FSAE Damper Project

Objective: *To design and build 2 sets of dampers (8 in total) for a FSAE racecar, that will perform better than any available off-the-shelf piece, and run them at the FSAE competition in May 2006.*

***Note**: Values specific to the 2006 MRacing FSAE vehicle have been excluded from this report to protect the competitive interests of the team.*

Abstract

Formula SAE is an international collegiate design competition sponsored by the Society of Automotive Engineers in which students design and build a small, formula-style racecar. Because of power-train regulations, cars that exhibit the best handling characteristics have a distinct advantage. Utilizing a short-long arm suspension, most cars use dampers designed for mountain-bikes. Because dampers play a pivotal role in vehicle handling, and the mountain bike dampers used on previous M*Racing* vehicles are not well suited for our vehicle, we feel that a significant performance advantage can be obtained through constructing a custom set of dampers designed specifically for our car.

Problem Description and Introduction

Our team intends to make two full sets of dampers for use on the 2006 M*Racing* vehicle which will compete in the annual Formula SAE competition in May 2006. Formula SAE (FSAE) is an international competition which started in 1981. The University of Michigan has been represented in this competition every year since MRacing was established in 1989. The Michigan team has been very successful over the years, with numerous top 10 finishes and an overall victory in 1994. More recently, the team finished 4th overall in 2002. In 2005 Michigan was featured in a Road and Track event for the 5 fastest cars in 3 dynamic events.

FSAE has become a truly global event, a total of 8 competitions will be held in 2006. As a result, teams are constantly trying to develop new and innovative ideas to get an advantage over their competitors. The FSAE competition is built around a commercial premise designed to better simulate an engineering project. Teams are asked to design, build and market a single seat formula style race car for the "weekend" auto-crosser. Auto cross competition involves navigating a course made up of cones, usually laid out in a large parking lot, in which the fastest single lap time wins. Consequently, the typical FSAE car is light, weighing less than 300 kg (including driver). Also, the cars are small and must be easily maneuverable.

Because the competition thrives on innovation, the rules are written to allow students to explore many different possible solutions to the problem at hand. The main focus of the rules is to ensure the safety of participants and spectators alike. This places emphasis on the handling ability of the vehicle, as the power-train rules are somewhat restrictive in the name of safety. Simply put, the car that handles the best is the fastest. In general, FSAE cars have a short-long arm (*SLA*) (*reference glossar*y) suspension layout and utilize mountain bike dampers.

The role of the racecar's suspension is to keep the tires in contact with the road surface and at an orientation that allows them to generate the most lateral load from corner entry to corner exit. The dampers control the vibration induced by the road surface and the lateral and longitudinal weight transfers of the vehicle. Finding an appropriate damper solution is very difficult for FSAE teams. The cars are so light that an automotive styled damper is too large. There are dampers especially designed for racing, however these are still too large and often times are prohibitively expensive. The mountain bike dampers, which have become a virtual standard on all FSAE cars are preferred because of their light weight and low cost. Because dampers play such a pivotal role in the overall handling characteristic of the vehicle, teams have started to explore alternative solutions. Several teams have run heavily modified versions of commercially available mountain bike dampers. In 2003 Cornell University used student design/built dampers and in 2005, the University of Western Australia ran an innovative hydraulic suspension system that not only provided damping control, but also replaced the traditional mechanical stabilizer bar found on virtually all FSAE cars.

By designing and building custom dampers we hope to achieve a higher vehicle performance threshold. Specifically, we aim to reduce the amount of time our vehicle spends in transient conditions for a given lap, as well as better utilize the tires through better control of the vehicle in these situations. Tires, by nature, do not produce as much force when subjected to changing conditions (transients) as they do under normal steady state conditions. Thus, lap times will decrease if transients can be kept to a minimum.

Benchmarks and Information Sources

Background Information

From the invention of the automobile the necessity of an automotive suspension has been wholly recognized, protecting the chassis and occupants from bumps, and controlling the vehicle through its maneuvers. The use of springs as energy absorbers resulted in a system very capable of storing the input energy from bumps and accelerations providing an effective solution to the issues facing a vehicle in its everyday environment. Unfortunately, these spring were also prone to releasing this energy in an un-controlled (though predictable) fashion. Enter the automotive damper, commonly referred to as the shock absorber. Dampers are especially important in high performance applications such as racing, where cars are pushed to the edge of their capabilities.

Currently, nearly all forms of automotive racing (from Formula 1 to motorcycles) require the use of shock absorbers to dampen the movements of the suspension and the chassis. Essentially, all dampers function similarly: energy dissipation due to fluid resistance. However, dampers vary significantly in behavior and construction for different applications; a mountain bike damper sees entirely different inputs than a damper on a LeMans racecar. A mountain bike damper needs to be lightweight (especially considering complete bikes weigh as little as 10lbs). This damper, in general, sees two kinds of input: a bobbing motion, due to the riders pedaling (high amplitude, low frequency) as well as large impacts, such as a 5foot drop (high amplitude and high frequency). Conversely, a LeMans racecar requires incredibly stiff springs because they produce nearly 5000lb of downforce, which gets transmitted through the suspension. To get maximum efficiency from their aerodynamics, the cars are run with virtually no ground clearance and very little suspension travel to keep the tires in an optimal orientation. This means that the LeMans racecar damper sees primarily low amplitude and high frequency inputs. See Figure 0 on the following page for a diagram.

An FSAE racecar does not need to absorb massive impacts from jumping off cliffs, nor does it produce massive amounts of downforce, requiring very stiff springs and dampers. Formula SAE regulations stipulate that an FSAE car must have 2 inches of suspension travel (+\- 1" from static ride height). The suspension design determines how much travel the damper sees. FSAE cars also run relatively stiff springs to keep the car from contacting the ground and also to allow for very rapid changes in direction. An FSAE damper sees two kinds of input, low frequency low amplitude inputs due to weight transfers as the vehicle navigates the track, also high frequency low amplitude inputs from bumps and other surface imperfections that are typical of our racing surface. Thus, there is not an "off-the-shelf" option for a FSAE racecar. Any mountain bike damper must be dismantled and revalved to change its Force-Velocity curve; any current racecar damper is too large to fit inside the compact chassis. Therefore, a compromise must be made for this application.

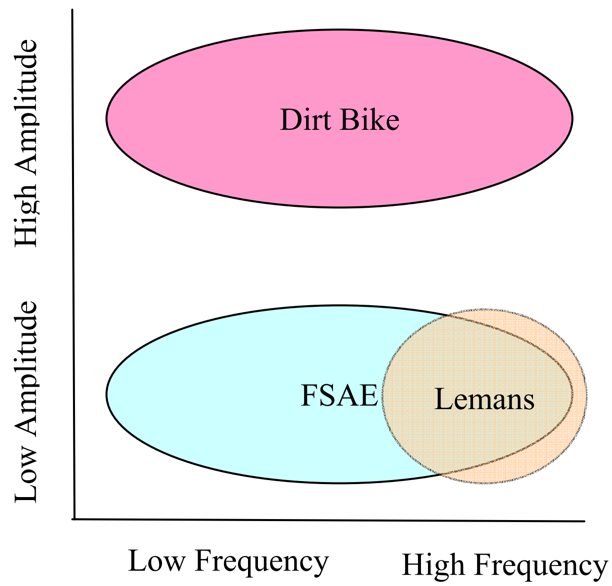


Figure 1: Frequency/Amplitude input ranges for dampers used in Dirt bikes, FSAE and Lemans respectively.

Existing Solutions:

One benchmark product for this project is the Fox Vanilla RC (see Fig. 2), a small (8" long) damper used on the rear wheel of mountain bikes. With its light weight and relative affordability, the Fox has been a very popular shock for FSAE racecars, where weight and packaging produce serious constraints on shock absorber choices. University of Michigan's FSAE has used the Fox Vanilla damper on the previous 6 racecars, so its benefits and limitations have been well documented. Weighing only 0.28 kg, the Fox is one of the lightest shocks available for a FSAE vehicle, and its compact size can be fitted in numerous suspension configurations. Furthermore, the Fox offering appeals to FSAE teams because of its low cost, just $250 per damper. However, the Fox was designed for use on a mountain bike, which (as discussed earlier) is an entirely different design criteria than a damper intended for use in motor sport. There are several performance issues present, which have compelled the FSAE team to seek alternate options.

The Ohlins ST44 damper (see Fig. 3) is one that is becoming more popular among FSAE teams. Its major selling point over the Fox damper is that it was designed with motor racing in mind. Thus, out of the box, it is much more suited to an FSAE application. This damper is significantly more expensive than the Fox damper, at $430 per damper. These dampers are growing in popularity because they are much more consistent than the Fox offering. Furthermore, they are easily modified internally.

The Penske Racing Damper 8760 series damper (see Fig. 4) is very popular among professional race teams; however it is far too expensive for the average FSAE outfit. At $900 a piece, one of these dampers is more expensive than our entire chassis budget for one year. They are virtually infinitely re-configurable internally; Penske will supply replacement pistons and shim stacks that can achieve virtually any desired force vs. velocity relationship. They also have 4 external adjustments, where most other dampers available only have 2. This gives the ability to independently adjust both high and low piston speed damping in both compression and rebound movements. The Penske dampers do offer a significant performance advantage, however the high cost makes them un-obtainable.

Fig. 2: Fox Vanilla RC

Fig. 3: Ohlins ST44

Fig. 4: Penske 8760

Issues and Solutions
Since mountain bikes only have one rear shock (the front is integrated into the fork), there is little obligation for a manufacturer to ensure consistency between models, meaning that one shock will likely behave differently than another. This is unacceptable for a FSAE vehicle, where differences between suspension sides will cause erratic behavior. Manufacturing discrepancies are unavoidable, but individual tuning of each shock with respect to a common target will eliminate the inconsistency between pairs. A large company may decide this is unnecessary for a mountain bike application, but it is required for these prototype dampers.

While the small size of the Fox Vanilla is a great benefit in terms of weight and packaging, there are significant drawbacks to using a small (21mm) piston. Friction is generally inversely related to the diameter of the piston bore, primarily due to the relationship between pressure and force, $P = \frac{F}{A}$. A larger piston increases the swept area, which means that the shock can have a lower internal pressure to achieve the same force as before. Lower internal pressures allow less radial force on the seals, which reduces the overall friction. High hydraulic pressures also put more load on valving components and static seals (O-rings), which increases the probability of failure. The prototype dampers will contain a 32mm piston bore allowing significantly lower internal pressures.

Effects such as fluid compressibility (lag) and acceleration sensitivity are largely dependent on the environment. Typically small stroke dampers with deflecting disc valves do not have significant acceleration sensitivity (where inertia can open up a valve reducing the damping force) because the inertia of the disc valves is very small. Fluid compressibility causes a lack of force during direction changes, and its impact is based on the shock oil used.

Quality Function Deployment

As a racecar team, the ultimate goal is to be faster than the competition and win races. However, FSAE is a unique combination of an engineering exercise, and a racing series. There are eight competitions throughout the year, but many teams can only afford to attend one. The design of an FSAE racecar is weighted heavily on reliability because there is only one race, and thus one chance for success, per year.

The requirements for a high performing FSAE damper are in conjunction with the team goals set in the beginning of the season. Our goals are to win every dynamic event by reducing the car's overall mass, and by completing the car earlier in the season to allow for proper testing, tuning, and driver training. The motivation to design and manufacture a custom damper is the culmination of years of research done by current and former team members. The following requirements illustrate the role dampers play on an FSAE racecar team:

- Completes the race season reliably
- Compliments the suspension geometry design
- Consistent performance
- More predictable vehicle control
- Externally adjustable
- Increased handling performance
- Light weight
- Internally re-configurable
- Low cost
- Survives unexpected force inputs
- Better ride comfort
- Chassis packaging
- Easily maintained

With a custom-designed damper, every parameter can be tailored to meet the car's needs. In addition, the mechanisms for internal and external adjustment are designed in-house, allowing the performance to be more predictable.

As a basic device, the damper is a hydraulic piston-in-cylinder assembly. The piston is actuated by the suspension, causing the fluid to flow throughout the damper body. A reservoir provides the fluid's static pressure. The reaction of the damper to the suspension input is vital to vehicle control, and is the result of the energy dissipation provided by the fluid. Our engineering specifications and targets follow the piston-in-cylinder baseline of geometric, fluid, and mechanical properties. Administrative specifications such as cost are also included in the following list.

1. Piston diameter (bore) – 32mm
2. Damper stroke – 50mm
3. Hysteresis – 25 N
4. Reservoir volume – 15000 mm^3
5. Eye-to-eye-length – 178mm
6. Shaft diameter – 8mm
7. Commercial value - $400
8. Distance between rebuilds – 500km
9. Total external adjusters – 2
10. Force vs. Velocity curve
11. Force vs. Displacement curve
12. Static gas pressure – 1 Mpa
13. Acceleration Sensitivity – 15N
14. Internal operating pressure – 5 Mpa
15. Material Costs - $107
16. Weight – 1.25 kg
17. Labor Costs - $115
18. (Dis)Assembly time- 60min/45min

Example Force vs. Velocity & Force vs. Displacement curves can be found in Appendix 4.

Utilizing QFD (Quality Function Deployment), we related the customer requirements to our engineering specifications (see appendix 1) to determine which were the most important. The above lists are in descending rank order of importance as a result of the analysis. The weight of each customer requirement was determined by comparing every combination one-to-one, and then tallying up the total (see appendix 1). Customer requirements were then correlated to the engineering specifications using four categories. Requirements and specifications that are highly related get a rating of “9”, a rating of “3” corresponds to somewhat related, “1” means slightly related and unrelated parameters received a “0” rating. For example, Eye to Eye length is strongly related to chassis packaging, and received a rank of “9”, while low cost and internal operating pressure are unrelated and received no rank. Results of how our three benchmark dampers perform to our customer requirements is also shown in the QFD diagram.

Concept Generation:
Global Layout
Throughout the roughly 100 year history of the automobile, the need to control the vehicle sprung mass motions has given rise to the shock absorber. During this time various solutions have been tried and tested, and for the most part there are two somewhat similar, yet distinct options that have survived the test of time. Today, most cars make use of either a mono-tube or twin tube damper. As the damper is compressed, the damper rod enters the cylinder proportionally with the piston displacement. Since the rod has a certain volume, the hydraulic fluid must compress or evacuate the cylinder during rod entry. Due to the low compressibility of typical damper oils, the fluid evacuates to a reservoir. When the damper is extended and the rod volume is removed, the fluid must re-enter the cylinder to prevent creation of vacuum. So long as a pathway to and from the reservoir is available, the pressure differentials between the cylinder and reservoir force the fluid in the appropriate directions. Twin-tubes and mono-tubes are simply two different methods to package the reservoir. Often used on production vehicles with McPherson style suspensions, the twin-tube damper contains the piston and cylinder surrounded by an external tube. As fluid is evacuated from the cylinder, it passes though a valve at the bottom of the cylinder and enters the external tube, which is pressurized by a gas (see Figures 4 & 5).

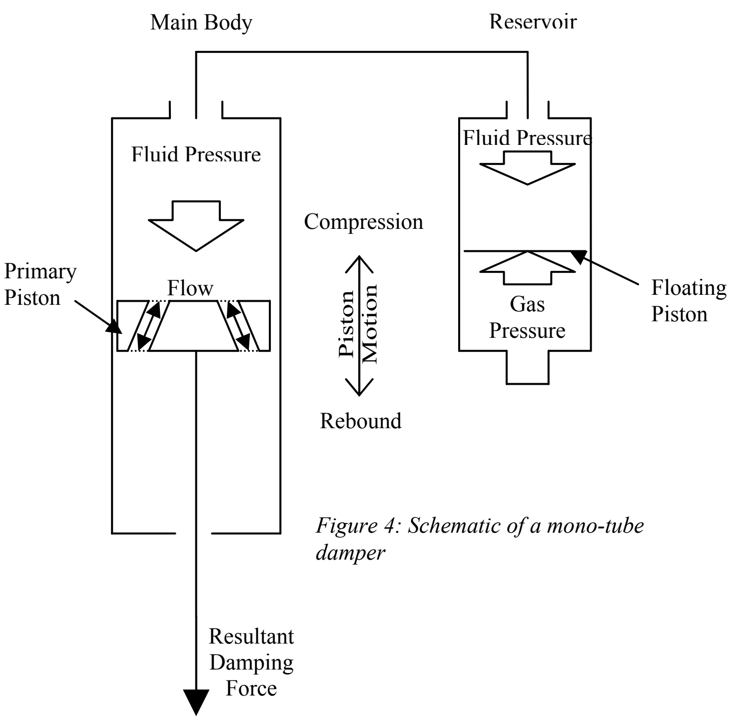


Figure 4: Schematic of a mono-tube damper

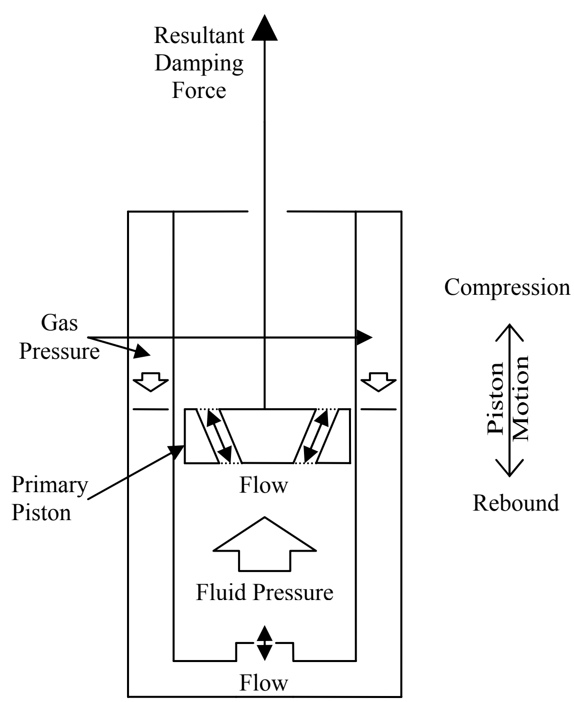


Figure 5: Schematic of a twin-tube damper

Damping Mechanism

Two primary methods for developing the damping force also exist. A typical production car application will consist of a spring valve on the compression side of the piston. A large portion of the compression damping is due to restriction at the base valve; in fact, if the base valve is less restrictive (in compression) than the piston, cavitation will result. The opposite is true on the rebound side. For performance applications, a shim stack is installed on both sides of the piston. This provides an extra degree of control over the damping characteristic and is less susceptible to valving hysteresis and inertial effects. By relying on the shim stack to dominate the damping characteristic, manufacturing tolerances in the valves become less important and the performance should be more consistent from one damper to the next.

Adjustability
In a racing application, it is important to be able to react quickly to changing conditions at the track and modify the vehicle performance accordingly. This could be due to a drastic change in temperature and/or rain. All dampers are inherently adjustable through the internal valves and/or shim stacks on the piston as described earlier. Most dampers intended for high performance use employ some form of external adjustment which can alter the damping characteristic without changing the internal components. This is desirable because the vehicle can be changed in seconds at the track, but it adds a considerable amount of complexity to the design. This is especially true because in order for the adjustment to be useful, the engineer must be able to adjust both compression and rebound damping independently.

Packaging
The remaining design considerations are driven mostly by packaging concerns. The most significant packaging issue is where to place the reservoir. Whether using a twin-tube or mono-tube layout, there must be enough space to allow for the damper to operate, without interfering with any other components on the car. While this is typically not a problem on production cars because of the McPherson strut configuration that has been widely adopted on sports racing cars. While on an FSAE car, space is at a premium, which requires some rather un-elegant solutions to overcome this challenge. A remote reservoir connected to the damper with a flexible hose can be mounted stationary allowing for the damper to move.

Selection Process and Final Design:
The design of a shock absorber is essentially open-ended with many potential directions leading to different final products. All shocks start out with a similar basic design: a pressure vessel filled with fluid resisting the motion of a piston. However, in order to meet as many criteria as possible, it is necessary to refine this design with an evolutionary process. Initial designs are altered as the design team discovers ways to improve damper performance.

Global Layout
The twin-tube configuration is easier to mass produce, though this solution would be harder for us to manufacture because of the specialized processes involved. The outer casing of a twin-tube damper provides a stiffer outer structure and also isolates the piston/inner-body interface from debris. Thus, a twin tube is better suited to survive unexpected force inputs (i.e. from striking debris). This would be very necessary in most forms of off-road racing, where trees, stones or other potential projectiles threaten the dampers; however our racing environment is much more civilized, so the extra protection is not really necessary for our application. Twin-tubes have many drawbacks that make them not well suited for our vehicle. A mono-tube damper provides us the maximum possible piston diameter for a given body diameter, i.e. a thin-walled tube. This helps to reduce both the internal operating pressures of the damper, as well as the necessary static gas charge pressure. Both of these effects help to reduce friction in the seals, as well as the overall hysteresis present in the damper as a whole. Since one of our major design goals is to

achieve lower hysteresis levels it did not make sense to limit our piston diameter with a twin tube design. The performance benefits afforded by the mono-tube layout far outweigh the increase in durability provided by a twin-tube according to our quality function deployment analysis, as discussed earlier. Thus, for this project, we have decided to employ a mono-tube layout.

Furthermore, mono-tube dampers can be inverted (since the gas is separated from the fluid), they are typically lighter and the fluid reservoir can be mounted in a more convenient location to ease any packaging concerns. We must overcome the decrease in rigidity compared to a twin-tube setup. This can be accomplished through the use of spherical bearings to mount the damper both on the chassis side and to the bell-crank, allowing the damper to rotate with the suspension linkage motion. This essentially means that the damper will operate in tension and compression only, whereas most twin-tube dampers must be designed to accept a significant bending load.

Damping Mechanism

The use of deflecting shim valves facilitates easier and faster rebuilds better control over the force vs. velocity characteristics and less mass. All of these things are desirable in a racing application, where we anticipate evaluating many different configurations on the vehicle to (1) validate our simulation results and (2) find a configuration that provides the best handling characteristic. We plan to utilize a deflecting shim valve design in our dampers because it allows us to better meet our customer requirements, as laid out in our quality function deployment analysis. Specifically, the valving style exhibits more control over the damping characteristic and the need to be easily internally modified.

Adjustability

To accomplish our customer requirements, and meet our engineering specifications, it is imperative that the damper be adjustable externally. Adjustable valving adds a considerable amount of complexity to the damper, so manufacturing and assembly will inherently become more difficult. However, the increase in time and cost will be offset by the ability to alter the vehicle handling characteristics without removing the dampers.

Implementing needle valves is the simplest way to create an adjustable orifice area, which varies the resistance to fluid flow at the piston and base valve. With suitable design of the needle profile the effects will be gradual over a large range; thus needle valves are used at the shaft and the base valve. It is also necessary that the external adjustments be independent, in other words, adjusting the rebound needle, should not also affect compression flow. To ensure tuning independence in rebound, a check valve is located in the end of the shaft: a ball closes preventing flow from entering the shaft during compression. The base valve provides very little resistance against rebound flow (to avoid cavitation at the piston) and thus the variable orifice does not significantly restrict compressive flow.

Packaging

The next major design decision was where to place the fluid reservoir and gas charge. Because of the relatively short length of our damper (~200 mm) placing a reservoir and gas charge inline with the rest of the damper (as shown in Figure 5) would reduce the available travel to unacceptable levels. Thus, a slightly more complicated design is needed to separate the reservoir from the main damper body. There are two generally accepted ways to do this, a piggyback style design, where the reservoir and necessary plumbing are affixed to the upper mount, as shown by the dampers shown in Figures 1, 2 and 3. The second solution is to mount the reservoir independently of the damper body, and connect them with lines to allow fluid to flow. While the piggyback design is elegant in its relative simplicity, it does present some packaging issues, namely, freeing up enough extra space near the chassis side damper mount to allow for adequate motion. In a racecar, where packaging space is at a premium it is necessary to minimize the size of all moving components to allow for a tighter package. On the 2006 vehicle, packaging the damper was the major deciding factor because of the proximity of the rear suspension to the engine and all of its subsystems. There simply was not enough room to allow for a piggy back style reservoir in the rear of the car. Thus we opted for the remote option. This allows us the freedom to place the reservoir in an area that is less congested. Furthermore, this makes both the upper mounts and reservoir less complicated to manufacture, which is important for a student team without access to 20+ years of machining experience and the most state of the art manufacturing equipment.

Figure 6: Location of right-side dampers in FSAE car (rear in left picture, front on right)

Final Design Specification Summary

To summarize, our final design specification calls for a mono-tube layout to provide the best possible performance characteristics. We will use a remote fluid reservoir to ease some packaging concerns as well as remove the reservoir from a hostile (i.e. hot) environment, especially in the rear of the car near the engine. We will use deflecting shim stacks on the piston to achieve our damping control in addition to the internal base valves. The shims provide another dimension for us to modify the force vs. velocity characteristic of the dampers, which is useful for arriving at an optimal solution. External adjustment, which is necessary in a racing application such as ours, will be achieved via needle valve adjustable orifices, both at the piston (for rebound control) and base valve (for compression adjustment).

Evolution of Components

In addition to the topics discussed above, several other components are necessary to create a working damper. The following sections outline these various components as well as some design decisions made regarding their functionality.

Body

Housing the piston, shim stack and most of the hydraulic fluid, the damper body closely resembles a thin-wall cylindrical pressure vessel. Its inner surface provides a guide for the axial motion of the piston and its outer surface must securely hold the spring. Since leakage past the piston will adversely affect the performance of the damper, the dimensional and surface finish tolerances on the inner surface must be strictly enforced to permit proper sealing and low friction. The two open ends of the body are enclosed with a sealing cap (chassis mount) and a seal spacer, which prevent fluid from leaving the ends of the body. Since the outer diameter of the body must hold the spring, the entire length is threaded, which allows the spring to seat on a threaded perch.

This provides the mechanism for adjusting the position of the spring, which can change the ride height of that corner of the vehicle. There has been little question towards the material for damper body: 7075 aluminum provides the machinability and light weight necessary of a small custom damper, and the stiffness and strength required of a pressure vessel. An area of major concern is the threads, which must be able to accept the load from the spring. Thus, a stub ACME thread is used because they are 30% stronger than a conventional UNC standard thread as there is more surface area per thread.

Chassis Mount

The chassis mount of a damper is essentially the cap which attaches to the damper body via an internal thread and projects load onto the chassis. Also made of 7075 Al, the chassis mount must seal the end of the body and meter fluid to and from the remote reservoir. There have been large design changes to the chassis mount, must due to concerns about stress near the spherical bearing; where the material experiencing the stress transferred from the body is at a minimum. Thus, a few design iterations were performed, relieving unnecessary sharp edges with large fillets and increasing the amount of material near highly stressed areas. Though this mount is more difficult to machine, it should be significantly stronger without adding lots of mass, and it is more visually interesting as well (refer to Figure A3 in Appendix 5).

Shaft

Though a shock shaft seems to be a relatively straight-forward component, shafts used in automotive dampers are deceivingly complex because of the tolerances and internal flow paths. Since the piston must be allowed to move throughout the damper body, there must be a sliding interface on the shaft that seals the hydraulic fluid inside the body. Not only must this interface keep the fluid inside, but it must keep air and debris outside and be able to properly transfer any radial load without deformation or excess friction. Because of the many constraints placed on the shaft, it requires the highest tolerances of any single part. Typically, the shock shaft is hardened and ground which provides the surface finish required of the seals, prevents scratching, and reduces localized deformation under load.

ArvinMeritor donated 11mm diameter shock shafts used in their snowmobile racing dampers. These are case-hardened ground 1035 steel rods, which meet the surface finish and hardness criteria needed by the seals. This greatly reduces manufacturing time, though there is still a significant amount of machining on the shafts before they can be assembled into the shock.

Initially, the shaft was designed similarly to the Fox shaft, with the end of the shaft connected to the top of the piston; see Figure A1 in the appendix. Discussions with engineers working on the Cadillac and Corvette racing programs at Pratt & Miller Engineering led to the conclusion that terminating the shaft at the piston causes unwanted variations in shim preload. If the number of shims is changed, then internal needle valve must change as well, because the needle gap increases inside the shaft. If the shaft passed through the piston then the needle gap would be unaffected by changes in shim orientation. Tuning the dampers will be much easier if the adjustable parameters can be modified independently of each other as discussed above. Thus, the shaft design was changed to a through-piston style (Figure A1); this requires considerable effort to machine the internal bores, both of which require excellent surface finishes for the seal and valve.

Compression Adjuster and Reservoir Cap

The reservoir was designed around meeting two main goals: simple and lightweight. The Penske, Ohlins, and Fox have complex valving for compression adjustment, the latter two is probably unnecessary (the Penske dampers have independent high and low speed control requiring a more complex solution than the needle valve we'll employ). Basically, the reservoir is very similar to the Fox, using the same Fox base valve and a similar separating piston. However, by re-orienting the compression adjuster 90 degrees compared to the Fox design, the apparatus is simplified. This also reduces the number of required components.

The first design had one significant weakness: the compression adjuster did not have a positive stop. This means that the adjuster, which is threaded to the cap, could be unscrewed to the point that it is removed from the cap. This is not a good situation for the team member making compression adjustments, who is likely unaware of the pressurized oil about to be ejected into his/her face. Thus the adjuster was redesigned as a two-piece system that is inserted from the inside and cannot be externally removed. This is illustrated in Figure A2 in the appendix. Also, the adjuster profile was modified. Instead of a sharp needle as seen on the other dampers, the adjuster features a rounded end, resulting in a more gradual increase in orifice area, thus reducing the typical "on/off" behavior of the sharp needles, and providing a wider range of useful adjustment.

Rebound Valve
Since the optimal shape of the valves can only be confirmed by CFD and testing, it is necessary to create multiple styles of valves. Thus different designs of the valve nose will be tested to find an adjustment range suitable for the engineering specifications. By keeping all other important dimensions (overall length of valve, OD, etc.) unchanged, the valves should be interchangeable.

Engineering Analysis
The main goal of our project is to construct a damper that is specifically designed for use on an FSAE sized car and subject to the kinds of inputs FSAE cars typically experience because a current solution is not readily available on the market. Thus, the main focus of our engineering analysis is on how to specify a damper that provides the best possible array of characteristics for our application. To do this, we conducted a comprehensive analysis of the vehicle in each of its principal motions, with a special emphasis on how the damper affects performance. Using this analysis we developed a baseline force vs. velocity curve that the dampers will initially be built to.

Before we can design a damper to a specific force vs. velocity profile we first need to understand (1) how the damper affects the vehicle in its various modes of operation and (2) what the "optimal" damper characteristic should be in each of these modes. To accomplish this, we used various vehicle models to examine the vehicle response. Using these models, it is possible to develop a baseline damping profile that is well suited for our vehicle. Once a target has been established we began an iterative process to design the shim stacks that will define the damper performance. It is virtually impossible to predict how various combinations of shims will react to produce a damping force without an incredibly complex numerical simulation model. This type of model would include a coupled computational fluid dynamics (CFD) and finite element mechanics (FEM) models. It is far simpler and more efficient for us to design the shim stack using an iterative process on a shock dynamometer. A simple analysis of how fluid moves when the damper is in motion will provide a basic understanding for how the shim stack should be designed, with the details to be worked out on a dynamometer. The purpose of this section of the report is to outline the process above that we used to develop the baseline damper curve, as well as a basic discussion of fluid mechanics in the damper that we used to design the shim stacks.

Vehicle Dynamics Analysis

By approximating the damper characteristics using a linear coefficient we could gain insight into its affect on the vehicle as a whole, in each of the distinct modes of operation. The vehicle has four basic modes of operation; roll, pitch, heave and warp.

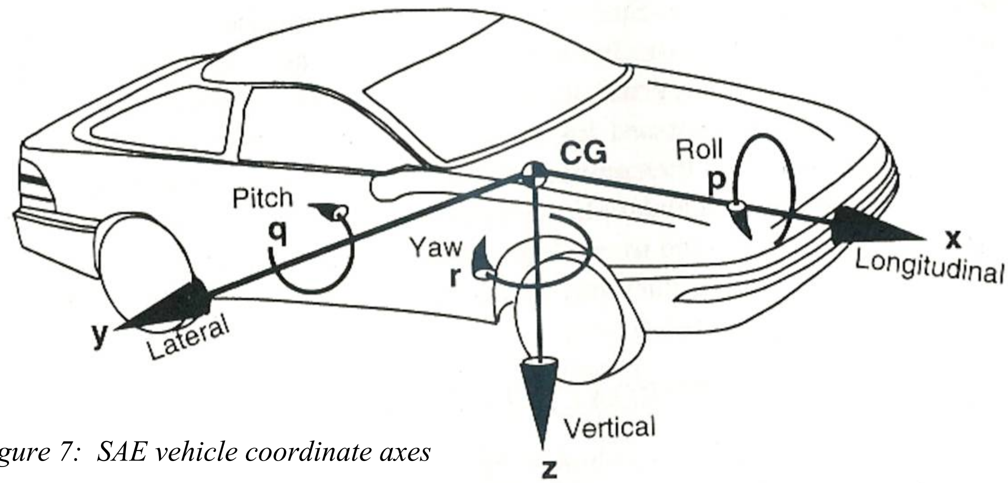


Figure 7: SAE vehicle coordinate axes

The first three modes (roll, pitch and heave) refer to motions of the sprung mass relative to the SAE coordinate system. Roll describes vehicle rotation about the “x” axis. Pitch refers to rotation about the “y” axis. Heave is motion of the sprung mass along the “z” axis. Warp traditionally describes motion opposite corners of the car in the same vertical direction, for instance the right front and left corners moving up; however this can be simplified as the vertical response of a single wheel, such as hitting a pothole or other road imperfection. This is the definition that we will use for the remainder of the report.

As discussed previously, the damper has two principal modes of operation, high speed and low speed. Most heave, roll and pitch motions actuate the dampers in their low speed regime, while single wheel inputs are typically high speed events. Recall that low speed refers to piston speeds below 2 in/s while high speed is piston speeds above 2 in/s. In effect, we are trying to use the damper to control motions in 4 distinct areas, high and low speed compression and rebound. By looking at how the vehicle reacts in each of these modes we can develop a baseline specification for a damper and then design an appropriate valve system to match this baseline curve.

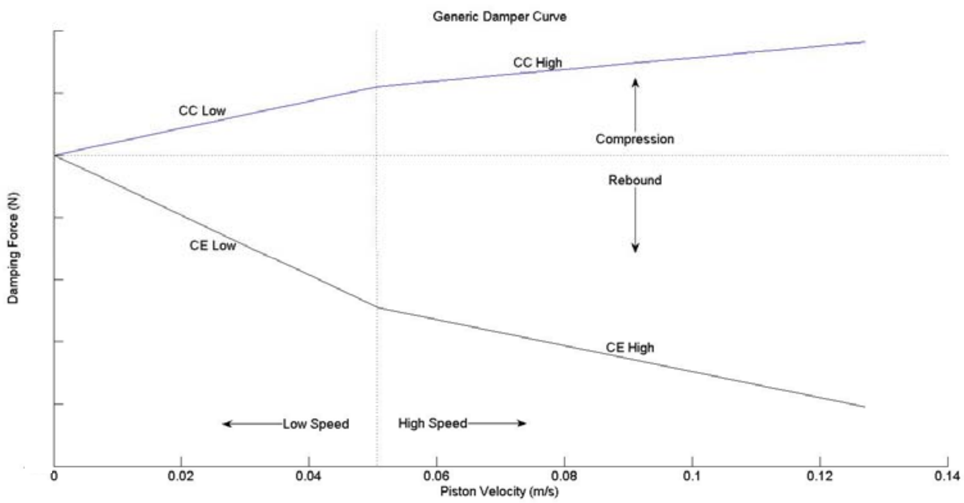


Figure 8: Generic Force vs. Velocity curve with 4 distinct regions

Since the vehicle has distinct modes of operation that actuate the dampers in different regimes it is reasonable to isolate the low speed and high speed damping regions and tune them independently. Both high and low speed regions affect damping in the compression and rebound directions and thus there are 4 separate linear damping coefficients that will describe the force vs. velocity behavior of our damper. These coefficients are shown in the area of the curve they affect in figure 8. This gives a sufficient approximation of the velocity behavior of the damper and provides a baseline for tuning.

While the behavior of a damper is quite complex, general characteristics can be calculated which can provide broad guidelines. By determining these general parameters, which are based upon information about the environment (in this case, what will make the fastest FSAE racecar), some specific data will be known before the damper has been built. There are three main parameters which are of interest in engineering targets: mean damper coefficient, compression/extension ratio, and progressivity. The equations for each parameter are as follows:

$$C_D = \frac{C_C + C_E}{2}$$

$$R_{CE} = \frac{C_C}{C_E}$$

$$\lambda,\ F = CV^{\lambda}$$

As described before, an automotive damper is bi-directional: its characteristics are different in compression and extension. Using a bi-linear damper model greatly complicates the equations of motion involved in a suspension simulation, thus it is very practical to use a linear damper coefficient (same in both compression and extension), C_D. Once this mean coefficient has been found, the bi-linear coefficients can be solved using the compression/extension ratio, R_{CE}. R_{CE} is most often empirically determined either from experience or from optimization of a baseline value during physical testing. The progressivity factor, λ, essentially described the type of damping (Coulomb: λ = 0, Viscous: λ = 1, Quadratic: λ = 2, etc.). For this project, the progressivity will be linear in each velocity regime, meaning that the is a linear coefficient for low speed rebound, low speed compression, high speed rebound and high speed compression.

Low Speed

As discussed above, we will use the low speed regime of the damper to control the sprung mass motions during transient events. The damper is idealized to be entirely sensitive to velocity and therefore does not produce any force unless the suspension is moving. In reality this is not the case, as there are several effects due to displacement and acceleration; for the purpose of this analysis, these effects are ignored.

As discussed above, we want to use low speed damping to control motions of the sprung mass, primarily due to driver inputs. As the driver operates the controls and makes his desired input, the tires react and produce a force. At times when the drivers input changes, such as starting to turn into a corner, the vehicle needs time to adjust to the new position. The time it takes for the vehicle to move from one steady state position to another is commonly referred to as a transient period. In order to maximize performance, it is important to keep the transient periods to a minimum over a given lap. The dampers are the principal means by which we can control the vehicle under these conditions.

To examine the behavior of the vehicle during the transient periods, we developed two state variable models to work with. The first model investigates the vehicle's response in cornering and the second looks at the response to longitudinal accelerations. A schematic for each model is shown below with all state variables shown. Code for the models and their derivation can be found in appendix 9.

To simplify the model several assumptions are made. In each model we assume that the vehicle is symmetrical in every plane about the CG. This allows us to look at the vehicle in a two dimensional space, which simplifies the equations of motion significantly. This is a reasonable assumption because our vehicle is very symmetric, with the weight of the vehicle supported evenly by all the tires; we can conclude that the CG of the vehicle is located at the geometric center between the contact patches. Secondly, we assume that the damping force is proportional to velocity and produces the same force in both compression and rebound. This assumption makes modeling easier, but is actually an undesirable effect. This phenomenon will be discussed later.

To further simplify the model, we consider the unsprung mass to be negligible. This is a reasonable assumption because under low speed conditions, we assume that the vehicle is being driven on a smooth flat surface. Thus, the positions of the wheels, which appear to move relative to the body fixed coordinates, do not actually move in the inertial coordinate system. Thus, the dynamics of the vehicle can be fully described with two dynamic equations; a full development of the pitch model is presented in appendix 9. The roll model can be derived using the same process, and is essentially similar to the pitch model.

By examining the vehicle response while changing the damping parameter, we can find a value that provides the best compromise between a fast settling time and low overshoot. For any given vehicle, a stiffer damper will slow the vehicles response to changing conditions. If the damper is too stiff, the vehicle may never reach steady state conditions. This vehicle will be slow in general and also difficult for the driver to control. Figures 9 and 10 show the results of our analysis.

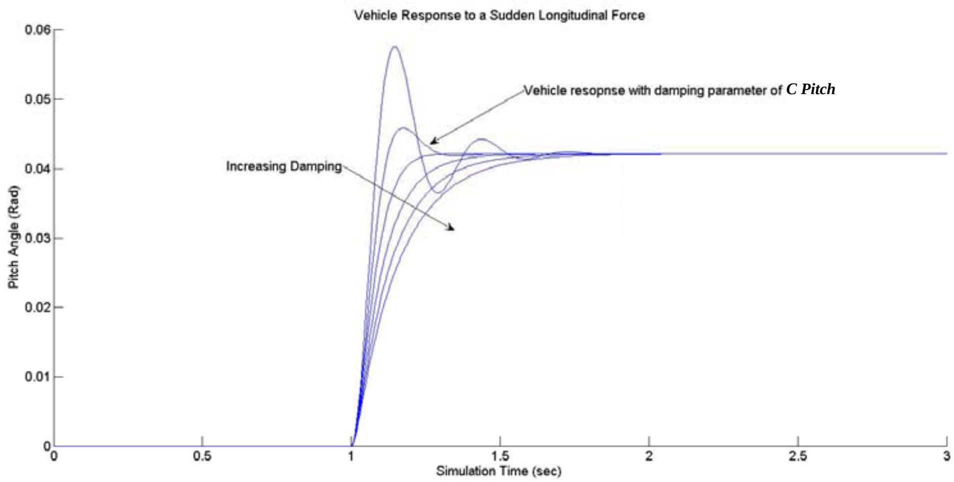


Figure 9: Response of vehicle to sudden longitudinal force for varying damper parameters

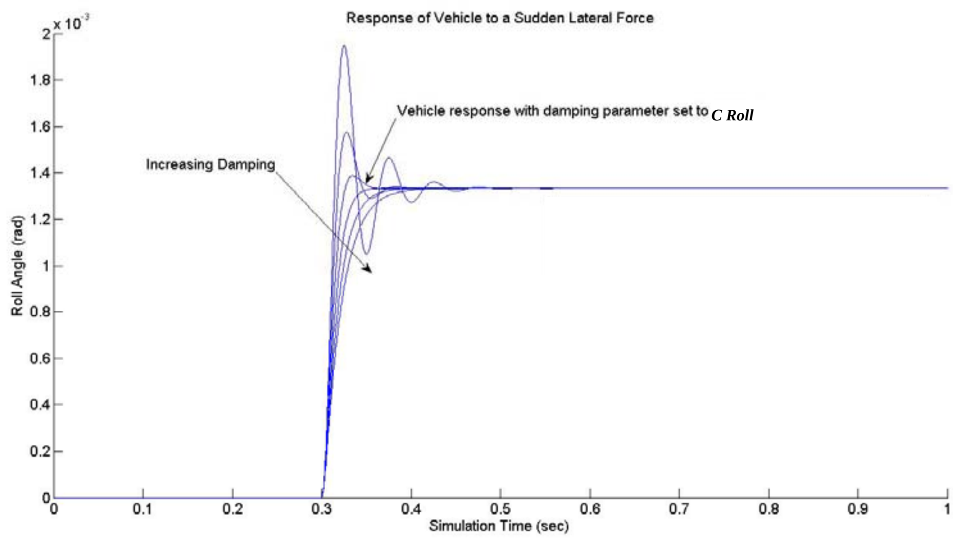


Figure 10: Response of vehicle to a sudden lateral force for varying damping parameters

In the pitch mode, we find the best compromise occurs with a mean damper coefficient of *C Pitch* kg/s, while in the roll mode this same parameter will be significantly over-damped. In the roll mode, the best compromise is found with the mean damper coefficient of *C Roll* kg/s. Since we cannot have different parameters for each mode, we must choose the parameter that provides the best compromise for each of the modes. The roll mode is more critical for vehicle performance because most of the time on track is spent in corners, and the peak lateral acceleration (and therefore force) is roughly 20% higher in cornering than in braking or accelerating. Therefore we chose a mean damper coefficient of *C Low* kg/s for our target curve in the low speed regime, which provides adequate control in pitch, without compromising roll performance.

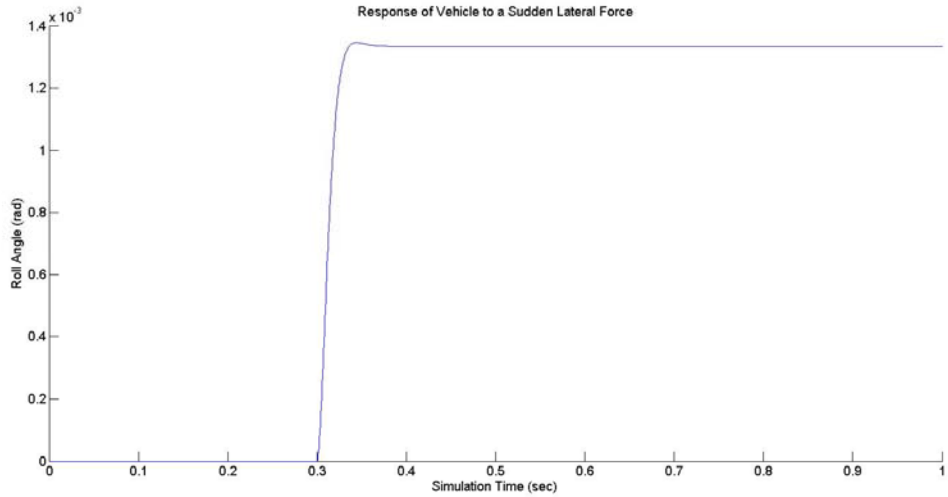


Figure 11: Response in Roll with mean damper coefficient of C Low kg/s

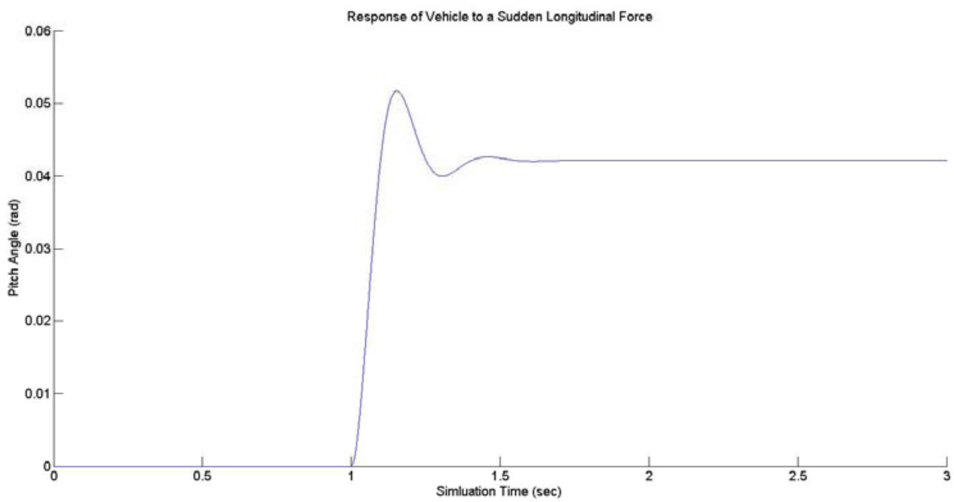


Figure 12: Response of Vehicle in Pitch with mean damper coefficient of C Low kg/s

While the damper is typically used to control vibrations of the sprung and unsprung masses, they also change the vertical loading of each tire during a transient maneuver, such as corner entry. Dampers are capable of adding or subtracting from the total vertical load at the tire, and thus can change the handling of the vehicle at different portions of the corner. The adjustability of the damper allows the low speed coefficient (in both rebound and compression) to be changed which will alter the distribution of vertical load at each tire.

High Speed

On a racing vehicle high speed events as seen by the damper primarily excite the warp mode of the vehicle. As such the goal of damping in the high speed regime is to isolate the sprung and unsprung masses from one another and as a result minimize the effects of any disturbances. This can best be modeled in what is commonly referred to as a quarter car model. The car is simplified to contain ¼ of its mass and effectively one suspension as shown in Figure 13.

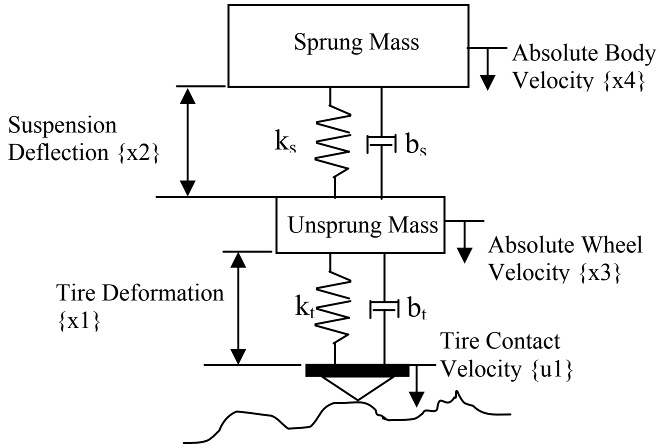


Figure 13: Schematic of the quarter car model.
All state variables and inputs are shown in {}

As a rule of thumb, a disturbance with a frequency higher than 0.5 Hz is considered a high speed event. This can be calculated from the shape of the "bump", and depends on vehicle speed. However the line of demarcation is arbitrary, so using 0.5 Hz is appropriate. There are two main phenomenons that occur in the frequency analysis of the quarter car model; the resonances of the sprung mass (called the body bounce mode) and the resonance of the unsprung mass (called the wheel hop mode). For a typical road car, the body bounce resonance occurs at ~ 1 Hz and the wheel hop resonance occurs at ~ 10 Hz. For our vehicles these frequencies will be higher because it weighs considerably less than typical production cars, but uses springs that are of similar stiffness.

Similar to the models for roll and pitch, the quarter car dynamics can be described with two equations of motion, one for the sprung mass and one for the unsprung mass. There are four state variables and one input. The state equations for the quarter car model, as well as the relevant Matlab code, are provided in appendix 9. Doing a sweep of damping coefficients, similar to the pitch and roll modes. We can look at the response of the sprung and unsprung masses to different frequencies. Figure 14 shows how the unsprung mass velocity gain changes due to varying the mean damper coefficient. Figure 15 is a similar graph, except it is the body velocity gain. These figures can be seen on the following page.

It appears that the body bounce resonance occurs at roughly 2.25 Hz and the wheel hop resonance occurs at closer to 20 Hz. These are roughly twice the values typically associated with road vehicles, which doesn't present a problem in and of it's self, it just means that things will happen faster on this vehicle than a road car. Also, the use of stiff springs means that a large force gets transmitted back into the sprung mass, even for comparatively small suspension displacements. This is not particularly good, because it is less comfortable for the driver and also it tends to upset the car more over rough surfaces. The stiffer springs do however let us run a lower ride height which offers more of a performance benefit than the driver's need to be comfortable.

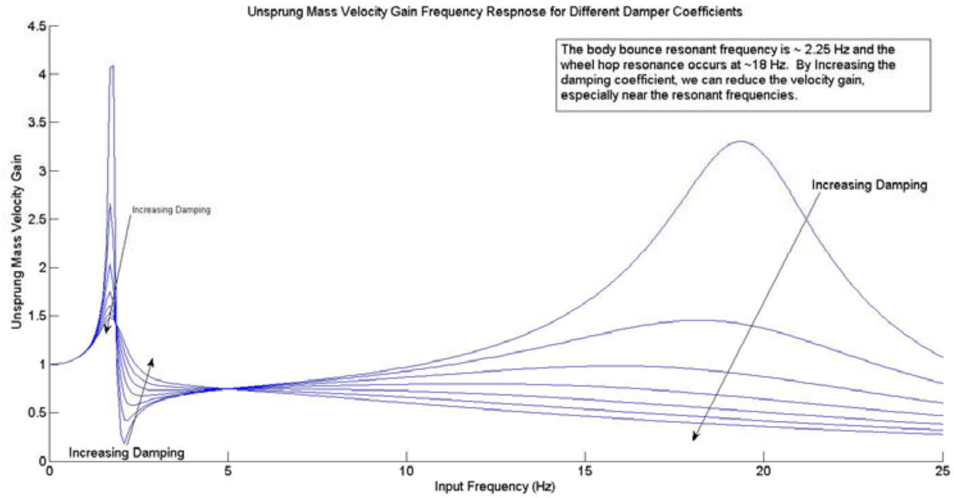


Figure 14: Unsprung mass velocity gain

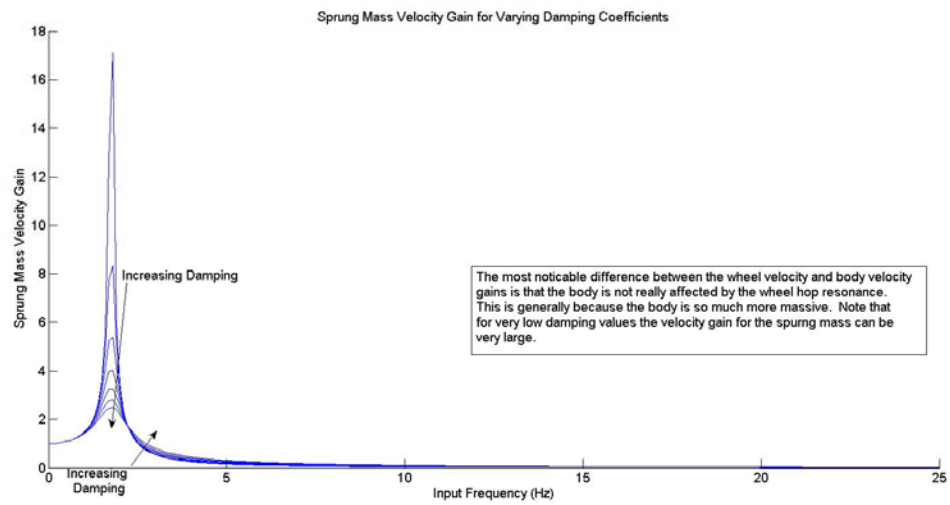


Figure 15: Sprung mass velocity gain

Based on these two graphs it would appear that in the high speed regime that more damping is always better. It is true that adding damping is very effective at reducing the amplitude of various excitations, especially around the resonant points. While these graphs are interesting, they do not tell the entire story. There are two considerable effects worth discussing. First, as the damping force is increased beyond a threshold, the vertical load variation increases drastically. Basically, as the tire rides up the bump, the large damping force increases the vertical load on the tire significantly; however as the tire attempts to ride down the bump, the large rebound/extension force reduces the load on the tire. Therefore, the tire experiences an oscillating vertical load with amplitude proportional to the damping force transmitted by the damper: the stiffer the damper settings, the larger the load variation which is bad for tire grip. Second, by increasing the damping in the suspension the force transmitted into the body due to any road irregularity also increases. This is not a desirable effect because it can lead to vehicle instability on rough surfaces. If the damping were increased infinitely, it essentially becomes a rigid link in the suspension and therefore any change in the road profile gets transmitted back to the body. This would place huge forces on the body and also leave the remaining corners of the car to soak up the disturbance. At this point the vehicle would be impossible to control and very slow. At high speed, the suspension is designed to isolate each corner from the sprung mass and therefore each other. Figure 16 is a plot of root mean square body acceleration for a series of changing damping coefficients. This simulates a vehicle driven over a road profile and calculates the acceleration experienced by the sprung mass due to the road irregularities.

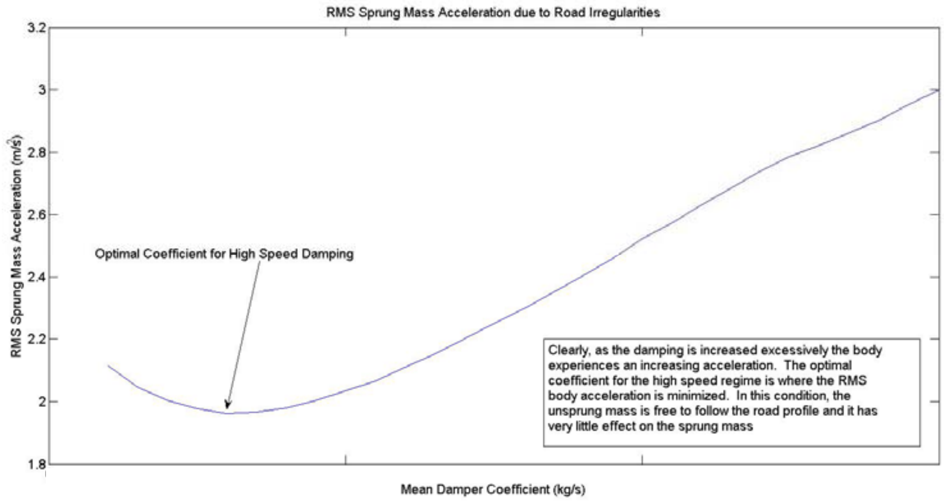


Figure 16: RMS sprung mass acceleration for a simple road profile

The RMS acceleration is very clearly increased as the damping coefficient increases. As discussed above, excessive damping creates a rigid link between the unsprung (wheel) and sprung (body) masses. In the optimal condition, we are looking for the minimal acceleration. This indicates that the body is being affected as little as it can by the road profile. Essentially, the body rides at roughly a constant height, while the wheel is free to vibrate vertically and follow the road profile.. For our car, this occurs with a mean damping coefficient of *C High* kg/s. With this information we have everything we need to make a baseline force vs. velocity curve.

Baseline Curve

We developed mean damper coefficients for the low and high speed damper regimes by looking at the overall vehicle response in these areas and finding the damper parameters that provide the best performance. For low speed damping a coefficient of *C Low* kg/s was found to be best, while in the high speed regime we want a coefficient of *C High* kg/s. With the separation between low and high speed damping at 50 mm/s we can develop the curve based upon the three damper characteristics discussed above (C_D, R_{CE}, and λ).

Through empirical testing (conducted recently by various people) and discussion with experts we feel that the ratio of compression damping to rebound (extension) damping should be roughly 45%. The theory behind using different damping characteristics in compression and rebound is that compression damping can be softer to allow for less force transmissibility as well as tire force variation. Oscillations, which are undesirable, are then controlled by an increase in rebound damping. Thus, the suspension is under-damped in compression, yet nearly over-damped in rebound. Using this relationship for the compression and extension coefficients and the mean coefficients we calculated, the following four coefficients are obtained.

	Low Speed	High Speed
Compression	*CC Low* kg/s	*CC High* kg/s
Rebound	*CE Low* kg/s	*CE High* kg/s

Table 1: Linear damping coefficients for baseline curve

Using these coefficients, we have a baseline curve that our first run of dampers will be tuned to. These curves will be evaluated through in car testing and further computer simulation to verify the accuracy of our predications as well as improve on the overall performance of the vehicle.

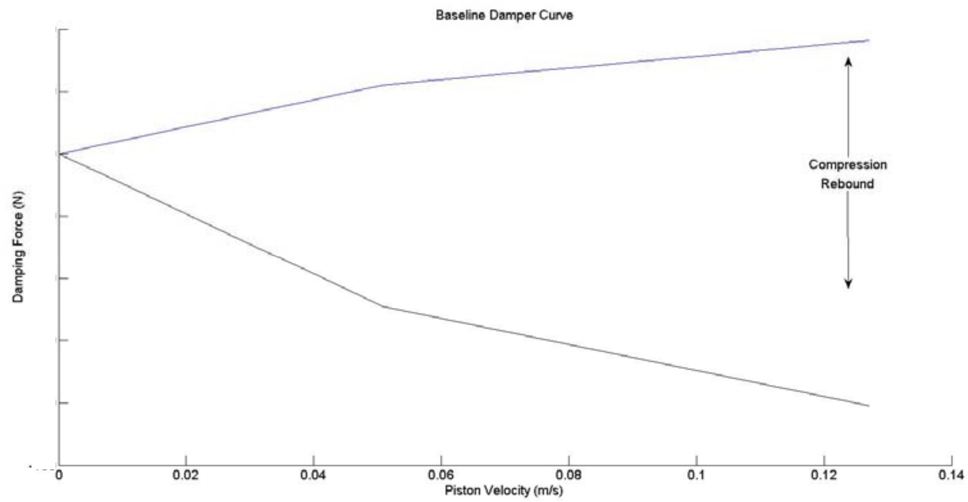


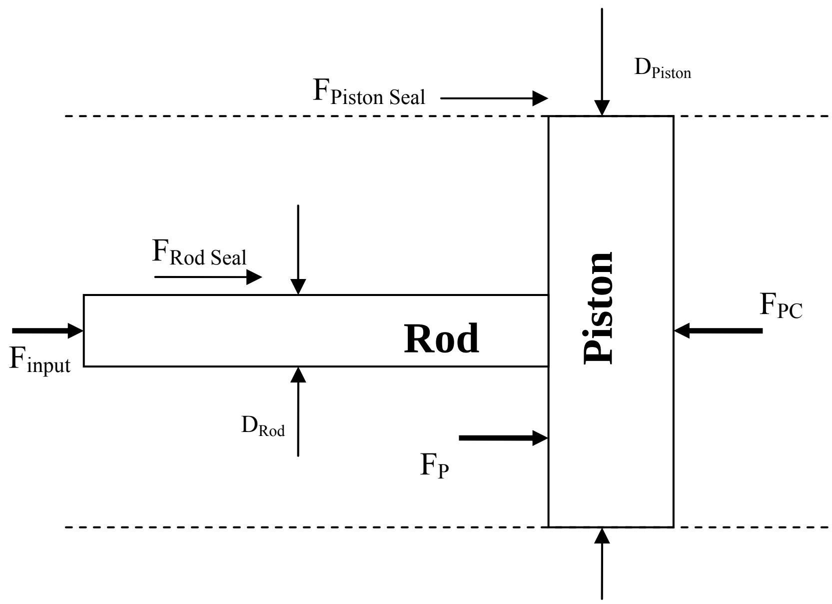
Figure 17: baseline damping curve as a result of linear system analysis

Hydraulic Model

In order to determine initial pressure values for the damper it is advantageous to analyze a simple hydraulic model based upon the geometry of the damper and the characteristics of the valves.

Forces

The force of the piston is related to the area of annulus, which is the difference between the piston and the rod area. This is slightly modified by the variable area of the needle valve in the shaft, but the effect on pressure is small and thus the effect can be ignored. The force from the base valve is related to the area of the rod only. Using the analysis of a free body diagram of a piston, the forces can be summed about the axial direction of the shock.



$$\sum F_x = F_{input} + P_E\left(A_P - A_R\right) - P_C A_P - F_{Gas} + F_{SF} = ma_x$$

Figure 18: Free-body diagram and newton's law in x-direction

This forces involved are the input force (the force from the tire into the damper shaft), the pressures from rebound and compression, P_E and P_C, and the combined force of friction from the sealing components. The mass of the piston is very small (1.4 mg mass) so the acceleration component is unimportant. However as the input force changes, the extension and compression pressures change as well. Also note that there is a gas force due to the pressurization of nitrogen in the reservoir, leading to the "gas spring force" $F_{Gas} = P_{Gas} A_R$. The cross sectional areas of importance are:

Piston: $A_P = \frac{\pi}{4} {D_p}^2 = \frac{\pi}{4}(0.032)^2 = 8.042x10^{-4}\, m$

Rod: $A_R = \frac{\pi}{4} {D_R}^2 = \frac{\pi}{4}(0.01084)^2 = 9.230x10^{-5}\, m$

Annulus: $A_A = A_P - A_R = 7.12x10^{-4}\, m^2$

The force of the piston is related to the area of annulus, which is the difference between the piston and the rod area. This is slightly modified by the variable area of the needle valve in the shaft, but the effect on pressure is small and thus the effect can be ignored. The force from the base valve is related to the area of the rod only.

Valve Flow Rates

Due to the cylindrical symmetry found in a damper, it is beneficial to analyze the behavior in terms of flowrates and pressure differentials across a valve. If the hydraulic fluid is assumed to be incompressible, the volumetric flow rate is the cross-sectional area times the fluid velocity. For analytical purposes, the valves will be idealized as linear, which makes hand calculation possible. Analysis of non-linear valves will require computer simulation of the actual geometry of the valves with CFD, which will be discussed in a later section. The differential pressures across the linear piston and base valves are:

$$P_{BC} = k_{FC}Q_{FC} = k_{FC}A_R V_C$$
$$P_{PC} = k_{PC}Q_{PC} = k_{PC}A_A V_C$$
$$P_{BE} = k_{BE}Q_{BE} = k_{BE}A_R V_E$$
$$P_{PE} = k_{PE}Q_{PE} = k_{PE}A_A V_E$$

where k is the linear valve resistance (Pa-s/m^3), V is the velocity of the piston in compression or extension, and A is the cross-sectional area of the rod or annulus. For example, the pressure drop across the piston during compression is equal to the valve resistance of the piston, multiplied by the area of the annulus, multiplied by the compression velocity of the piston. Refer to Figure 17.

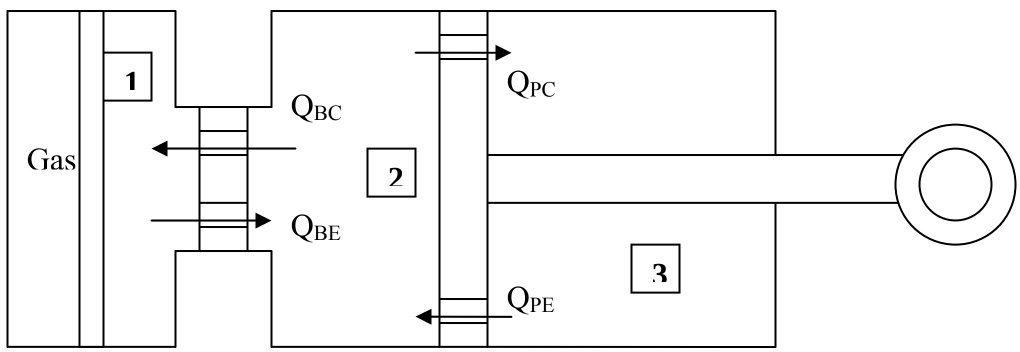


Figure 19: schematic showing different flow-rates through damper

Note that the flow rates through the piston and base valve are in opposite directions. This is because, for example, as the piston is forced in compression, fluid from region 2 must be forced either through the piston (enters region 3) or through the base valve (enters region 1). The numbered chambers can be renamed as follows: 3 is the Extension chamber, 2 is the Compression chamber, and 1 is the Reservoir. During the compression stroke, the chamber pressures can be calculated as follows:

$$\begin{aligned} P_C &= P_R - P_{BC} = P_R - k_{BC}A_R V_C \\ P_E &= P_C - P_{PC} = P_C - k_{PC}A_A V_C \end{aligned} \quad \text{(Eqs: 1)}$$

such that the force on the piston-rod assembly due to F_{PC} becomes

$$F_{PC} = P_C A_P - P_E A_A$$

which can be described in terms of the valve resistances and reservoir pressure as follows

$$\begin{aligned} F_{PC} &= P_C\left(A_A + A_R\right) - P_E A_A \\ &= P_C A_R + \left(P_C - P_E\right)A_A \\ &= \left(P_R + P_{FC}\right)A_R + P_{PC}A_A \\ &= P_R A_R + k_{BC}A_R^{\,2}V_C + k_{PC}A_A^{\,2}V_C \\ F_{PC} &= P_R A_R + \left(k_{BC}A_R^{\,2} + k_{PC}A_A^{\,2}\right)V_C \end{aligned}$$

A similar analysis of the force on the piston-rod assembly in extension results in an analogous equation.

$$P_C = P_R - P_{BE} = P_R - k_{BE} A_R V_E$$
$$P_E = P_C - P_{PC} = P_C - k_{PE} A_A V_E \qquad \text{(Eqs: 2)}$$

Resulting in a total force on the piston rod assembly of

$$F_{PE} = P_C A_P - P_E A_A$$
$$F_{PE} = P_R A_R + \left(k_{BE} A_R^{\ 2} + k_{PE} A_A^{\ 2}\right) V_E \ .$$

There are a few conclusions that must be pointed out regarding the operation of the damper:

1.) The "gas spring force" from the reservoir pressure acts on the rod area and is not affected by the velocity of the fluid.
2.) The linear valves give a force that is proportional to velocity.
3.) Forces from fluid resistance through the valves are proportional to area squared and D^4.

This third point shows the sensitivity of viscous damping to the diameter of the piston. If the piston diameter is doubled and the shaft diameter is doubled (in order to provide adequate stiffness), the damper provides 16 times the initial force.

Cavitation

During damper operation, there is a potential for the fluid to cavitate causing de-solution of gas within the fluid. This condition can occur during normal operation due to poor valve design which leads to vacuum forming in the fluid, typically occurring on the rod side of the piston (Extension chamber) during rebound or on the compression side of the piston during compression. During the compression stroke, the extension chamber pressure must be above a minimum pressure.

$$P_E = P_R + P_{BC} - P_{PC}$$
$$P_{BC} \geq P_{PC} - P_R + P_{E,\min}$$

The 3 pressures must be larger than a certain $P_{E,min}$ or cavitation will occur. However, the piston flows much more fluid than the base valve, thus for linear valves this means:

$$k_{BC} A_R V_C \geq k_{PC} A_A V_C$$
$$\frac{k_{BC}}{k_{PC}} \geq \frac{A_A}{A_R} \qquad \text{(Eqs: 3)}$$

During the extension stroke, cavitation will occur in the compression chamber unless a minimum P_C is exceeded.

$$P_C = P_R - P_{BE}$$
$$P_R - P_{BE} \geq P_{C,\min} \qquad \text{(Eqs: 4)}$$
$$P_R - k_{BE} A_R V_E \geq P_{C,\min}$$

Therefore the resistance of the base valve must be small. Two conclusions can be made from Equations 3 and 4:

1.) The base valve must be very asymmetric (rebound resistance versus compression resistance) to prevent cavitation at the piston. It is shown that the k_{BE} must be very small and k_{BC} must be larger than k_{PC} by a factor of A_A/A_R.
2.) The reservoir pressure, P_R helps prevent the onset of cavitation in both cases.

Thus the reservoir pressure will have an optimal range for damper performance, equal to the minimum pressure at which cavitation is prevented during operation. Attaining this pressure will be beneficial in damper consistency both from the reduction of hysteresis and also the prevention of cavitation (which can damage valves and ruins the hydraulic fluid used).

Once testing has commenced, the actual values of pressure can be input, and the valve coefficients can be calculated from the above equations. However, known information about the design of the damper does allow for some predictions to be made. For example, the area ratio A_A/A_R is approximately 7.71, meaning that k_{BC} must be at least 7.8 k_{PC}, which will be attained by using a stiff shim stack on the compression side of the base valve. Discussion with various performance shock experts has lead to a suggested range of reservoir pressures of about 0.6 to 2 MPa, which will be starting points for tuning. The rest of the values can be determined from CFD results and/or from t
he flowbench data.

Understanding the Results

Our proposed design will meet the customer requirements better than our current solution (Fox-Shocks Vanilla RC dampers). The first major difference is the piston diameter. By increasing the piston diameter, compared to the Fox dampers, the damper will operate at lower pressures. This takes load off of the seals and will greatly reduce any friction between the seals and various moving components. Friction is detrimental to damper performance in two ways. First, friction adds an un-necessary component to the overall damping force (the net force will be higher than required). Also, friction is very difficult to control, and may not be repeatable from one shock to the next, recall that our vehicle needs 4 dampers that have identical force vs. velocity characteristics.

Secondly, by changing the valves we have better control over the force vs. velocity curve than we did with the Fox. The shim stack on the piston and base valve will see less pressure than in the fox, so a much “softer” stack can be used without the concern of plastic deformation or fracture of the shims. Obviously any degradation of the shim-stack over time would lead to a deterioration in performance of the stack. We have experienced this with our current Fox dampers, which degraded to the point of being unusable before their desired window of use. This again ties back into the use of a larger diameter piston to reduce the operating pressures.

With a baseline curve and some basic pressure calculations it is possible to make an educated guess at what kind of shim stack we should aim for. However, without a complex fluid/mechanical simulation model it is not possible to fully design the shim stack to the desired specification. Instead we took an iterative approach and developed the stack with various analyses on a shock dynamometer. This procedure was slow at first, however by making small changes, we could fully understand and evaluate how the shims interacted in the different stacks. As a result, we are able to hone in on the final curve after only a few dynamometer sessions.

Final Design Description

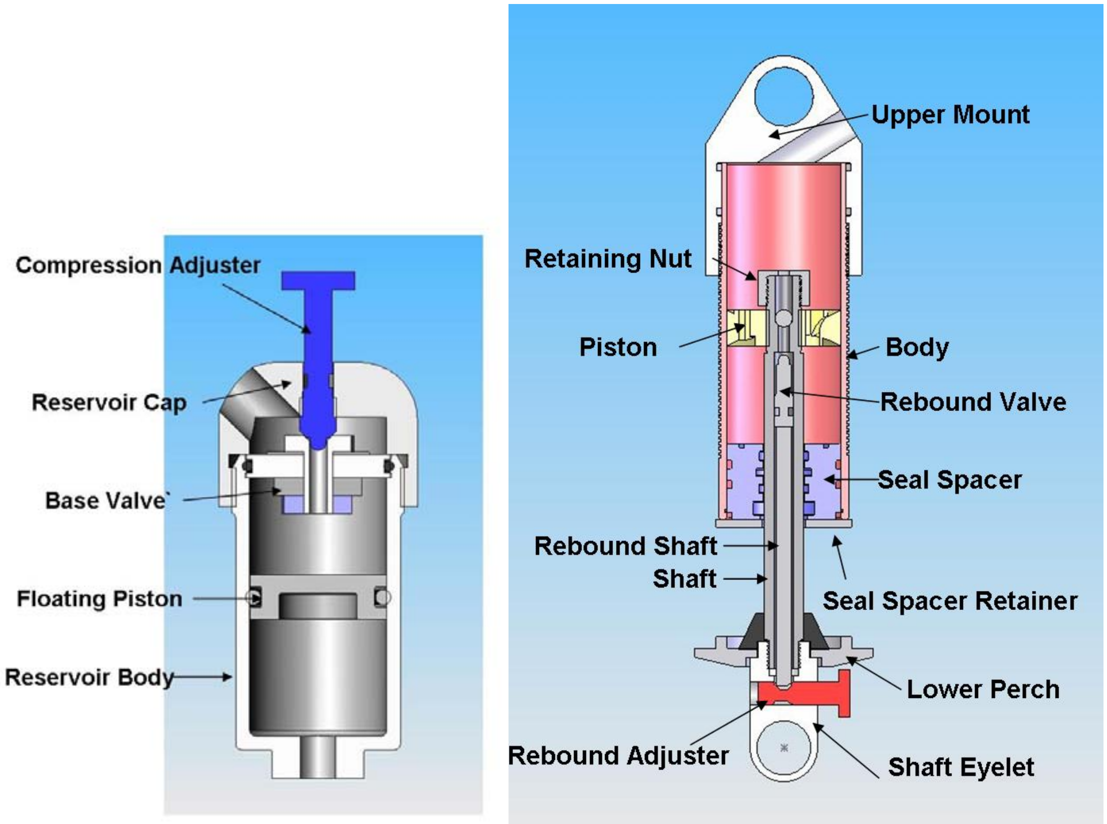


Figure 18: final design with labeled parts

From our vehicle system analysis, structural intuition, and benchmarking, the final design follows from the final mono-tube concept shown earlier. The damper has a 32mm bore and eye-to-eye length of 178mm. Fluid flow through the piston and base valve are controlled by two mechanisms, deflecting valve shims, and externally adjustable bypass orifices. Shim configuration is determined before assembly of the damper, and can only be changed between re-builds. External adjustments exist to allow for on-track tuning.

A majority of the machined components of the damper are made of 6061 aluminum. Aluminum is used for its light weight, machinability, strength as a result of the T6 heat-treatment, and low cost. As an exception, the damper body itself is made of 7075 alloy aluminum for its superior strength-to-weight, and toughness when compared to titanium or heat-treated steels. The shaft, piston and base-valve retainer are made of steel alloys because of our off-the-shelf piston choice, and packaging constraints. See appendix 7 for the bill of materials.

All machined components are made using CNC technology to minimize operator errors and maintain consistent quality. Special machining processes required are the surface grinding and plating of the shaft, and honing of the main bore and reservoir bore. Honing is needed to minimize surface roughness, thus reducing the sliding friction inherent in a piston/cylinder mechanism.

Above all performance criteria, the damper must be durable enough to last the race season. The intention is to manufacture two sets of dampers. But, the design should allow for only one set to be needed on track, while the other set can be used for off-vehicle testing on a dynamometer or a flow-bench. Therefore, the harder the dampers can be tested off the race car, the more confidence we have in their consistency on track.

The damper is broken down into three assemblies. One is the piston/shaft assembly which attaches to the suspension. The second is the damper body which is attached to the chassis. The third is the reservoir assembly, which is also mounted to the chassis and connected to the damper body through a flexible hose.

Main Damper Assembly (Drawings 1-3, in Appendix 8)
The design of the damper body is crucial to the whole damper, in that most components are directly mated to it. The tube is threaded on the outer diameter to accept the upper mount and the upper spring perch. On the opposite end from the upper mount is the seal spacer, held on the inside with a bolt-on retainer. The entire assembly must seal on all sides, therefore close machining tolerances are imperative to ensure proper performance of seals. The bore of the damper body is honed to a surface of 9 µin to reduce friction between the piston sliding along the inside while the damper is in motion. With operating pressures on the order of 5 MPa, the damper must not fail under pressure from yielding or fracture.

The upper mount connects the damper to the chassis. A spherical bearing is mounted in the eye to account for three dimensional articulation of the damper, resulting in predominantly axial loading. In addition, a flexible, steel-braided hose is attached to the upper mount to transfer fluid to and from the reservoir.

With high pressure fluid flowing inside the damper and a shaft moving in and out of the body, the assembly must seal. The seal spacer is machined to hold all the necessary O-rings and guides to keep fluid from leaking while the shaft is moving.

Shaft assembly (Drawings 4 -9)
The relatively small steel diameter shaft transfers load between the fluid inside the damper and the suspension. Careful attention to varying wall thickness and changing cross section is important because of high cyclic loading of the damper. Through the inside of the shaft is the mechanism which allows for rebound damping adjustment. As the damper extends, fluid is allowed to flow through the shaft, thus bypassing the piston valving. At the shaft eyelet, rotation of the rebound adjuster actuates a shaft which moves the rebound valve. This adjustment varies flow through the shaft, and thus the rebound force generated by the damper.

Reservoir assembly (Drawings 11-16)
In its basic function, the reservoir is just that, a place for fluid to go while the shaft assembly displaces fluid in the damper body. Also contained in the reservoir is another shim valve assembly held stationary in the reservoir body. The function of the base valve is analogous to the piston mounted on the damper shaft. The stationary base valve regulates flow through orifices arranged on its face. Deflecting shims are held on with a retainer and restrict flow through the valve itself. The retainer is hollow, allowing for fluid to bypass the shims and the base valve. However, the position of the compression adjuster determines how much flow is allowed to bypass. Therefore, like the shaft assembly, the base valve is both internally (through different shim configurations) and externally (rotation of the compression adjuster) adjustable.

Prototype Description

The final design was developed with the understanding it would be fully fabricated to match that of the prototype and used on the competing 2006 FSAE race car. The prototype does not just demonstrate a novel and sound engineering approach, but is a vital component in ensuring the competitiveness of the University of Michigan FSAE race car through improved vehicle handling. There will be few differences between the prototype and the final design. Such changes might include the shape of the internal adjustments. These changes are dependent on physical testing results and continued computer aided analyses. Other changes will be considered unexpected and likely due to manufacturing complications.

Prototype Manufacturing

Associated with a damper are both small and large components, all of which require tight tolerances to ensure engineering criteria targets are met. The manufacturing processes of such parts can be complicated and require specific machining parameters. Of these processes, the more significant ones are explained (see appendix 10 for all machining operations and appendix 7 for all input materials).

The manufacturing of the damper body is complex because of its thin wall that has threads on one side and a honed surface on the other side. It requires that a steel jig, an arbor, is made so that the turning operations can be performed. When using this type of jig its important to consider what clamping forces we are exerting on the part. Too much force and the part will yield, too little and it will move on the jig during machining operations.

Another notable component is the rebound valve. This is a smaller part that is turned and grooved on the lathe. During these operations it is acting like a cantilever beam, with a long length compared to its diameter. Because of this, the part is more likely to deflect at high spindle RPMs and during material removal. This requires more time and material to setup machining speeds and feeds, parameters noted in the tabulated machining processes. Other slender parts include the compression adjuster and rebound shaft.

The seal spacer is a part that only uses the lathe for its machining operations, but requires a total of seven tool changes. With the large number of tools being used and the clearance complications associated with internal grooving, it’s important to setup tool definitions within the CNC machine. Without this precaution the risk is higher of running the tool into the part or having incorrect tool paths.

The damper contains hydraulic fluid and operates at high pressure. Because we neglect compressibility effects in our design, we must do as much as possible to avoid them through our assembly techniques. To assure a 100% concentration of fluid inside the damper, the entire piece is assembled in a bath its working hydraulic fluid. For simplicity, the three sub-assemblies may be put together outside the bath because of the small parts involved. After which, they can soak in the bath and jarred around to force air bubbles out. Avoiding the entrainment of air into the system also avoids the onset of cavitation effects.

Because of the complexities described above and others not mentioned, it’s that much more important to research machining methods and tools in conjunction with designing.

Validation
It's important for us to understand how well criteria targets are met. Many of these targets can be confirmed given the aspects of the final design. Others need to be validated by means of physical testing and engineering analysis.

Beginning with the assembly of our first prototype damper we began the process of finalizing the internal adjustments (shim stack) of the design based on our engineering analysis. In total, we built 8 dampers as test solutions before arriving at our final shim stack design. With the help of Pratt and Miller Engineering, we were able to test each of these dampers using a shock dynamometer. We started with a shim stack baseline, from which we made small changes to fully evaluate how each of the different shims affect the force vs. velocity response of the damper. The last step in the validation process is running the dampers on the vehicle. Unfortunately at this time the dampers have not been run on the vehicle during testing. This will occur in the coming weeks, as we continue to test and tune our car for the FSAE competition in May. We have few concerns about durability or performance of the dampers on the vehicle and are excited to see exactly what gains are possible when we run the dampers on the car.

CFD Simulation Models
In order to investigate the flow properties of the final design a CFD simulation was run using FLUENT v. 6.2, with the overall goal of assessing the contribution of individual components of the damper towards the overall damping force.

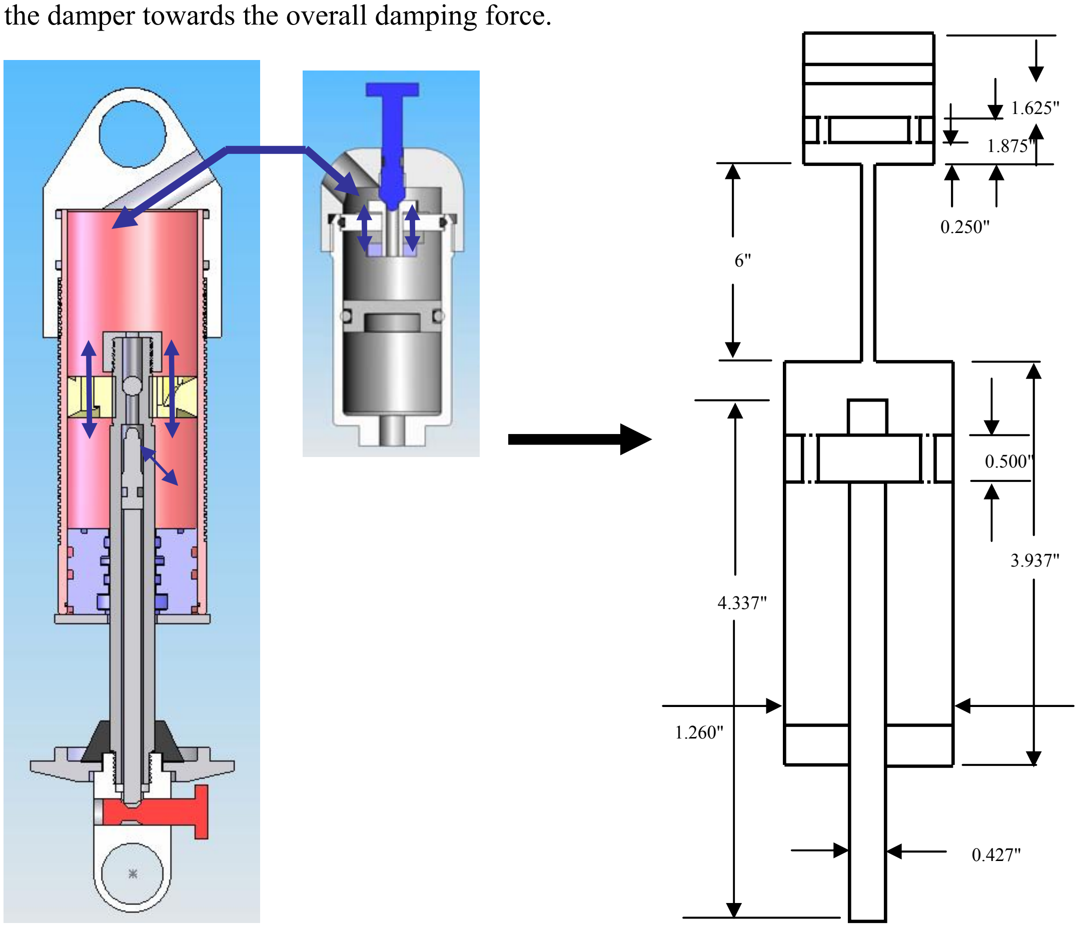


Figure 19: FSAE damper CAD cross section vs. CFD model domain

To simplify the damper's operation, the flows can be split into four regimes: low piston speed and high piston speed and in the compression or extension strokes. In the low speed regime the flow is not large enough to deflect the piston-mounted or base valve-mounted shims, such that all flow through the piston is through the small bleed circuits inside the shaft. These bleed orifice diameters are adjustable via a needle valve, which can be adjusted externally. In the high speed regime, the resistance through the bleed circuit becomes large and the shims deflect (this is the "knee" in Figure 20) and the bleed circuit becomes insignificant. Now the effective orifice is dependent upon the deflection of the shims and the holes through the piston.

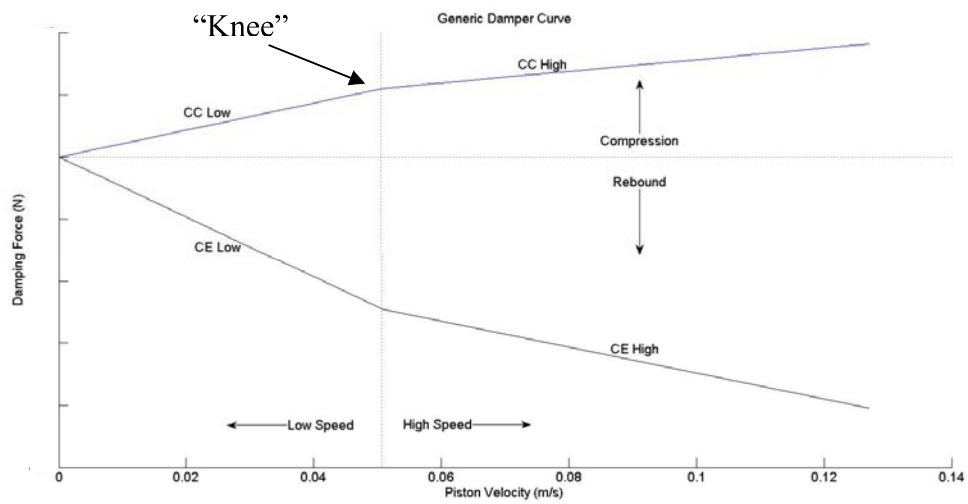


Figure 20: General Force vs. Velocity Curve showing different operating regimes identical to the curve shown in Figure 17 above.

Due to the coupled relationship between shim deflection and differential fluid pressure (across the piston), the entire damper mechanism cannot be modeled with CFD alone. This would require a FEA simulation of the piston shims to be performed in conjunction with the CFD of the damper fluid; the results from each analysis would be used to solve the other system at every time step. Shim deflection is only relevant during the transition from low-speed flow to high-speed flow however, and thus the fluid mechanics of each flow regime can be analyzed independently. The first is the low speed case, where the shims are undeflected and all fluid flow is through the bleed circuit bypassing. The second case is the high speed case, where the shims have fully deflected allowing fluid to flow unobstructed through the piston holes. By appending these two separate analyses together, a simulated force-velocity as shown in Figure 20 can be produced. As changes are made to the valving, the force-velocity curve will be altered and new vehicle behavior will be expected. The reservoir also imposes added complexity because its dynamics are dependent on the reservoir fluid pressure. We attempted to write a user-defined boundary function to integrate the fluid pressure on one side and compare it to the current gas pressure, resulting in piston movement. But we were unsuccessful.

The design of a shock absorber is certainly an iterative process and using tools such as CFD is a valuable resource, eliminating much of the guesswork from the process. Making small adjustments to the valving components is necessary during damper development, particularly since damping force is highly sensitive to valve geometry. Thus it is more efficient to run numerous simulations and examine relative data rather than manufacture separate valves and physically measure damper forces using a dynamometer. This will provide a significant reduction in re-development time as valving changes can be tested through simulation before manufacture of a new component.

Reynolds number definition

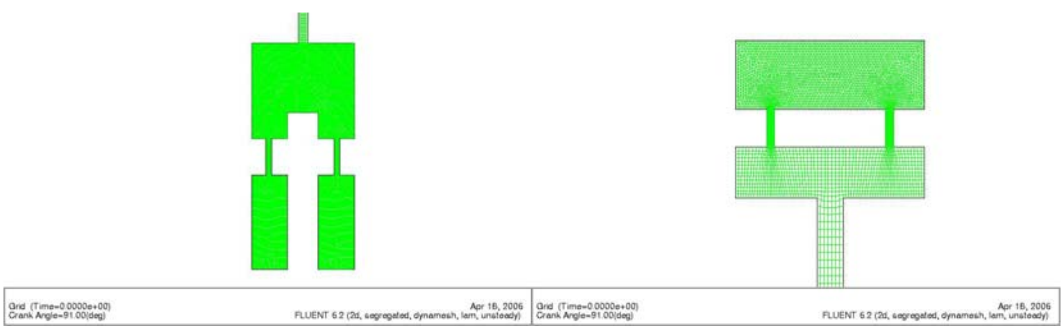
Based on the bore of the damper, average fluid properties for Fuchs Silkolene RSF-5wt hydraulic suspension oil, and shaft velocity ranges applicable to the vehicle's operation, we defined the following Reynolds number regime.

$$\mathrm{Re} = \frac{\rho U D}{\mu}$$

Where ρ is 850 kg/m^3, U ranges between .0127 m/s and .254 m/s, D is 32mm, and μ was kept constant at 0.2 kg/m-s. Density and viscosity were found at 300 K. Therefore the Reynolds number ranges between approximately 80 and 2000. So, we used FLUENT's laminar viscous flow equations for every case.

2D simple damper

To begin the project, we chose to use FLUENT's dynamic in-cylinder meshing model in 2D space. The 2D model consists of a piston with two 2mm diameter holes, and a basevalve with two 1mm diameter holes. These orifice dimensions serve as approximate orifice areas, since the actual piston is a three-dimensional object. The geometry of the model is shown in Figure 19. A uniform mesh size of 0.2mm was used for the piston holes and 0.1mm elements were used in the base valve holes. Paved triangular elements of 2mm were implemented throughout the remaining model interior. Figure 21 shows the grid for our first model.



In order to use a dynamic mesh, it was necessary to define the boundary zones of the model to allow certain volumes to deform and be remeshed. Zones making up the piston undergo rigid body motion which is controlled by the "In-cylinder" motion generator defined in Fluent originally meant for IC engine modeling. See Figure 22 for a diagram of the zone definitions implemented on the damper.

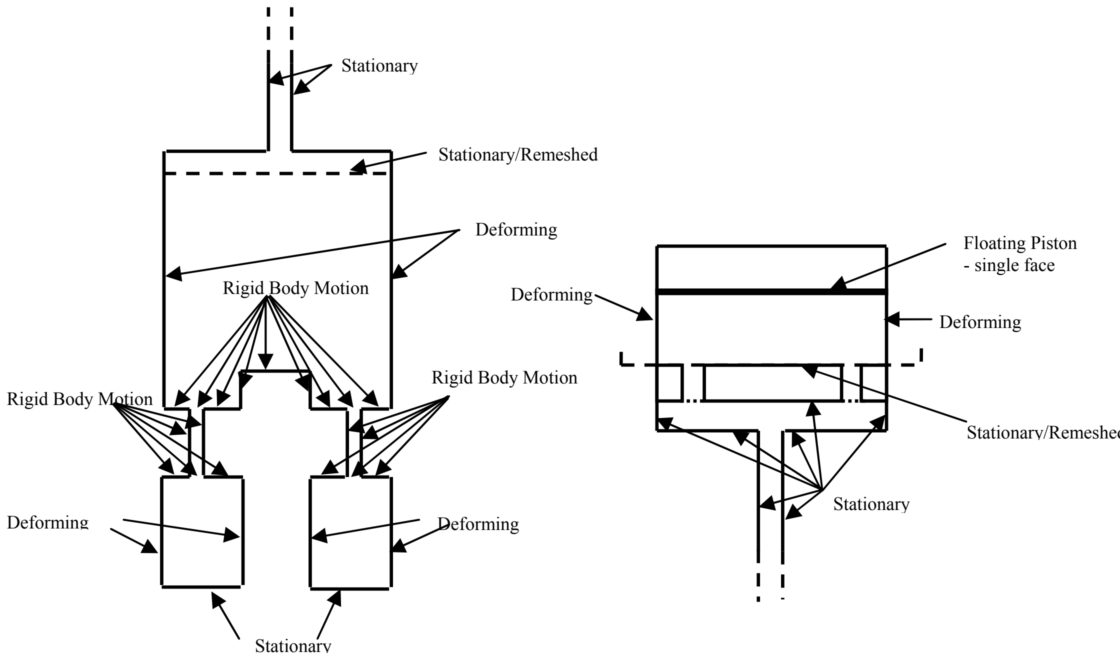


Figure 22: Boundary conditions used in dynamic mesh.

In setting up the dynamic mesh parameters, it was necessary to determine optimal values for the smoothing, layering, and remeshing functions, or else, the mesh was subject to failure from creation of a negative volume. Each function provides a technique to prevent element dimensions from becoming problematic as the volume is deformed. Smoothing represents nodes as being interconnected by linear springs, and places equilibrium conditions on each node, layering applies split and collapse factor criteria which must be met, and remeshing will split elements based upon skewness and/or length scale criteria. All three of FLUENT's dynamic meshing options were used in our models.

The dynamic mesh model requires the unsteady solver to be used. Our viscous model was laminar as mentioned above. Initial pressure in the system was set at 620 KPa, which is the damper's static pre-load.

This model was solved using 500 RPM piston frequency. This is way out of the operational realm of our damper, but the linearity of the solution allows us to use only the velocity ranges that are applicable. By monitoring the velocity and force coefficient along the axis of the damper shaft, we plotted our Force vs. Velocity curve (shown below in Figure 23)

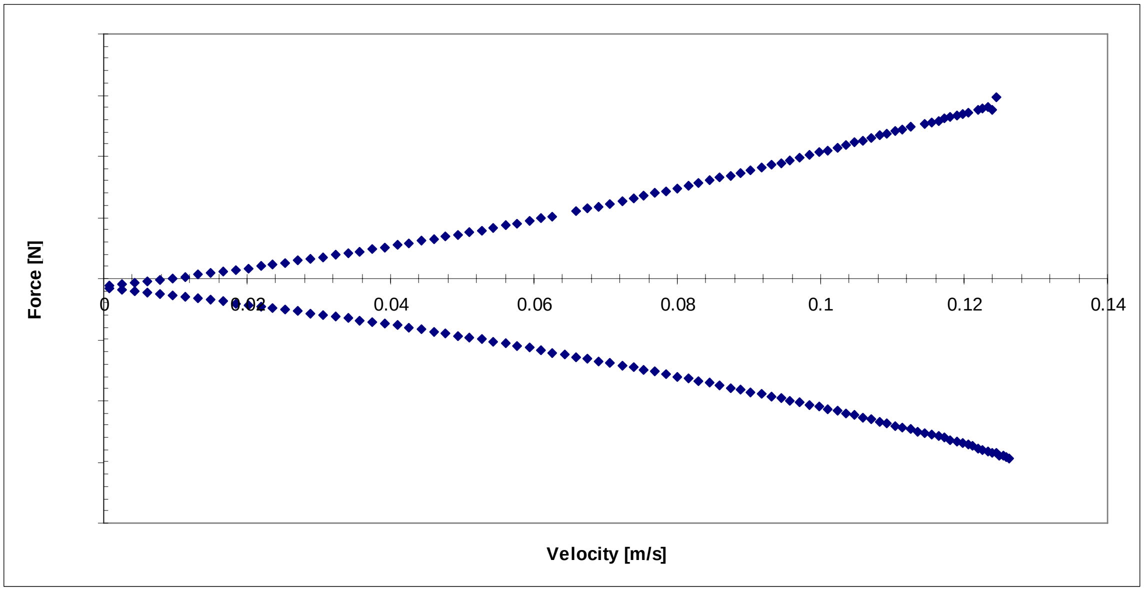


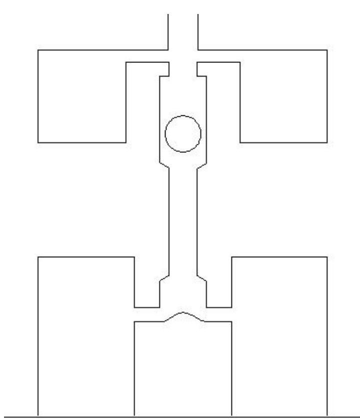
Figure 23: Force vs. Velocity result from initial model

In observing the above plot, we see the force is on the same order of magnitude as our curve in Figure 20, which is successful for our first try.

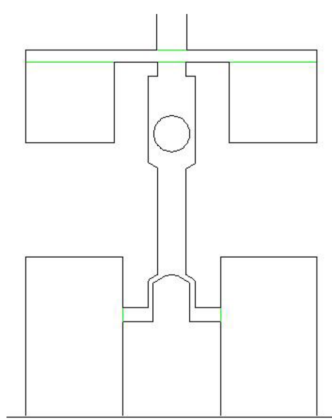
Low-speed rebound model

Of the four damping regimes of the car, the low-speed rebound is one of concern for us. In a corner, it is this behavior of the damper that prevents the car from lifting an inside wheel. Any wheel lift on a racecar is extremely detrimental to overall grip. So we chose to focus our design analysis on this area of the damper's operation.

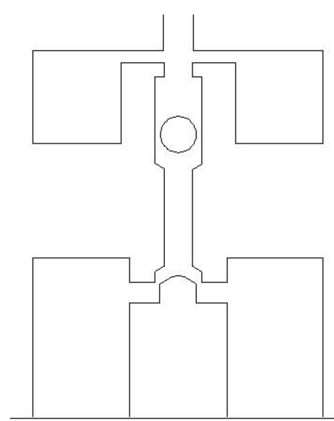
We used the same in-cylinder dynamic mesh model as in the simple damper case. We can only analyze the extension stroke, because rebound flow through the shaft is controlled with a one-way ball valve. We tested two different valve geometries a needle type, and a bull-nosed type. We tested five different positions, two needle positions, and two bull-nose positions.(see figures below) Based on our 'low-speed' definition, we ran a solution at 23 RPM, which corresponds to 0.055 m/s max piston velocity.



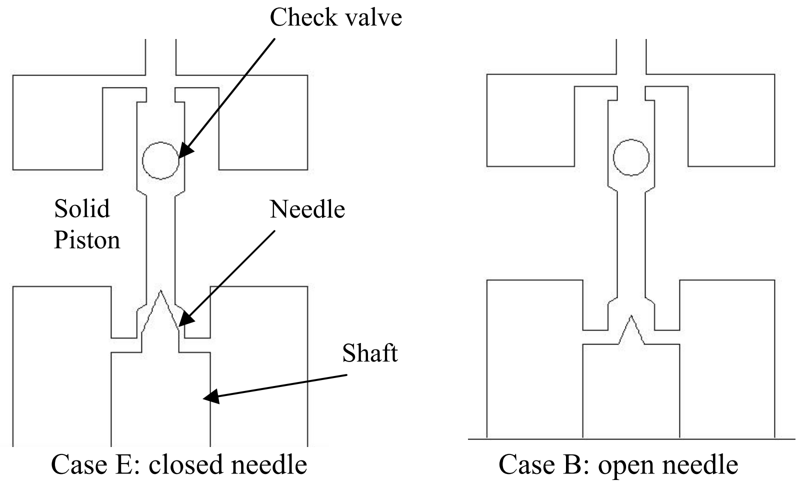
Case A: open bull-nose



Case D: closed bull-nose



Case C: bull-nose halfway



We used a 2D model to look at a slice along the diameter of the damper to cut down on simulation cost. Without thorough fluid property measurements, we did not model any compressibility effects or viscosity changes due to temperature. And without any reasonable source for heat generation in the system, there was no need to solve the energy equation for these models. Friction heat generation comes from prolonged damper operation, but we are only modeling a single stroke.

For our dampers, the reservoir flow influences the damping force by 10% at most. Furthermore, the damper design shows even lower reservoir influence in the low-speed rebound regime due to its one-way nature. So, we defined a pressure-inlet boundary condition on the compression side of the base valve (see Figure 24.). The pressure was held constant at 1.06 MPa, which is approximately the gas pressure at the top of the damper stroke. This was calculated by using ideal gas law and the volume displaced in the damper by the intruding shaft/piston assembly. Each geometry case was meshed using the same size elements in the same regions to prevent bias. The bleed path through the shaft and the immediate areas around the inlet/outlet were meshed with 0.2 mm elements. The rest of the damper was paved with 0.5 mm cells. Once we simulated the mesh regeneration successfully, we ran each case once from the top of the damper stroke to the bottom. The velocity varies sinusoidally with time and the max velocity is at mid-stroke.

Figure 24: Computational Grid of low-speed rebound model

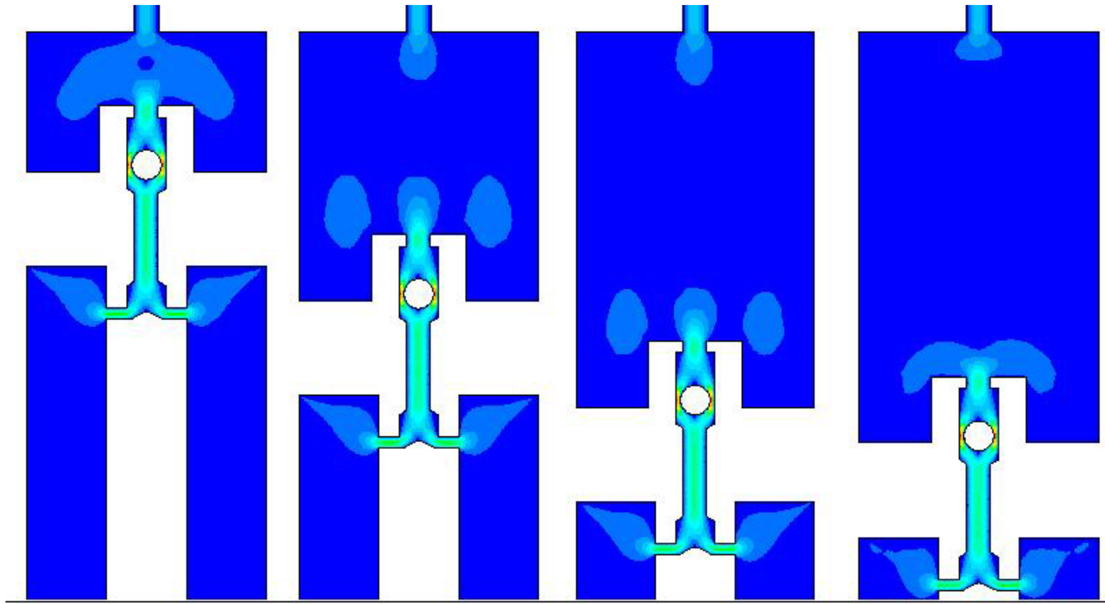


Figure 25: Velocity visualization of full rebound stroke. Bull-nose valve in the open position.

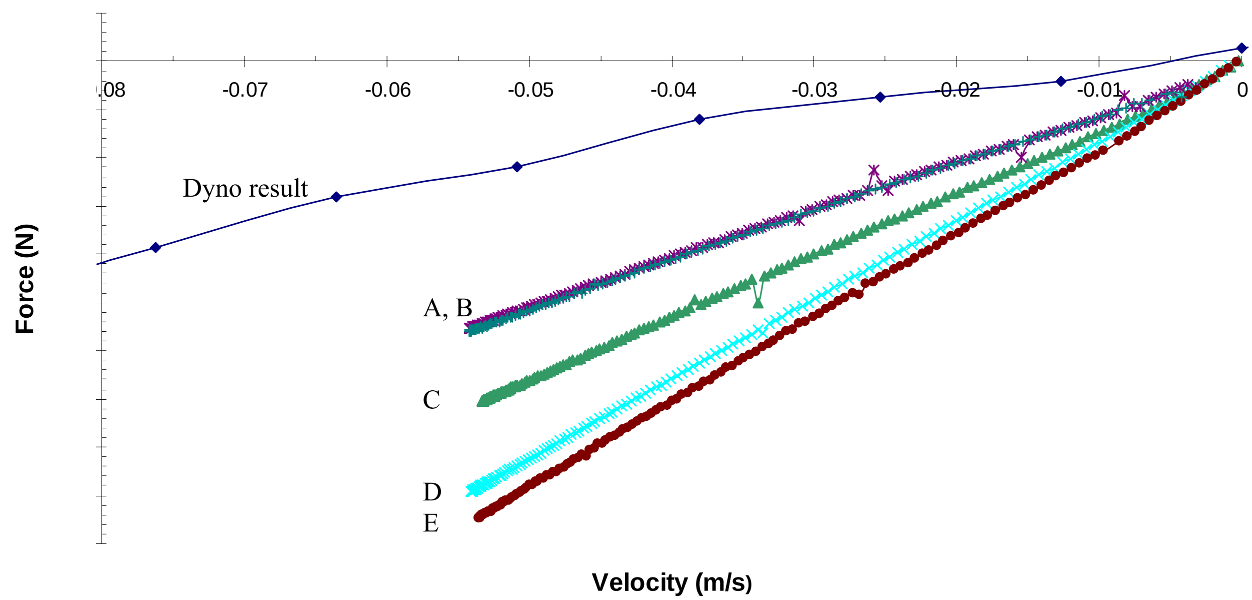
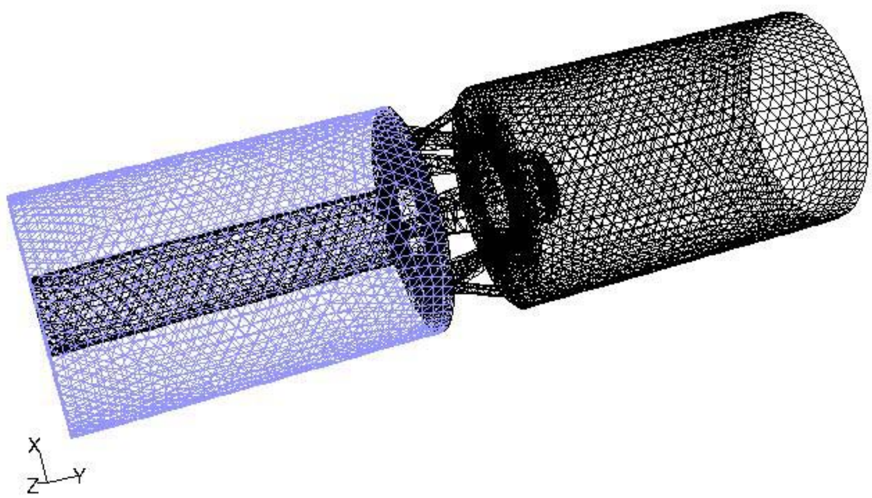


Figure 26: Results of rebound valve design iterations

The results show that decreasing the bleed through the shaft increases the damping parameter, which is an intuitive result. This result can be used effectively to understand the trend in rebound valving and will aid in the design of the external adjustment mechanism. Notice that the data measured on the dynamometer does not coincide with what we had modeled. The damping force is greater by a factor of five at most. The reason for the inconsistency is in the piston bleeding. In the damper assembly, the rebound valve tended to get stuck inside the shaft. Therefore, to prevent the damper from locking, we needed to allow fluid to pass through the piston shims. The additional bleed resulted in a decrease in the low-speed damping. Further testing on the dynamometer will better determine our modeling results.

3-D Simulation
Moving on to the high speed case, the damping mechanism in the damper has now changed. As previously mentioned, the shims which cover the piston holes have now fully deflected and flow through the piston is now proportional to the orifice geometry of the piston holes. This can be approximated by removing the shims altogether from the model and measuring flow properties through the piston. Since the piston holes are now discontinuous, the piston must be modeled three dimensionally. This poses some new challenges, particularly in terms of meshing a component with largely varying length scales in the holes compared to the chambers.

Using a dynamic mesh as was done in the low speed models is far too computationally expensive to perform. The remeshing of the grid in particular is particularly difficult to resolve with Fluent, as element dimensions are not controllable regardless of the updating parameters. The piston motion function: "piston-full" could not be properly implemented with the piston faces; while the model could be run, it was visually obvious that there was no actual motion of the piston. Thus is was determined that while a dynamic mesh of the 3-D damper would give potentially more comprehensive results, it was not possible within Fluent's resources. This is not necessarily a poor assumption; however, as nearly all physical testing of damper components must be done with individual parts, such as the piston. This is to allow numerous styles of parts to be tested with flow benches without having to retrofit the test rig. According to industry engineers, physical testing of valves (as was done in the low speed case) should be separated from the piston and shims, and the results can be summed appropriately.



A picture of the 3-D wall grid in shown in Figure 27.

The use of a non-dynamic mesh allows for a steady-state solution, since piston motion is now constrained, which means that an appropriate far-field velocity must be chosen and implemented. It was decided that a velocity inlet be used at the rebound end of the damper to provide a constant velocity projected at the piston to model relative motion. This requires a pressure outlet at the other end of the piston to allow oil to propagate through without causing unrealistic compressibility.

According to data acquisition obtained from suspension sensors on the 2005 FSAE racecar, the maximum velocity seen by the damper is roughly 0.25 m/s. Therefore a constant axial inlet velocity of 0.25 m/s is defined at the velocity inlet (denoted in Figure 27 by the purple/light grey cylinder). This represents a typical high speed flow through the piston. Next, the velocity inlet is changed to 0.38 m/s and finally 0.20 m/s, which will give three data points such that a linear curve can be fit. Initially, a turbulent simulation was run with a Realizable *k-e* viscosity model using non-equilibrium wall functions, and the results were identical to the laminar case meaning that a turbulent analysis is unnecessary for this project (indicated by the Re range). The solver combined a second order upwind scheme in momentum and pressure with the SIMPLE function coupling pressure and velocity. Also, the energy equation did not change resultant temperature gradients by a significant amount, most likely due to the small impact of viscous heating from the short simulation time. As was done in the low speed case, the drag coefficient was measured along the piston faces and damper shaft such that a resistive force from the oil can be compared in analogous fashion to the 2-D shaft valves. Fluid properties were kept the same, as were the reference values of the drag coefficient.

High Speed Results
All solutions converged within 500 iterations. The drag coefficient was written per iteration and experiences an initial transient after which it converges towards a constant.

$$C_D = \frac{F_D}{\frac{1}{2}\rho A V^2}$$

Recall that the reference values were all set to 1, thus the drag force magnitude, F_D is ½ of the coefficient. The resultant coefficients of drag are displayed in the following Table.

V (m/s)	0.20	0.25	0.38
C_D F_D *(N)*	214.5 107.3	367.9 184.0	845.1 422.6

With the above values, a linear slope is created which describes the Drag Force linearly with inlet velocity (piston velocity), as displayed in Figure 28.

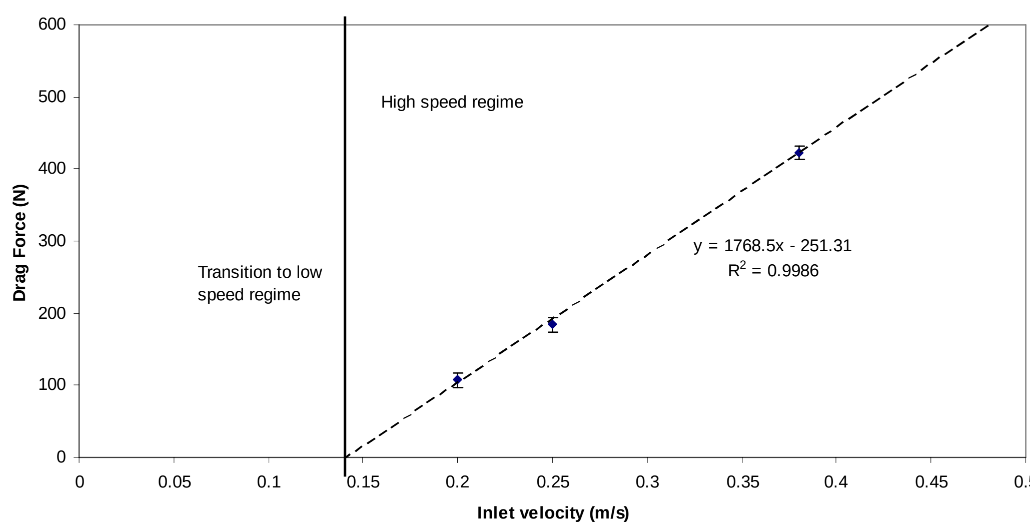


Figure 28: Data points for high speed flow around piston

According to the plot in Figure 28, the drag force slope is calculated to be *1768.5 N/(m/s)*. As expected the coefficient of drag is much lower for the piston than for the low speed valving, which is intentionally designed into automotive dampers. High speed events, such as striking a pot-hole or a curb, would cause unneeded harshness and reduce tire grip. This is the reason for the “knee” transition shown in Figure 20.

3-D Flow Visualization
Contoured with velocity magnitude, the path lines corresponding to $V = 0.25\ m/s$ are shown in Figure 29. Note that the relative direction of travel is along the appended arrow.

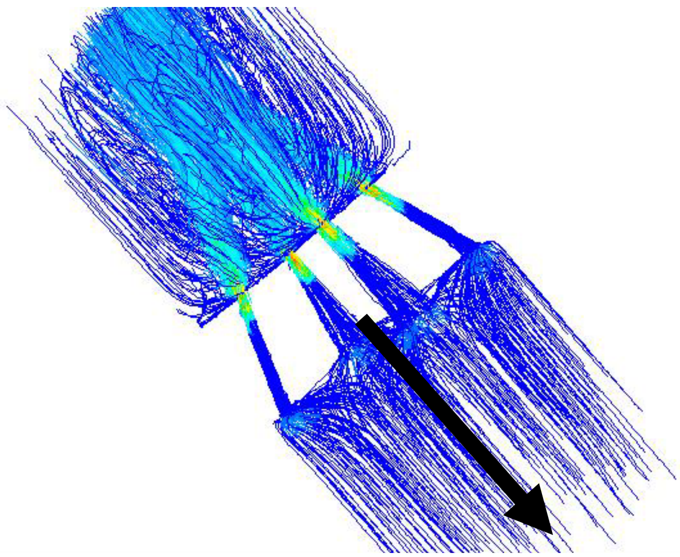


Figure 29: path lines contoured with velocity

The path lines show a large increase in velocity at the exit of the piston orifices, with the 4 streams converging at the axis center roughly 13 mm past the exit face of the piston.

Construction of Force-Velocity Curve
Essentially dampers have two sets of orifices in parallel: the bleed circuit and the piston holes. When flow causes the shims to deflect, both orifices provide flow paths. Thus the above slope can be appended to the drag force results from the 2-D case, in order to construct an entire Force vs. Velocity curve. Using the data from the high speed simulation, a F-V curve can be created which combines damping characteristics from both regimes.

This is done by summing the high speed results with the low speed at each respective velocity. As seen in Figure 28, the drag force from the piston is defined after ~ 0.125 m/s. Therefore, the valve data will be plotted from 0 to 0.125 m/s after which the high speed data becomes defined.

Figure 30 displays the resultant plot obtained by superimposing the low speed, two-dimensional and the three-dimensional high speed analyses.

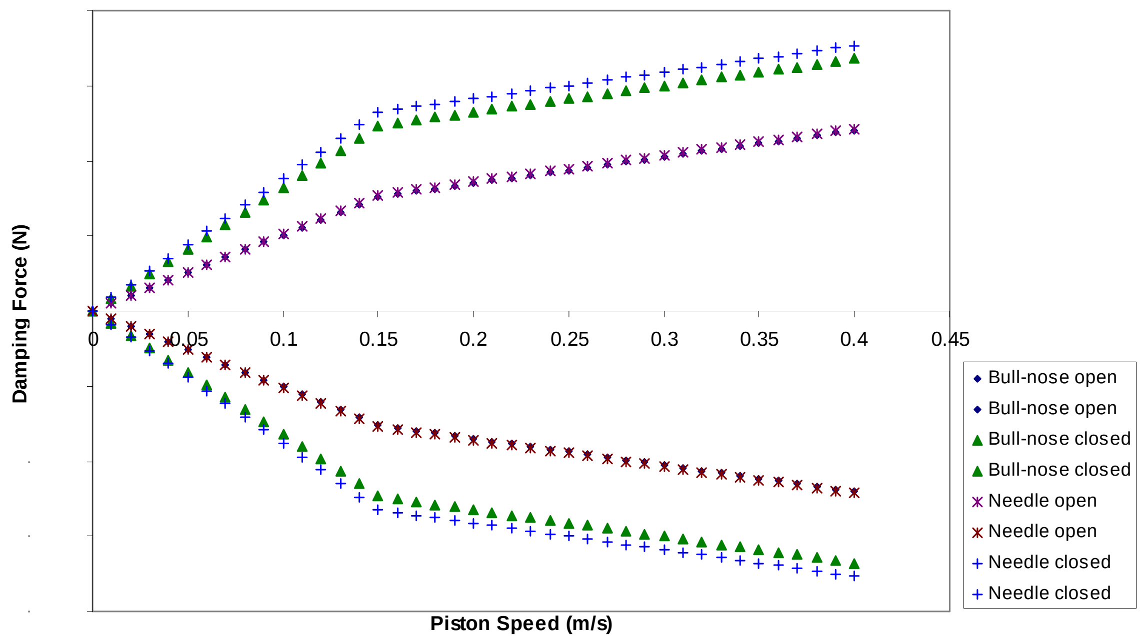


Figure 30: Constructed F-V curve created by superposition of results

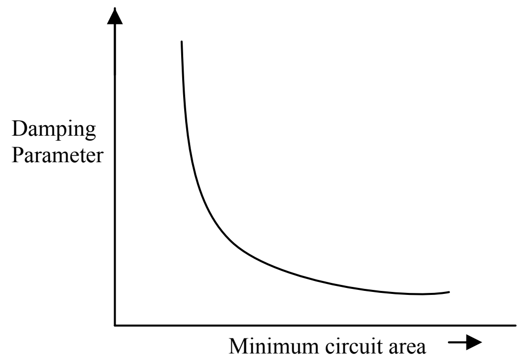
The graph features a considerable initial slope that transitions to a more shallow high speed slope, which is a found in many racing dampers. Force magnitudes are larger than what is expected by a factor of 3, most likely due to the aforementioned assumptions made during modeling and geometric discrepancies. One can see that if the transition speed (where the high speed curve takes over) were lowered, the curve magnitudes would be much lower; unfortunately the transition speed is a function of shim stiffness, which was neglected in the analysis. Also, the valve orifice is still open during high speed flow, and thus the high speed drag force is overestimated.

Recommendations for CFD modeling

As shown in the report, there are many assumptions made to complete our model. However, given the overall complexity of the problem, it is more comprehensive to the engineer to be able to break the problem down into different modes of operation [regimes]. Therefore, given the amount of computing power available, a different model would need to be made to model certain activities. Our models can be used as building blocks into other models. For example, to model the compression adjuster (see Figure 19), we would initially use a totally solid piston/shaft and disregard the volume on the underside. The deflecting shim valving on the base valve would be closed, and we would need to model the movement of the floating piston either with a constant pressure outlet, or a boundary UDF.

For temperature/multiphase related issues, we would need to obtain further properties about our working fluid. For petroleum based fluids, this can be very difficult because the conductivity, specific heat, viscosity, and density will all vary with temperature. All of these properties would have to be obtained through experiment, or some assumptions will have to be made. For damper design, this plays a large role in the durability and consistency. We would want to know how long the damper will last, and at what point do its performance characteristics begin to fade.

In regards to external adjusters, the damping parameter [slope of Force vs. Velocity] may be able to be predicted based on the minimum area observed through the bleed circuit. If the circuit is modeled as a series of resistances, then the dominating resistance plays the largest role in determining the flow. If the flow is reduced, damping is increased. This would be quantified in the following graph.



Other geometry parameters will play a role, so the relationship may not reduce to a single curve, but it will be helpful in the design process.

Because the dampers have not yet run on the vehicle it was necessary to perform further simulations to evaluate how to scale the force vs. velocity curve in terms of compression and rebound damping. We mentioned earlier in the engineering analysis section about having a damper that is stiffer in rebound than compression. After further analysis, this kind of damper may not be well suited for our application. The primary reason to do this is to provide a car that rides well over road imperfections. Since the surfaces that we race on are very smooth compared to the average Michigan road, this benefit is of no real use to us. In fact, it could be detrimental to vehicle performance. Because of the low un-sprung mass associated with our car, and the high damper forces in rebound, most FSAE cars lift either the front or rear inside wheel during cornering. This is not good because obviously you want all four wheels to be in contact with the racing surface (and evenly loaded) at all times to ensure optimal performance, as a result, we created another model to investigate the effects of an offset damper curve on sprung mass dynamics in a handling situation.

Non-linear Handling Model

In order to investigate the effects of damper characteristics on response of the FSAE racecar, it is necessary to simulate the dynamic handling of the vehicle and be able to visually interpret the response. It would be quite simple to perform such an analysis in one of the many commercially available suspension analysis software, however we want all significant model assumptions and limitations be well understood. Therefore it was decided that a simulation model be created using Matlab and Simulink where all major assumptions would be realized. Specifically, the "Nonlinear 3DOF Handling" model developed for the ME542 lecture would be modified to allow for non-linear damping and the creation of shock velocity histograms.

Simulink Model

A state-space method was employed to create and solve the equations of motion of the racecar. In the state-space (or time domain) approach, state-variables are arranged in a matrix format to produce a system of linear 1st order differential equations of the form:

$$\dot{\mathbf{x}} = A\mathbf{x} + B\mathbf{u}$$

where ***x*** is a state-space vector representing the state-variables of the system and ***u*** is the vector of external inputs. Note that ***u*** is distinct from the forward velocity of the CG, *u*. Here *A* and *B* are the linear coefficient matrices of the state-vector and input vector, respectively. By solving this equation, the response of the vehicle due to external inputs can be determined. However, the above equation is valid for a linear system only; therefore any nonlinear effects must be resolved outside of the state-space model and entered as an input.

A 3 DOF handling model was implemented, consisting of roll, yaw, and lateral velocity degrees of freedom. This results in a state and input vector:

$$\mathbf{x} = \begin{matrix} r \\ v \\ \phi \\ p \end{matrix}, \quad \mathbf{u} = \begin{matrix} F_{yF} \\ F_{yR} \\ F_{D,FL} \\ F_{D,FR} \\ F_{D,RL} \\ F_{D,RR} \end{matrix}$$

where *r, v, φ,* and *p* are the yaw velocity (rad/s), lateral velocity at the CG (m/s), roll angle (deg), and roll velocity (rad/s) about the roll axis. F_{yF} and F_{yR} are the front and rear axle lateral forces, and F_D is the vertical damping force at each wheel. Since both the damping force and tire force generation is nonlinear, these effects cannot be included in the state-space model, but are determined using a separate Simulink model, and treated as an input.

This model was developed in a similar way to the models discussed above in the *Engineering Analysis section*. The major difference is the introduction of nonlinear tire and damper characteristics. The state space analysis is still employed to develop the linear portion of the equations of motion, while non-linear effects are treated as inputs to the system. A schematic of the Simulink diagram is shown in Figure 31.

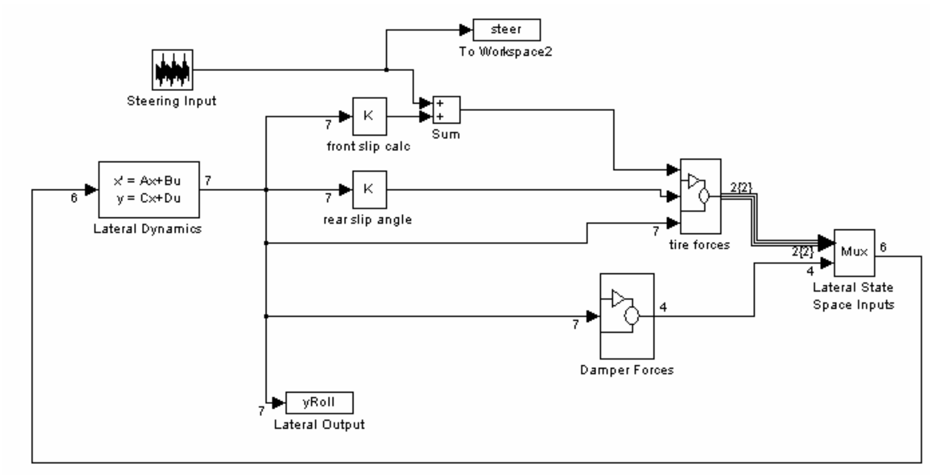


Figure 31: Main Simulink model, with nested "Tire Forces" and "Damper Force" Model

Steering angle at the front wheels, δ, is the sole external input to the system, and is generated by a user-defined function which is representative of open-loop driver control. This is summed with the front axle slip angle and input into the "Tire Forces" model, which produces lateral tire forces at the front and rear axles. Along with damper forces at each wheel, these forces are input into the aforementioned linear handling dynamic model, which outputs the 4 state-variables as well as front and rear roll moments and the lateral acceleration of the CG. Roll angle and lateral acceleration are not used by the model but are graphed for response visualization.

The "Damper Force" model takes roll velocity p as the sole input, and converts it into a linear velocity at each wheel. This velocity is then input into a look-up table defining damper force, $F_{D,I}$ as a function of velocity, according to the damper curve specified. Damper force at each wheel is then output to the dynamic state-space model. The Simulink model is shown below in Figure 32.

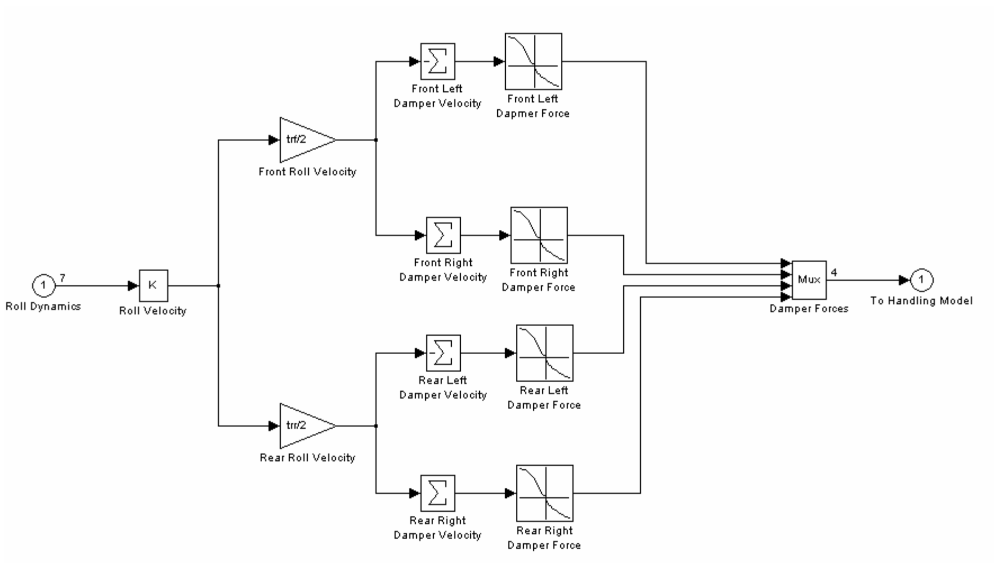


Figure 32: Damper Force model generates damping force at each wheel externally of the handling model.

Similarly, the tire force is generated using a 2 dimensional lookup table. First outputs from the state space model are analyzed and used to model the effects of weight transfer, which will ultimately change the force generated at each tire. Slip angles for the front and rear (calculated based on the vehicle velocity condition) are applied and a side force is calculated and then applied to the state space model. A schematic of this Simulink model is shown below in Figure 33.

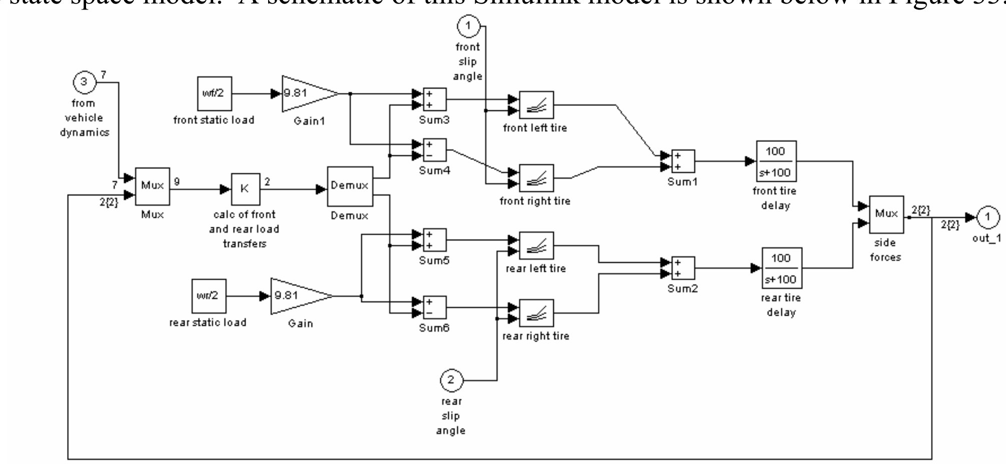


Figure 33. Schematic of nonlinear tire force generation Simulink model

Data Acquisition Implementation

Results from the model can be compared with physical data acquired from the 2004 University of Michigan FSAE racecar, as a means of validation. The 2004 vehicle was equipped with a large amount of data acquisition including:

- lateral and longitudinal accelerometers
- wheel speed sensors (front and rear)
- steering rack potentiometer
- suspension potentiometers (all 4 wheels)

amongst many other chassis and engine sensors. Particularly useful to this analysis is the steering potentiometer, which allowed actual data from a full lap to be input into the Matlab simulation. By running the simulation with physical inputs, the model response can be compared to the actual vehicle response to determine its accuracy. Physical data was taken from a test session at the Bosch Proving Grounds on April 18th, 2004; a track map of the course is shown in Figure 34.

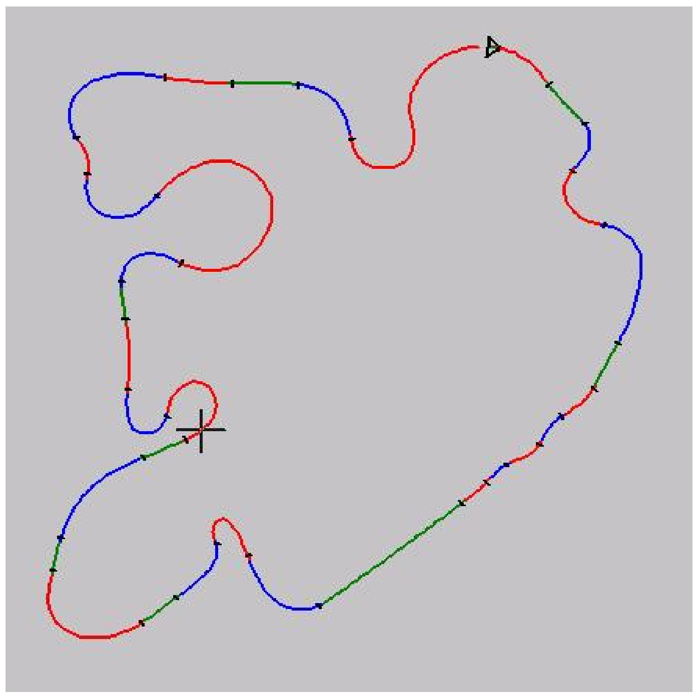


Figure 34: Map of the course at the Bosch Proving Grounds during a 2004 test session.

There is a large range of speeds and turn radii providing many different handling cases, thus there is little concern about the input data being unrealistic. Steering data is plotted versus time in Figure 35 with time beginning at the lap start, denoted by the black triangle.

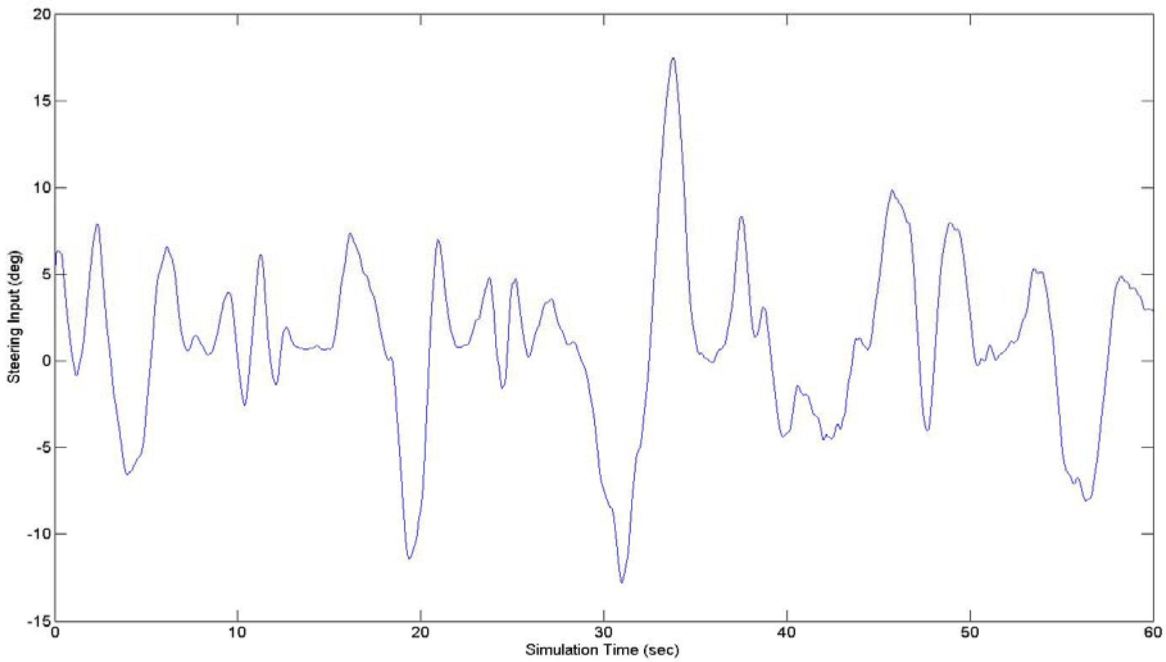


Figure 35: Plot of steering data obtained by the 2004 racecar, input into the model.

Inputting physical steering data into a simulation model will not provide realistic results if the assumptions made in creating the model are not treated. One of the major assumptions made in the analysis is the constant vehicle speed. Obviously any racing vehicle negotiating the track shown in Figure 34 will not be traveling with constant forward speed (*U*), however the inclusion of *U* as a state variation adds tremendous complexity to the model, as described above.

Further complicating the problem was that if *u* was chosen to be large then the vehicle would not be able to negotiate the course successfully. If *U* is chosen to be too small, then the simulated results will not be representative of the data acquired from the physical vehicle. By examining the velocity recorded on the 2004 race car, a forward speed, *U*, of 10 m/s was chosen as representative of the entire lap. This speed is slightly faster than the slowest speeds on the track, but somewhat slower than the average lap speed. By setting *U* to 10 m/s we obtained a response that is representative of reality, without causing an unstable vehicle condition in the simulation. Figure 36 shows the vehicle velocity recorded over one lap in relation to the limit of our simulation.

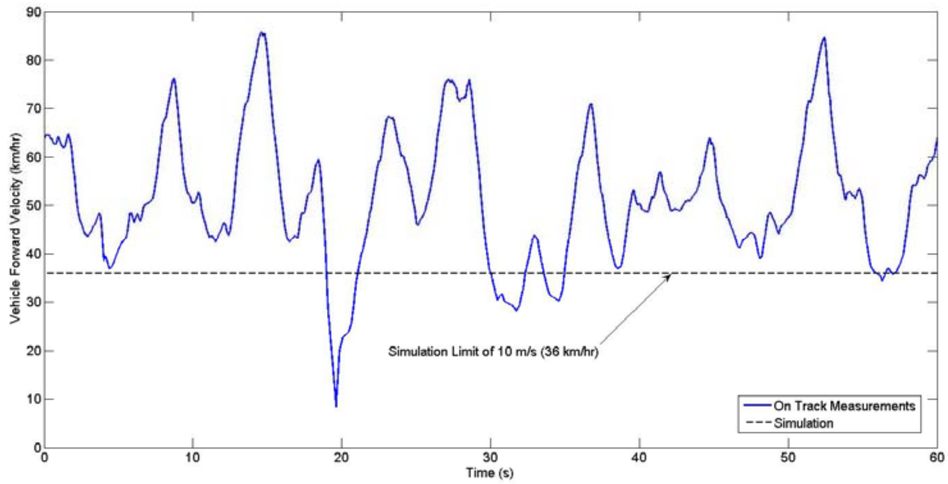


Figure 36: Vehicle speed (front wheels) of 2004 racecar with forward speed limit of simulation denoted by the dashed line.

Results

In order to verify that our model is providing a reasonable approximation to what is occurring on track, we will compare the simulated vehicle lateral acceleration to the acceleration actually measured on track in 2004. As demonstrated in Figure 37 below, the general trend appears to be correct. The magnitudes are scaled; however this is due to the fact that in the model, the forward speed of the vehicle is fixed. We can however conclude that the simulation is a rough approximation of the vehicle on track.

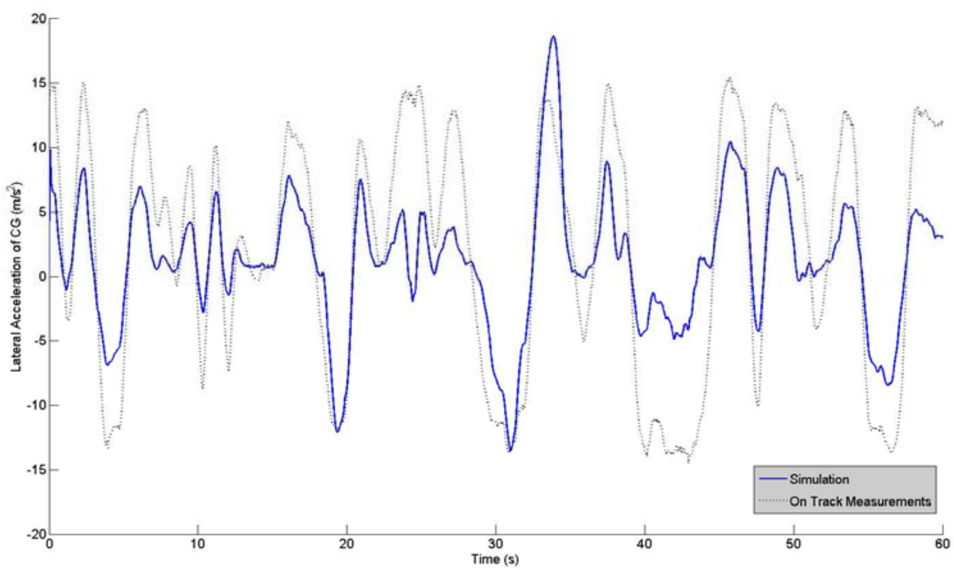


Figure 37: Simulated lateral acceleration vs. real world measurements

In general, the vehicle acceleration of the CG has three terms as shown below.

$$a_{yCG} = \dot{v} + Ur + \dot{p}h1$$

The $\dot{v}$ and $\dot{p}h1$ terms (which represent the derivative of lateral CG velocity and angular acceleration of the CG about the roll axis times the roll moment arm) will be well represented by our model, however the centripetal term *Ur* will not be captured well because our model assumes a constant forward speed, *U*. This can be seen in Figure 37 because the lateral acceleration magnitude is not consistent with the on track data in corners where the forward speed of the vehicle is significantly higher than what our simulation allowed. It is good news that the trends are similar in both the simulation and measured data.

Since the model appears to be behaving correctly, we can now use it to make some conclusions based on its performance. We looked at two cases at the extremes. In the first case, the damper is only able to dissipate energy in the compression stroke, and in the second case it can only dissipate energy in the extension (rebound) stroke.

Looking at the roll angle response, we can see a difference in the two cases. In general, the "compression only" damper responds to steering inputs faster than the damper with a "rebound only" characteristic. Figure 38 below shows the roll angle response of the simulation with two different nonlinear dampers.

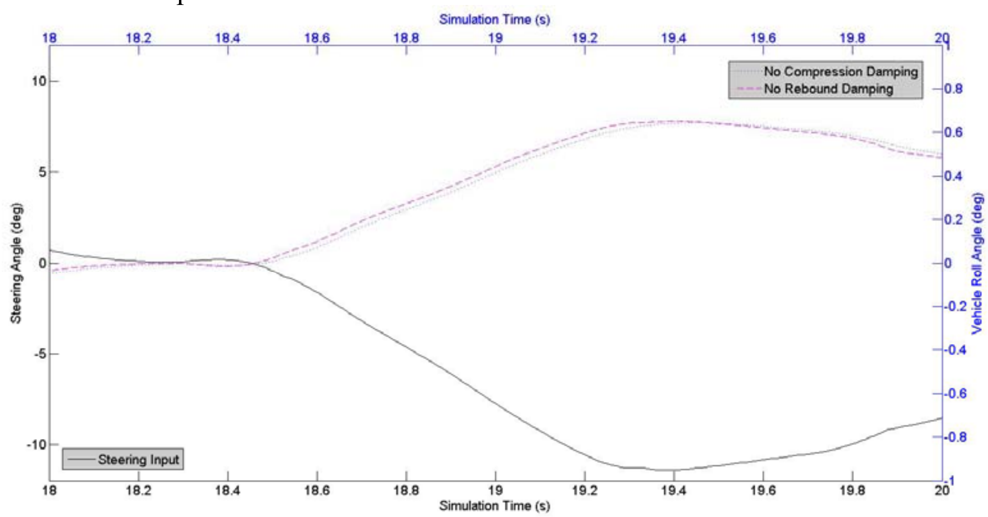


Figure 38: Roll angle response with two different dampers

It is obvious that the "compression only" damper exhibits a faster response to the steering input. This trend is also evident in other corners on the track, Figure 38 shows just one example corner so that the effect can be seen. This graph shows that the vehicle responds faster, and settles down more quickly when you only remove energy on the compression stroke. Obviously it is not ideal to build a damper this way, because there are cases where you do want to remove energy on the extension stroke. Another thing that we notice is that the difference is relatively small.

With this in mind, we have decided to build the final damper spec in a roughly symmetric manner and use the adjustments to alter the proportion of rebound and compression damping. Figure 39 below shows the final damper target curves developed with the nonlinear handling model and the dynamometer output from tests conducted at Pratt and Miller.

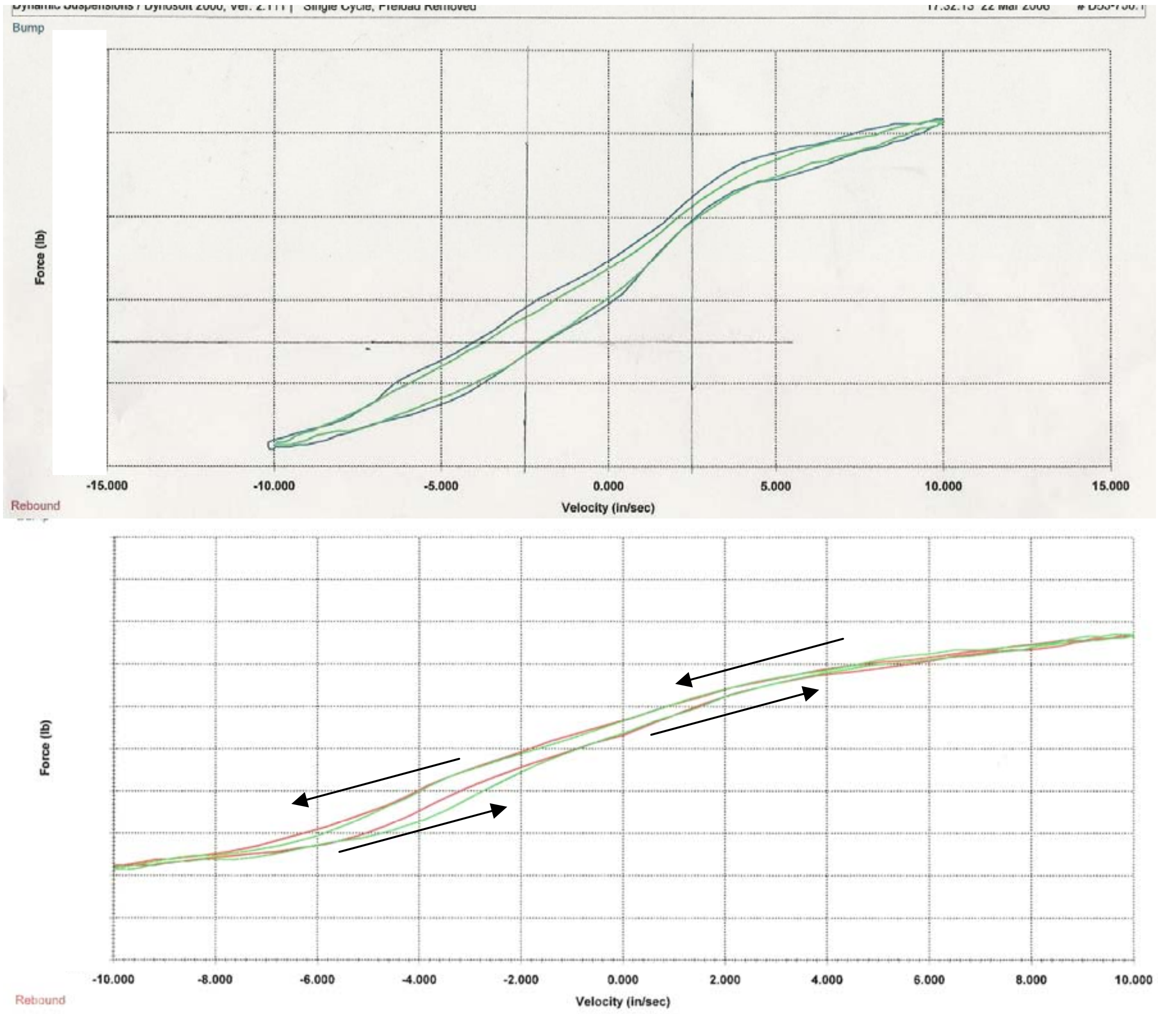


Figure 39: Force vs. Velocity relationship of damper prototype. Upper graph is from the initial build, while the lower graph was measured a week later.

These graphs are obtained by starting at the fully extended position, and then completely compressing the damper and returning back to the original extended position while measuring the force continuously. Since the actuator follows a sinusoidal pattern, the entire range of speeds (from 0 to 10 in/s) is observed with maximum velocity occurring at the mid-stroke position. The lower graph features a different shim stack (8 stacks have been tested so far), new shock oil and larger bleed paths through the adjusters. Each plot features two tests taken at the same settings. There is a significant reduction in hysteresis in the second plot, which is defined as the force offset between each direction of travel. This is due to the increased orifice area through the piston and valves as well as the reduced shim stack overall stiffness; the remaining hysteresis on the rebound stroke can be removed with a reconfigured rebound adjuster which we will test at our next session on April 24 2006.

As mentioned before, we were unable to test the dampers on the car at the time of this writing; however they will be tested for the first time during a session on April 27 2006. The ultimate criterion for evaluating the damper performance is the lap time of the racecar. We expect our dampers to provide the driver with a more predictable vehicle. This is especially important considering the relative skill of our drivers. Because our car will only be as fast as the driver, their comments will be used to shape future developments to the damper. Also, we will use various data acquisition tools to evaluate the performance of the dampers on track. This system is a set of potentiometers that span each damper and measure damper displacement versus time, distance, or frequency. With this data, important relationships such as shaft velocity histograms can be examined. By examining histograms and making external adjustments to a damper, its performance can be best tuned for given track conditions and driving styles (see Figure 40). The data acquisition is especially important as a means to verify the driver feedback.

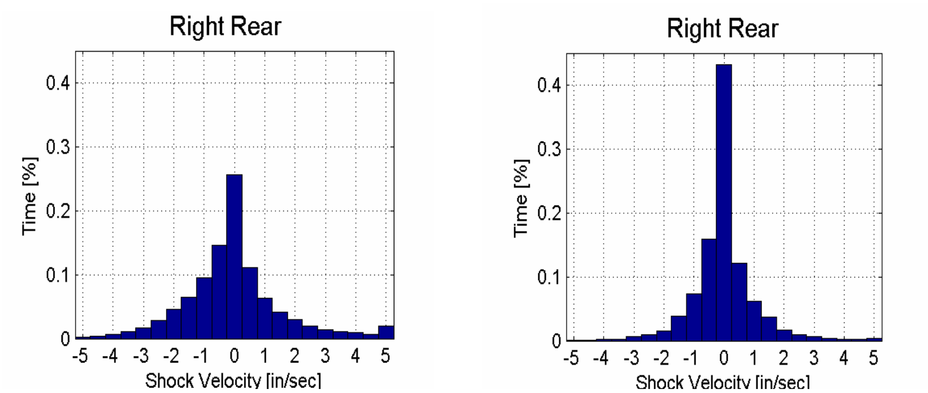


Figure 40: on track data used to tune damper for best performance before (left) and after (right) adjustments

Design Critique

We would label the 2006 MRacing damper project as a success; ultimately there are 8 dampers that we'll be able to run at competition that will improve the vehicle's performance in all events.

The damper operates, in that it seals and should last a race season. This was a concern at the beginning of the project because of some of the machinery processes used. A damper that does not seal would be grounds for failure at competition. Furthermore, this would lead to a deterioration of the damper performance over time, which would violate certain customer requirements.

Damper performance is adequate. The force vs. velocity characteristics are very close to the values we desired based on our analysis (see Fig 39). This tells us that the internal valving (read shim stack) is appropriate and functioning well. Also, the flat force vs. distance curve indicates that the damping force does not depend on the position of the damper and that the damper does not demonstrate a large amount of sensitivity to piston acceleration (see Fig 41). In comparison to previously used mountain bike dampers, we were able to decrease the amount of hysteresis by about 70%. Finally, the damper fits well in the 2006 vehicle, despite foreseen tight packaging constraints.

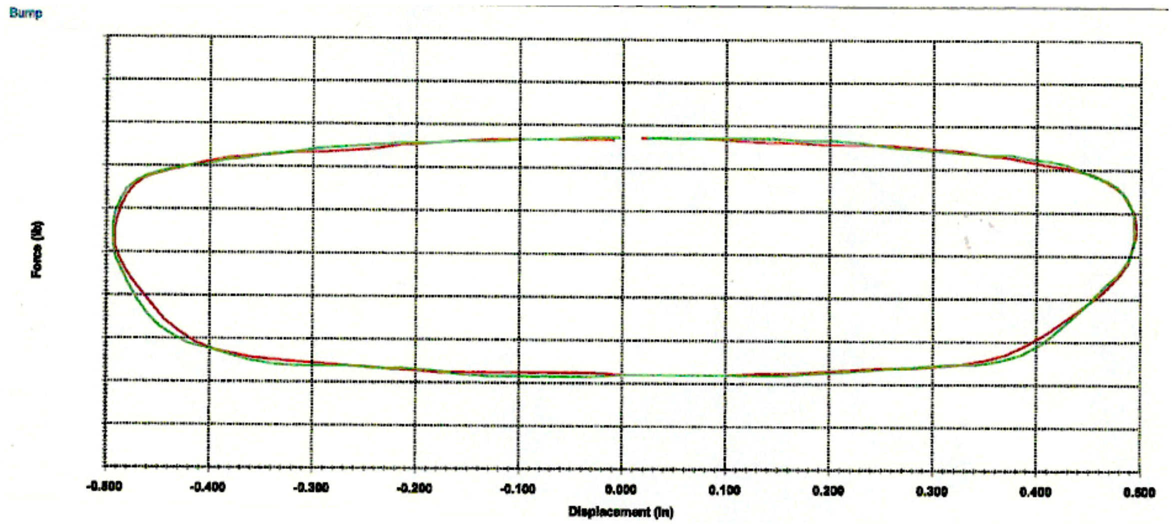


Figure 41: Force vs. Position for the final FSAE Damper shim stack

The items discussed above make this project a huge success and a tremendous step forward in terms of suspension design for the M*Racing* team. There are, however, a few areas that we feel could specifically be improved:

Adjustability

The 2006 dampers feature two external adjustments which operate independently of one another. These adjustments do function, however they do not provide as much of a range of adjustments as we hoped. During our validation process, we had to compromise the adjustment effectiveness to prevent cavitations and/or compression of the fluid at certain settings. Also, the design of the rebound needle valve needed to be modified to prevent the valve from becoming permanently lodged in the shaft. A spring was added to prevent this from happening; however the spring does change fluid flow in the area slightly. As a note for teams who plan to continue this work, the adjustment method needs to be reconfigured to allow for a greater range of adjustment. While the current range will allow the car to be fine tuned to a certain extent, it does not allow for the total damping characteristic to be drastically altered.

Reservoir Design

The design of the reservoir is another area of the design that could be improved. The floating gas piston functions well, however it does add to the friction of the system. Furthermore, the assembly process is complicated because the position of the floating piston is critical, if the piston is too high, there will not be enough fluid to allow for full rebound travel, while to low & the piston will run out of travel before maximum compression has been reached. An interesting solution to this problem is applied on a damper made by Stratos (see Fig. 42 below). This solution uses a bladder to separate the liquid-gas interface and will ensure consistent assembly as well as eliminate any friction provided by a floating piston setup. There are some problems with this solution, namely selecting a bladder that does not allow "seepage" of the gas charge into the fluid volume causing emulsification; however it is an interesting solution to be considered.

Figure 42: Example of a "bladder" to contain the pressurized gas charge.

Conclusion

The Formula SAE competition places an emphasis on vehicle handling through various regulations as well as the nature of the competition. As a result, teams have begun to develop new solutions to improve vehicle handling with dampers. To stay ahead of our competition, we will design and construct a custom damper for our vehicle and race it at the FSAE competition in May 2006. Our design will improve on the standard mountain bike damper by implementing functionality of some high end racing dampers and implementing it into a package that is sized appropriately for FSAE competition. The end result will be a damper that offers performance benefits on track as well as a boost in the design event.

Recommendations

For future MRacing teams we feel there are 3 specific areas where our damper performance is lacking. These areas are damping hysteresis, adjustability, mass and hydraulic fluid. These are the area's we recommend future developments be concentrated, at least initially.

As mentioned in the design critique, while we did meet our targets for friction/hysteresis, there is still room for improvement. One possible solution is the use of a bladder to replace the current floating piston design and providing a larger bleed path through the base valve. Regarding adjustability, we were not quite satisfied with the total range of adjustment afforded by our current design. Basically, the problems were a result of not enough bleed, which resulted in a large spike in force and the need to isolate the compression adjustment at the base valve. One possible solution to this problem is to make use of a check valve (similar to the rebound adjustment) which would allow for a bleed path independent of the adjustment as well as further isolate the compression and rebound adjustments. Two things that we did not fully investigate were the choice of piston and hydraulic fluid. We recommend that future MRacing teams use fluid dynamics analysis to evaluate the effects of different kinds of piston as well as some different hydraulic fluids. While we are pleased with what we were able to achieve, there is probably some room for improvement in this area.

Acknowledgements

We would like to thank all of the sponsors who helped us throughout this project, without their generosity this project would not have been possible.

Arvin-Meritor provided invaluable access to a strong technical knowledge base on damper and suspension technology. Much of the initial design was done with feedback from engineers at Arvin-Meritor. They also donated the material used in manufacturing the shafts.

Motorsport engineers at Pratt and Miller provided intellectual feedback and access to a damper dynamometer. From which, we were able to validate our design and better assure damper performance was suited for our application.

Liberty's Gears donated their services for the high tolerance, complicated machining on the damper body and upper mount. This included machining the piston bore and the stub acme thread on both the damper body and upper mount.

Alro generously donated all of the aluminum we used for the damper project. We wanted to be able to have 10 fully operational dampers assembled at any one time. This required an large and expensive amount of aluminum. They donated over 16 feet of 2.75" OD 7075 Al. Using this material was crucial in meeting engineering specifications of the damper.

Kennametal provided all of the tooling necessary to complete the damper manufacturing. Specifically, they provided a wide array of solid carbide end mills, lathe tools (grooving inserts, etc.) Without access to these tools, we would not have been able to manufacture some of the more complicated parts, nor would we have been able to hold the tight tolerances necessary.

Busak-Shamban supplied our team with seals for the rod guide.

G&L tooling donated their services by machining both the reservoir body and cap. These parts involve high tolerance dimensions and some areas requiring 'mico' smooth surface finishes.

Tokico provided us with the shims that we used for the base valve and piston shim stacks. Furthermore, they provided technical support to our team as well as access to their shock dynamometer.

Aeroquip provided a discount to our team on the lines used to connect the damper body to the remote reservoir.

Steve 'Vice-Grip' Emanuel

Without 'Vice-Grip' our project simply would not have been possible. He allowed us access to the graduate machine shop and put up with our shenanigans throughout the manufacturing process. Vice has put in a lot of long hours helping us with manufacturing this project and other projects associated with FSAE. Thanks for all the work Vice, we really appreciate it.

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