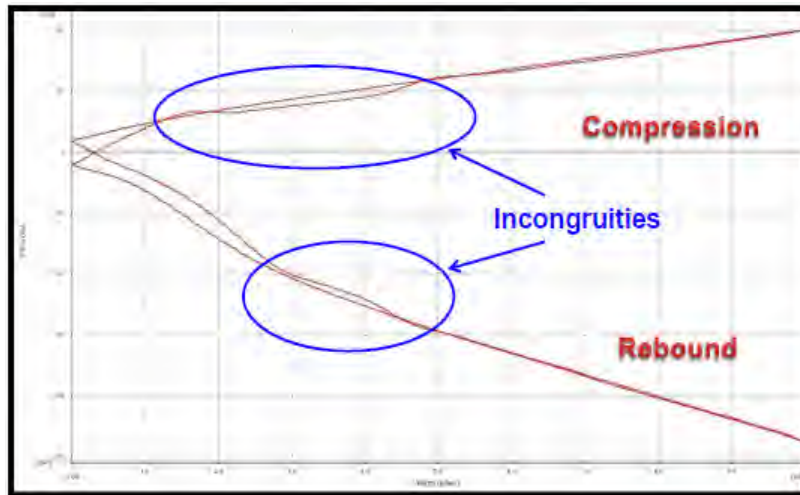


Figure 23

### Cavitation and Lag

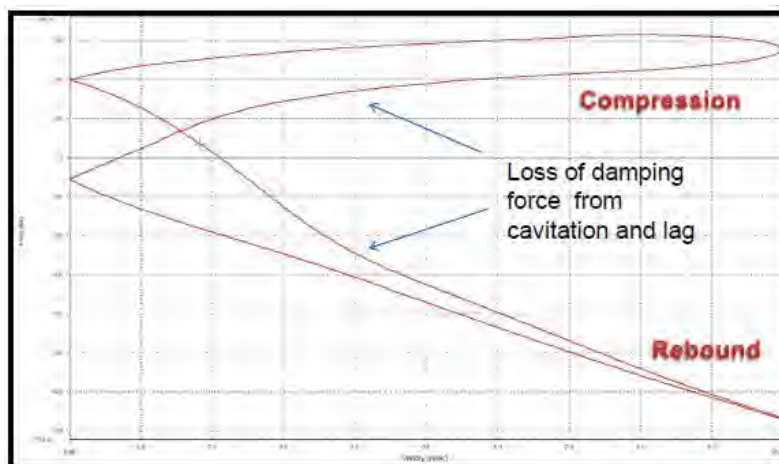


Figure 24

Once each of the dampers is performing properly, you want to make sure that the damping forces match for a pair of dampers. In other words, the damping forces for the left front and right front should match at the same setting. The same should be true for the left and right rear. Figure 25 illustrates a properly matched pair of dampers, while Figure 26 shows a pair that do not match at the same settings.

## Force Matched Dampers

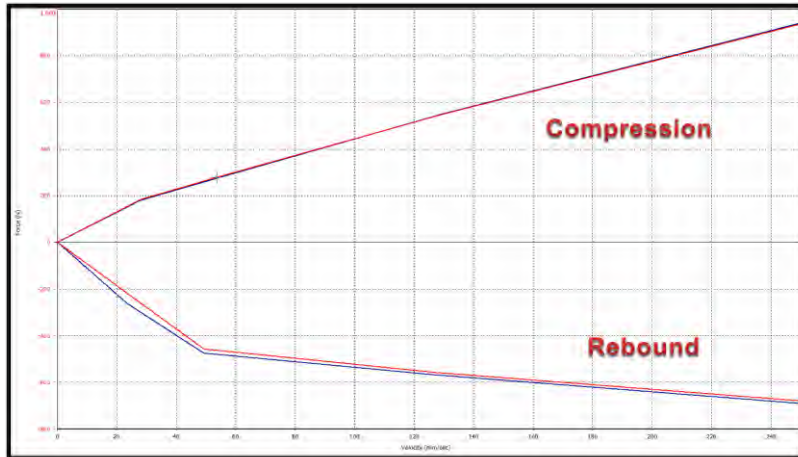


Figure 25

## Dampers that are not force matched at the same settings

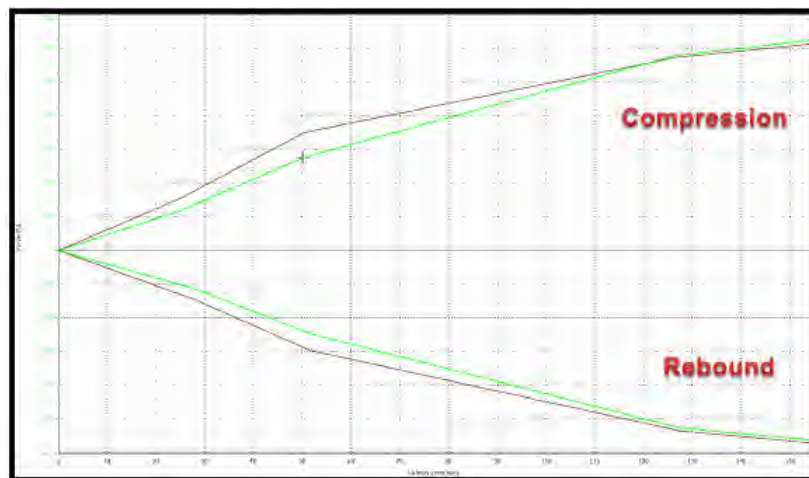


Figure 26

If the damping forces for a pair of dampers do not match at the same settings, you can try tuning the adjustments to get them to match. If this is the case, make sure you record the differences. Also, be aware this may not be the same through the range of adjustments.

You also want to know the range of adjustment and the change in damping force for each increment of adjustment. Figure 27 illustrates a dyno test for a full range of damper adjustments. Note that each adjustment does not change the damping by the same amount. You need to have this information to know how much you are changing the damping when you make an adjustment. Again, note that the change is different if you are at low damper settings or high damper settings. So you need to know where you are in the adjustment, and how much an increment of adjustment will change the damping from that setting.

## Damping Adjustment Range



Figure 27

### Damper Fit Check

The final piece of damper preparation is to make sure the dampers fit properly on the vehicle. First verify that the lengths you have chosen give you the desired wheel travel. To do this, install the dampers on the vehicle without the springs. With the chassis on stands move the suspension through its entire range of travel. Be sure there is no binding or interference between the shock and the chassis through the entire travel. Pay particular attention to the attachment points. Be sure there is clearance between the damper eyelets and the mounting tabs at all times.

Next install the springs on the dampers. I suggest installing the springs with zero preload, and setting the ride height by adjusting the length of the push or pull rods. Installing the springs at zero preload minimizes the spring force on the dampers. This will result in the least amount of spring and shock friction due to spring load. I recommend using thrust washers on both ends of the springs. This will also reduce spring friction. If your suspension uses a direct acting damper, you must set the ride height by moving the height of the spring perch.

Next install the dampers on the car. With the vehicle at race weight (driver, fuel and fluids in the car), adjust ride heights using the push or pull rods. Be sure the dampers are positioned at mid stroke at final ride height. This should give you sufficient compression and rebound travel. If you find the dampers are bottoming or topping during vehicle testing, you can change the damper position at ride height to eliminate the problem. With the suspension at ride height, measure and record the gap between the damper body and the jounce bumper. This measurement is called the packer gap. You will want to know this to see if the damper is contacting the jounce bumper at anytime on the track.

### Data Acquisition

Before you begin vehicle testing, make sure you set up your data acquisition system to properly look at damper response on the vehicle. The first item is damper displacement. Make sure your displacement sensors directly measure damper displacement. If not, create a math channel for proper damper displacement. Be sure the acquisition rate for your displacement sensor is set at a minimum of 250 Hz, preferably 500 Hz. Acquiring at a rate less than 250 Hz will result in substantial error. In fact, at 100 Hz there will be nearly 150% error in your displacements and velocities. Also, be sure you do not filter the raw displacement channel before calculating velocity. This will also produce a substantial error.

I also create all my damper analysis channels prior to testing. The first are unfiltered damper displacement for all four corners. I make sure these are the raw signals from the damper displacement sensors, or the calculated damper displacements from the raw signals. Then I create unfiltered damper velocity channels by differentiating the raw damper displacements. Next I create a second set of filtered damper displacement and velocity channels. These are the channels I filter for various types of analysis. By having a set of unfiltered displacement and velocity channels I am able to look at actual displacements and velocities at any time. This speeds up my analysis and prevents me from misinterpreting data in the heat of the battle.

### **Vehicle Damper Tuning**

In this section I will only address damper tuning on the vehicle. I highly suggest you read Steve Lyman's section ***Chassis Development and Suspension Tuning*** which addresses tuning the entire suspension.

There are four phases of damper tuning on the vehicle: Low frequency heave and pitch balance; Wheel control over bumps; Handling; and Testing rebound and compression adjustments.

The primary job of the dampers is to properly damp the sprung and unsprung masses. Since the unsprung mass is the largest, heaviest and most difficult to control, that's where I begin. The upper portion of Figure 28 is an illustration of a low frequency hump.

Low Frequency Hump and High Frequency Bumps

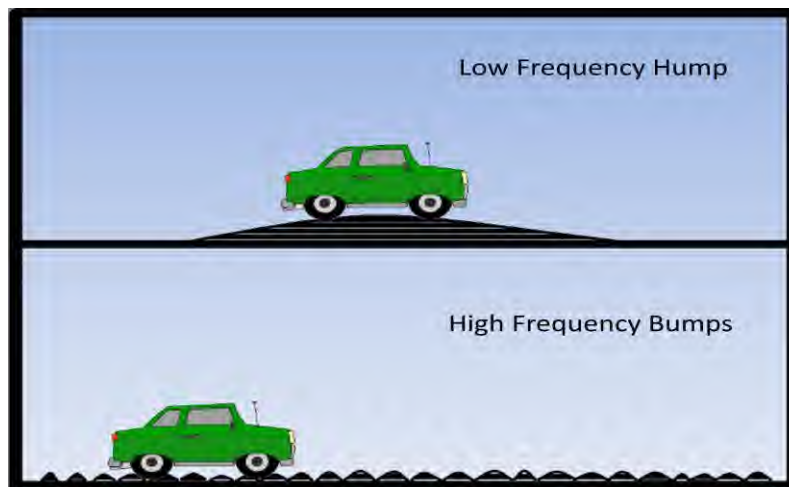


Figure 28

Find a hump that will make the body or the sprung mass oscillate at the sprung mass resonance or natural frequency. Set all damper adjustments to full soft. Then drive the car over the hump at increasing speed until you see the car body move up and down. You want the car oscillate 1 ½ to 2 cycles before the motion stops. If this is not true, check the suspension for binding or excessive friction. You may have friction controlling vehicle motion instead of your dampers!

Now begin adjusting your dampers to balance and damp the car. First you want to minimize the pitch, and then tune to damp the heave. Watch the response of the vehicle after the hump to observe which end of the car is moving the most. You can also observe this by looking at the wheel or damper displacement plot for one side of the vehicle. Typically the rear will be moving more than the front since the rear has the higher mass and wheel rate. Figure 29 shows left side damper data from a car that is underdamped. You can see the whole car oscillating, and the rear moving more than the front.

## Underdamped Oscillations

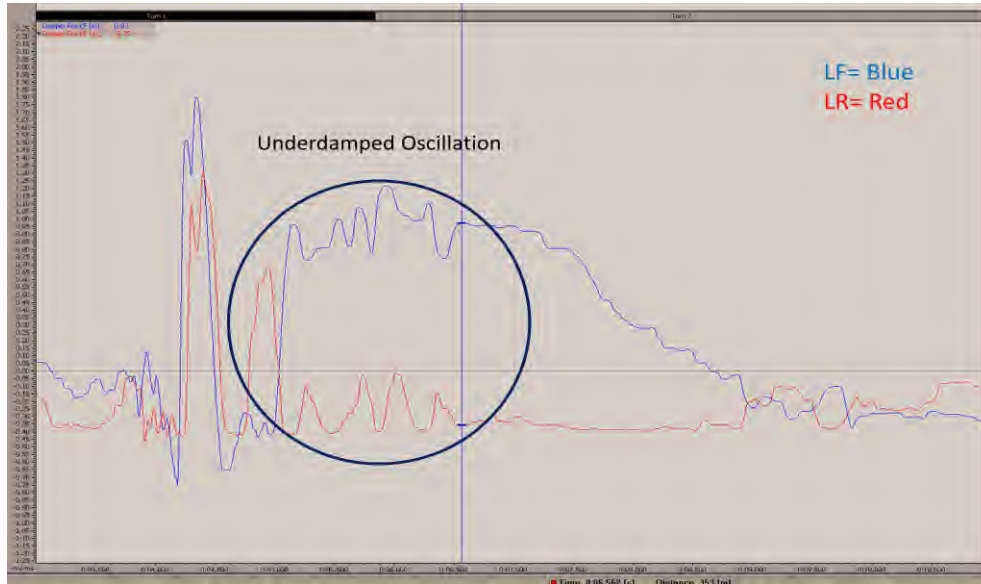


Figure 29

I begin tuning by adjusting rebound. I start with the end of the car that is moving the most. Look at your damper adjustment range (see Figure 27). Make small but significant adjustments to reduce the oscillations and get the car to heave not pitch. After several rebound adjustments, try adjusting compression. You want the car to move in compression and rebound. Figure 30 is an illustration of front and rear displacements for a damped vehicle. Since the rear of the vehicle hits the bump after the front, it must oscillate at a higher frequency than the front for the car to transition from pitch to heave. Figure 30 illustrates this. Tune the dampers until you achieve 1 ½ to 2 oscillations. At this phase in the tuning it is better to be less than more damped. As long as the sprung mass is under control, less damping will give you less tire force variation and better tire grip. Typically more damping will increase tire force variation and reduce grip. As far as tire force variation or grip is concerned, less damping is better.

## Balancing Front and Rear Damping

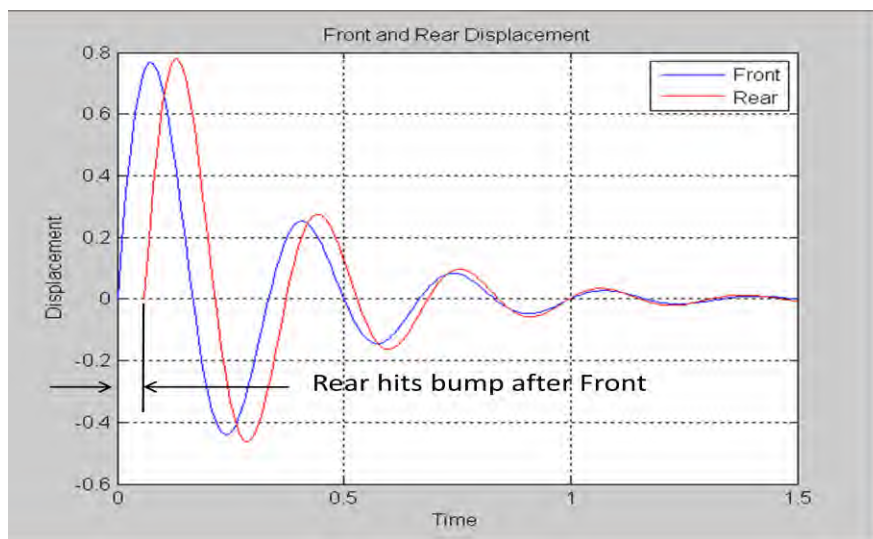


Figure 30

## **Wheel Control over Bumps**

Next you want to make sure the wheels stay in control over bumps. Find a rough section of pavement that will make the wheels resonate or bounce. Set the dampers to your heave/pitch balance settings. Drive the car over the rough section at increasing speed. Watch to see if the tires bounce or go into resonance. The driver may also complain of lack of grip on the end of the car in resonance. Again, you can observe this visually and by looking at damper displacements in the data analysis program. Note the wheels may already be in control. If you don't see the tires bounce, move on.

Now here's where it gets a bit tricky. To find the best wheel or sprung mass damping you will want to try more AND less damping for both compression and rebound. If you have high speed adjusters change these first, then the low speed adjusters. If the wheel is oscillating, start by adding compression control. Observe the response visually and using data. Next set you compression back to your initial setting and add rebound. If adding compression or rebound makes the tire visually bounce more on the bumps, try reducing compression and rebound damping. Be aware that as you add damping the damper displacement data traces will show a reduction in displacement. This is because you are restricting the motion of the suspension and thus the wheel. This is why you want to visually observe the tire. Too much damping will "lock up" the suspension, and you will bounce on the tires! When this happens you will see fluctuations in the RPM data trace, indicating the tires are losing grip and traction.

Once you find the best settings for wheel control, rerun the low frequency heave and pitch balance test to make sure the sprung mass is still in control. This is especially true if you have reduced damping for wheel control. If you added damping for wheel control at one end of the car you may have to tune the other end to reduce pitch.

## **Handling**

The dampers can be a key part of your suspension tuning tools. Unlike springs and anti-roll bars that generate force with displacement, dampers generate force with velocity. This gives them two unique properties that are highly useful; 1) They control the RATE of weight transfer by the springs and anti-roll bars, and 2) They work like dynamic springs. When controlling the rate of weight transfer the dampers are timing devices, delivering or removing load at the right time in the corner to maximize the grip of the tires. As dynamic springs, the dampers momentarily work like springs, exerting force based on velocity instead of displacement.

With these properties the dampers are very influential to the handling characteristics of the vehicle during transitions. These transitions include braking, turning into the corner, transition from turning to acceleration, and acceleration. Figure 31 breaks down a corner into several sections. Let's discuss what influence the dampers have on handling in each section of the corner.

## Handling Corner Sections

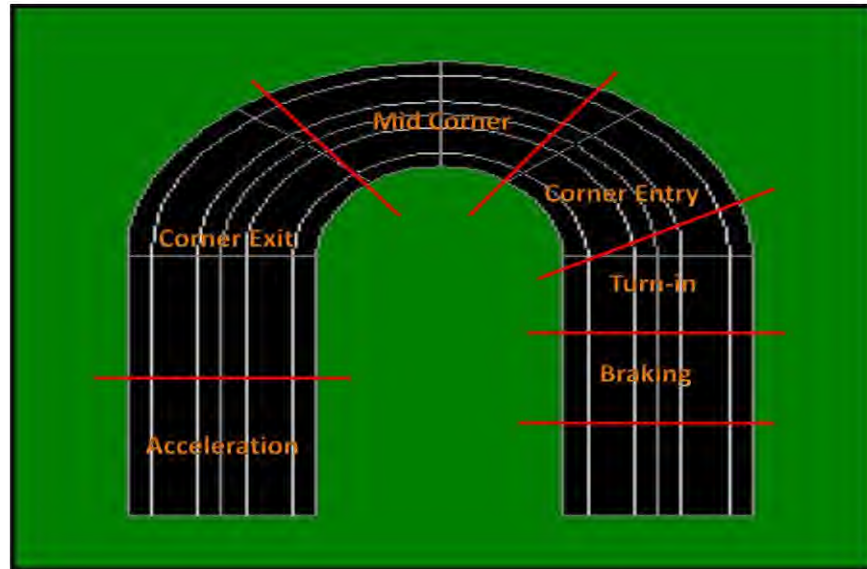


Figure 31

### Braking

During the braking portion of the corner, the dampers control the rate of weight transfer from the rear to the front tires. If the weight transfer is too fast (typically too little front compression), the front tires will be unable to absorb the rapid increase in load and will lose grip. If the weight transfer is too slow (too much front compression or too much rear rebound), braking will not be optimal and the driver may have to delay turn in to the corner. Also, the wrong amount of rear rebound damping can make the car unstable during braking. Too little, and the rear of the car will pop up when the brakes are applied. Too much rear rebound will tend to lift the rear tires off the ground and the rear tires will lose grip.

### Turn-in

When the car enters this portion of the corner the downshifting should be completed and the majority of the braking is done. If trail braking is used it will primarily take place in this portion of the corner. This is where the initial turn in to the corner takes place. The driver wants the car to respond to his steering input without hesitation.

During the turn-in portion of the corner you can use the compression of the outside wheel dampers as dynamic springs. While the springs and sway bar require an amount of displacement to load the outside tire, the dampers only require velocity. Using the dampers you can quickly load the outside tires as soon as the car begins to roll. This will give the car good turn in response. Again, you must find the correct amount of damping. Too much will overload the tire producing understeer. Too little and the car will be lazy on turn in. Also, the damping must be correct for the springs and anti-roll bar stiffness.

The dampers are also controlling the rate of weight transfer, particularly rear rebound. If there is too much rear rebound it will try to lift the rear tires, producing oversteer on turn in.

### Corner Entry

During this portion of the corner the car continues to transfer weight to the front from the rear and to the outside wheels. The dampers are used as timing devices to control the rate of weight transfer. I primarily use the compression at the outside wheels to control the rate of weight transfer; less compression to speed the rate, more to slow the rate of weight transfer. You can also use the dampers to control the rate of roll, and the phase between front and rear. Again, I

start by using compression at the outside wheels. If I start to lose grip but need more control, I add rebound to the inside corners of the car.

### Mid Corner

If there is a steady state portion of a particular corner, this is it. The car has reached equilibrium and will corner based on front and rear roll stiffness. Since the suspension is essentially not moving (except for bumps) the dampers have no effect on handling during this phase of the corner.

### Corner Exit

This is the portion of the corner where you are transitioning from turning to acceleration. Here you will be transferring weight from the front to the rear of the car, particularly from the inside front to the outside rear. Again, you can use the dampers as dynamic springs. You can use rear compression to support the rear of the vehicle during this transition phase. However, too much rear compression will act like too much rear spring and cause the car to oversteer. The dampers are also controlling the rate of weight transfer from the front to the rear tires, particularly the inside front to the outside rear.

There are several scenarios to consider during this phase of the corner. Let's start with the car oversteering, which is the most common in this phase. If the car is oversteering, the first adjustment I make is to soften the rear compression. This will reduce the dynamic spring rate of the rear dampers and speed weight transfer to the rear. Reducing front rebound will also increase the rate of weight transfer to the rear. If decreasing rear compression control makes the situation worse, increase rear compression. If neither of these work, try a spring or anti-roll bar change.

If the car is understeering during this portion of the corner, typically the weight is being transferred too quickly away from the front tires, causing them to lose grip. To slow this transition, we can increase the front rebound or the rear compression. My first adjustment is to increase front rebound. If I need more, I increase rear compression. However, increasing rear compression will increase the dynamic spring rate of the rear dampers and may produce oversteer. Also, too much front rebound may try to lift the front wheels off the ground producing understeer.

### Acceleration

During this phase of the corner, like the braking phase, the dampers are controlling the rate of weight transfer from the front to the rear tires. You want to transfer the weight fast enough to maximize traction but not too fast to break the tires loose. I start with rear compression to control the weight transfer. Remember compression in the rear dampers also works like a dynamic spring, so more rear compression may produce loss of traction. If this is the case, try increasing front rebound to control the rate of weight transfer.

Since this section is about damping, I did not address all aspects of suspension tuning. I suggest you refer to ***Breaking Down the Course*** in Steve Lyman's ***Chassis Development and Suspension Tuning*** section of this book for a more in-depth look at suspension tuning.

### Testing Rebound and Compression Adjustments

At some point in your testing you will want to turn all the knobs on the dampers to see what they do to the vehicle handling, and what the driver says about the changes. I start with my "best" settings to date, and then adjust each setting firmer and softer in two steps. This looks like starting with front compression and adjusting it half way to full firm. Next I change it to full firm. Then I go back to the base setting and let the driver re-baseline the car. Then I soften the front compression in two steps, and then back to baseline. From there I move on to front rebound, rear compression and rear rebound. I do not tell the driver what I am adjusting. I ask them to just tell me what each change does to the car, or if it does nothing at all. Although going through this routine requires time and patience, it will pay off later when you are fine tuning the car.

### Comments About Damper Adjustments



Dampers are not the big knobs for suspension tuning. If you make multiple big changes to the dampers and this has little or no affect on handling you are changing the wrong thing! You need to first dial in springs, anti-roll bars, camber, toe, etc. Dampers are the fine adjustment. Make sure you get the springs and anti-roll bars right for steady state handling before tuning the transitions with the dampers.

When you are making damper adjustments, don't be afraid to make big changes. A rule of thumb is to change the damping force 20% at 50mm/sec (2 in/sec). This is why you need to know your damper adjustment range (Figure 27) BEFORE you start vehicle testing. Also, try more and less of a given adjustment (i.e. front compression) before trying another adjustment.

### **The Care and Feeding of Your Dampers**

All the information I have presented is useless if your dampers are not working properly. Dampers must be maintained like any other part of the race car. As discussed in the ***Damper Testing*** portion of this section, the only way to know if your dampers are performing correctly is to test them on a damper dynamometer. Look at a full cycle of the force/velocity graph (Figures 22-24) to be sure there is no evidence of lag or incongruities. Also, make sure the damper forces match (Figures 25-26) on each end of the vehicle.

The objective of damper care and feeding is to keep the dampers working properly. What you want is dampers that are repeatable and reliable. I recommend dyno testing your dampers at least once a year, and I encourage you to test them before each major competition. It is nearly impossible to diagnose a "bad" damper on a car. It is also nearly impossible to get even good performance from a car with a bad damper. Check the gas pressure in the dampers before each day of testing or competition. Low gas pressure is the major cause of loss of performance in dampers. Look for oil leakage from the dampers. There will always be a slight film of oil on the rod. This is to keep the rod seal lubricated. However a puddle of oil under the damper indicates a leak and means you need to have the oil seal replaced.

### **Last Word**

There is a wealth of information on our website at [www.kaztechnologies.com](http://www.kaztechnologies.com). There is technical information about all our products as well as damper dyno data. There are Tech Tips to help you understand the workings of the damper. You can also download copies of all our technical seminars. If you have questions, please e-mail them to [fsae\\_shock@kaztechnologies.com](mailto:fsae_shock@kaztechnologies.com).