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Formula SAE Shock Absorber Design

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“Sometimes I think that I would have enjoyed racing more in the days of the friction shock. Since you couldn’t do anything much to them or with them, I would have spent a lot less time confused.”

Carroll Smith

Tune to Win, 1978
ABSTRACT

The design and manufacture feasibility of a damper customised to a Formula SAE style vehicle has been considered in this paper. Research into the history of dampers, racing dampers and the concept of the damper in Formula SAE style vehicles, governs the direction of the new design. Vehicle dynamics theory and incompressible fluid flow theory have been applied through Simulink and Matlab models for specific component design. 3D CAD of the proposed design has been developed according to the design targets specified by the numerical models. Manufacturing and cost issues have been explored, with the conclusion that manufacturing a set of dampers at the University of Queensland would be technically feasible.
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1. INTRODUCTION

For the past three years, the University of Queensland has entered the Australasian Formula SAE competition, achieving mixed results. In 2003, the design philosophy of the car was altered, the team aim to construct a simple, lightweight and reliable car. After achieving an overall rank of 3rd at the competition, the 2004 team chose to continue to refine their designs while keeping with the same overall design philosophy. This decision lead to the team seeking increased technical depth in many areas, with shock absorber development being of primary importance.

Except for go-carting, all forms of motorsport incorporate various damping arrangements into their designs for the same reasons as passenger vehicles; to control unwanted oscillations arising from the suspension springs. Formula SAE is no different, however, the diminutive size of the vehicles pose significant challenges to the suspension designer regarding shock absorber placement and appropriate damping control.

1.1 Aim

This thesis takes the form of a feasibility study investigating the design and manufacture of shock absorbers for a Formula SAE vehicle. Through research, data acquisition and computer modeling, this thesis will determine the whether significant benefit can be gained by manufacturing a set of shock absorbers to suit a Formula SAE vehicle.
1.2 Objectives

The primary objective of this project is to determine the benefits associated with manufacturing customized dampers. To achieve this, the following questions must be answered,

- What deficiencies exist within current the design?
- How can these be improved?
- How difficult are the components to manufacture?
- How do the purchase and manufacturing costs compare?
- Do the performance gains outweigh the time invested for manufacture?

1.3 Scope of Work

The scope of this work is bounded by the following statements,

- The design must comply with the rules set out by SAE International for the 2004 Australasian competition.
- The benchmark for design and analysis will be the shock absorbers currently in use on the 2004 vehicle.
- The shock absorbers are designed for the track conditions expected during testing and competition with the Formula SAE vehicle.
1.4 Procedure

To achieve the project goals, a design based on research into current technology will be formulated and benchmarked against the shock absorbers used on the 2004 vehicle. The work can be divided into seven sections as follows,

- Background Research
- Component Analysis
- Applicable Theory
- Data Acquisition & Analysis
- Numerical Modelling
- Detailed Design
- Recommendations & Conclusions
2. **BACKGROUND RESEARCH**

Background research was conducted to gain an understanding of the history of damper design, current damper technology and trends in Formula SAE vehicles. Furthermore, this initial research would provide direction for the design phase of the project. Presented henceforth is a brief discussion of the findings of the research.

2.1 **History of Dampers**

The need to arrest unwanted oscillations from suspension springs dates back to the beginning of the automobile era. Dampers were introduced between the sprung and unsprung masses to control these oscillations, with the first primitive “racing” units being installed on the 1906 French Grand Prix Renault. Forces were generated by passing a vane or piston through a fluid. The 1920’s saw the advent of friction dampers and rotary vane style units. In the early 1930’s, Munroe adapted the telescopic design damper found in aircraft landing gear, for automotive application. This design was widely adopted throughout the automotive and racing industry, thus will form the basis of the design presented in this project. (Milliken, 1995)

Several different types of telescopic damper exist, dual tube, monotube and monotube with a floating piston separating oil and gas. Figure 2.1 shows the dampers construction of each type.
Figure 2.1 – Damper arrangements
2.2 Racing Dampers

A limited number of companies exist globally which produce automotive style dampers, an even smaller number of which have racing interests. Companies such as Ohlins, Sachs and Penske produce some of the highest quality racing dampers, their products being widely used in the upper echelons of the motorsport fraternity. These units also come with a price-tag which is prohibitively expensive to anything other than a fully fledged race team. The expense of these specialized units stems from their ability to provide a wide range of adjustments, their excellent build quality and the company’s reputation.

The need for adjustable shock absorbers was recognized early in auto racing history. The ability to be able to adjust the performance characteristics of the car to different track surfaces and even different weather conditions, can mean vital lost lap time. Shock absorbers will generally be adjusted in pairs, front or rear, to affect the transient balance of the vehicle. To achieve accurate damping adjustment between a pair of shock absorbers, a high level of build quality must be maintained.

2.3 Mountain Bike Dampers

The damper trend in recent years at the Australasian Formula SAE competition has seen many teams adopt mountain bike dampers for their suspension designs. This trend is driven by the fact that these dampers are relatively inexpensive at US$350 per unit when compared to a full race spec damper, offer some level of adjustability and are of
relatively lightweight and compact design. The one characteristic that is often overlooked or overshadowed by the sum of their other virtues, are the dampers’ force-velocity characteristic. When the damper is viewed in its intended application, it becomes readily apparent they are designed to operate under conditions of rapid shaft speed and consequently produce high damping forces. One would conclude that the adjustability provided by the manufacturer would hence be tailored to the high shaft speed region of the dampers’ force-velocity curve.

In 2003, the UQ team chose Risse Racing Jupiter 5 shock absorbers, figure 2.2, for their suspension design. The team has continued to use these dampers in 2004 while further investigation and testing is completed. These dampers were initially chosen for the following reasons,

- 2 ¼” of mechanical travel
- Independent damping circuits for compression and rebound
- Externally adjustable compression circuit
- Externally adjustable rebound circuit
- Large piston area allowing wide damping adjustment range
- Gas charged reservoir eliminating cavitation
- Adjustable spring preload
The shock absorbers employed in the suspension design of the UQ Formula SAE vehicles is a variation on the design shown in figure 2.1(c), with the floating piston being relocated to a separate reservoir in an effort to reduce the overall length of the unit.

2.4 Shock Absorber Concept & Formula SAE

The presence of dampers in Formula SAE suspension designs is articulated in chassis section of the competition rules. Specifications on permissible suspension travel are also outlined in the clause,

3.2.3 Suspension

The car must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50.8 mm (2 inches), 25.4 mm (1 inch) jounce and 25.4 mm (1 inch) rebound, with driver seated. The judges reserve the right to disqualify cars which do
not represent a serious attempt at an operational suspension system or
which demonstrate unsafe handling.

One of the largest restrictions placed on the design of a Formula SAE vehicle, and hence
the shock absorber selection or design, stems from the following two rules,

3.5.1.1 Engine Limitations

The engine used to power the car must be a four-stroke piston
engine with a displacement not exceeding 610 cc per cycle. The
gine can be modified within the restrictions of the rules. If more
than one engine is used, the total displacement can not exceed 610
cc and the air for all engines must pass through a single air intake
restrictor (see 3.5.4.3, “Intake System Restrictor.”)

3.5.4.3 Intake System Restrictor

In order to limit the power capability from the engine, a single
circular restrictor must be placed in the intake system between the
throttle and the engine and all engine airflow must pass through
the restrictor. Any device that has the ability to throttle the engine
downstream of the restrictor is prohibited.

The maximum restrictor diameters are:

Gasoline fueled cars - 20.0 mm (0.7874 inch)
The design implications of these rules and track specifications listed in Appendix A, force Formula SAE vehicles to be constructed such that the minimum possible weight is achieved, as the maximum attainable engine power has been limited by volumetric displacement of the engine and the mass flow rate of air it can draw through the restrictor. One way to reduce the overall mass of the vehicle is to package the components tighter, hence produce a physically smaller design. When this idea is applied to suspension design, the engineer faces a significant challenge when positioning the shock absorbers. Damper placement issues around the driver compartment and around the engine bay abruptly present themselves, as shown in figure 2.3, leading to the requirement that the specified damper be as small as possible.
3. Definitions

The following section is included to provide the reader with a point of reference as the technical aspects of damper design are discussed henceforth.

3.1 Shock Absorber Components

The telescopic damper is made up of several key components, each performing a specific role. Brief descriptions of listed components follow,

- Body
- Piston
- Valve
- Main Shaft
- Adjusters
- Reservoir

3.1.1 Body

The body of a racing telescopic shock absorber performs several unique functions and is composed of several pieces. Primarily, the body must contain the fluid being used to provide the damping force, usually oil. The inner face of the body also forms the sealing surface at the extremity of the piston and the body must be strong enough to withstand
the hydrostatic pressures induced by compression and extension of the main shaft. Furthermore, the body must provide support for the main shaft against loadings that contain components misaligned to the plunge axis. In modern racing shock absorbers, the body also provides provision for the coil spring mounting and preload adjustment as well as the mounting point to the chassis, see figure 2.2. Figure 3.1 shows a cutaway view of a damper body.

### 3.1.2 Piston

The piston divides the body into two sealed oil chambers. Provision for oil to flow between these two chambers is accommodated for via holes, called ports, in the piston, see figure 3.2. The shape and size of these ports determine the high speed characteristic of the shock absorber. To facilitate different damper characteristics in both compression and rebound, the ports are positioned such that they are covered by valves in one flow direction. The extremity of the piston accommodates a sliding seal to the inner face of the damper body. The faces of the piston may also be dished to provide preload for the valve stack, see figure 3.3.
Figure 3.2 – Piston Ports

Figure 3.3 – Piston Dish
3.1.3 Valve

The role of the valve is to control oil flow through the piston. Valves take the form of annular discs with varying thickness and diameter, and are often stacked on top of each other. This arrangement is known as the valve stack and is shown in figure 3.2. The valve stack controls the transition from low speed to high speed damping.

3.1.4 Main Shaft

The main shaft or rod shown in figure 3.1, serves to connect the piston to the shock absorber eyelet mount. The eyelet is connected to the vehicle’s suspension, providing freedom for the unsprung mass to move relative to the chassis. In adjustable dampers, the main shaft usually houses the low speed rebound adjuster. The main shaft must have sufficient inertia to withstand buckling loads imposed by road surface inputs.

3.1.5 Adjusters

Adjusters take two forms, oil metering and valve stack. Oil metering adjusters are used to control the low speed damping characteristics of the shock absorber. Oil metering adjustment is achieved with needle and seat style valves, see figure 3.1, with the needle moving further from the seat as less damping force is required. Valve stack adjusters control the amount of preload the valve stack sees, and hence, the pressure required to open them. An example of a valve stack adjuster is shown in figure 3.4. Valve stack adjusters are used to alter the transition to high speed damping.
3.1.6 Reservoir

Reservoirs are secondary chambers attached to the body of the some dampers via either rigid or flexible couplings. The reservoir provides the ability to add pressurization to the damper, reducing cavitation, while maintaining a minimal overall length. The oil is separated from the gas by a floating piston, which is free to move along the axis of the reservoir and avoids the gas forming an emulsion with the oil. Compression adjusters are also housed in the reservoir in adjustable shock absorbers, see figure 3.4.
3.2 Suspension Components

Suspension components used in forthcoming discussions will be described here for the readers’ benefit.

3.2.1 Rocker

Aerodynamic considerations and the quest for better suspension geometry have forced open wheel racecar designers to accommodate the vehicles’ shock absorbers inboard of the suspension links, onto the chassis. This is achieved with a bellcrank or rocker, which translates linear motion of the pushrod into rotational motion. The rocker is effectively a lever, and hence the motion of the damper relative to the pushrod is governed by the lever ratio $L_1/L_2$ of the rocker, as shown in figure 3.5, called the motion ratio.

Figure 3.5 – Rocker & Motion Ratio
4. APPLICABLE THEORY

This section describes the relevant theory that should be considered when selecting or manufacturing a shock absorber. Beginning with a consideration of the shock absorbers role in the vehicle, the section continues investigating the dynamics of the shock absorber and finally exploring the applicable fluid theory.

4.1 Dynamics of the Car

There are several different models of varying complexity which can be considered when investigating the role of the racing shock absorber. Each of these models can be used to gain insight into how the vehicle will respond to different road surface or driver inputs. An explanation of the listed models considered in this project will follow.

- ¼ car 1 or 2 DOF models
- ½ car model - pitch
- Complete car model - roll

4.1.1 Quarter Car Models

The simplest of all applicable damper models is the quarter car model. Quarter car models consider only one quarter of the car as a dynamic system, meaning that the model captures the effects of road surface inputs for any single wheel on the vehicle,
irrespective of what the others are experiencing. Hence, conclusions about vehicle performance parameters from this model should be used with care.

The most common quarter car models are either single or dual degree of freedom. These models consist of a spring, mass and damper with the extra degree of freedom coming with the removal of the assumption of a rigid tyre. An illustration of the quarter car model is given below, see figure 4.1.

![Basic Quarter Car Model](image)

**Figure 4.1 – Basic Quarter Car Model**

While this model is a serious oversimplification of the actual system, it is the basis for some fundamental results. These first of these is the natural frequency of oscillation for the system. Expressed mathematically as,

\[
\omega_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \text{ (Hz)}
\]

where \( k = \text{spring stiffness (N/m)} \)
m = sprung mass (kg)

The natural frequency represents the rate at which the undamped system would oscillate for a given road surface force input. In vehicle design terms, the natural frequency is used by engineers as a crude estimate of the overall stiffness of the suspension. The measurements are subjective and vary between sources, however frequencies from 0.8 – 1.5 are generally considered to give soft suspension characteristics, 1.5 – 2.0 for more sporty equipment and 2.5 and above where the focus of the suspension designer is on handling. As a point of interest, the UQ Formula SAE car currently runs at around 3 Hz.

Another important relationship that can be defined from this model is that of the damping ratio. The damping ratio, $\xi$, is a ratio between the damping coefficient $C$ (Ns/m) and critical damping, $C_{crit}$ (Ns/m). Critical damping is defined by Milliken as the level of damping that allows a mass to return to its steady state position with no overshoot, given a step input (Milliken, 1995). Mathematically,

$$\xi = \frac{C}{C_{crit}} = \frac{C}{2\sqrt{km}}$$

and is equal to one for critical damping. The damping ratio characterises the vehicles transient response to a step input and gives the designer an easily quantifiable metric to use. Figure 4.2 illustrates how the displacement of the system mass varies with the time normalised to the period of the undamped natural frequency.
Figure 4.2 – Varying Damping Ratios

\[
x = \frac{t}{t_n} = \frac{\text{Time}}{\text{Period of Undamped Natural Frequency}}
\]
An interesting point to note is that the first point at which the initial overshoot is the only oscillation occurs at approximately $\xi = 0.7$. Various sources in literature and from industry suggest that achieving this damping ratio in heave (vertical motion of the car) works quite well.

### 4.1.2 Half Car Model

The half car model, sometimes called a bicycle model, expands on the quarter car model by coupling two via an assumed rigid chassis. This model allows the designer to gain insight into the pitching behaviour of the vehicle as it passes over disturbances. While passenger car designers must provide suitable damping coefficients to ensure supple ride characteristics, racecar designers are more focused on the effect the dampers will have on the handling of the vehicle. For instance, instability under brakes can be caused by excessive damping forces, excessive front rebound can cause the front to wander and excessive rear rebound can unsettle the rear, making the car prone to spinning.

The solutions to the pitch and displacement equations governing the half car model are time dependent, and hence better suited to computer simulation. A Simulink model of this has been developed and will be discussed in the following chapter.
4.1.3 Complete Car Model

While not explicitly detailed within vehicle dynamics literature, the concept of a complete car model is not difficult to imagine. The model involves the expansion of the half car model to simulate the vehicle turning, allowing the designers to estimate damper speeds and forces during cornering. This information is of primary importance to the design of the shock absorber, as their main function in racing is to control the transient behaviour of the car. Understanding the operating range of the shock absorbers allows the designer to size oil metering orifices and valve stacks appropriately, such that maximum performance and adjustability can be incorporated into the design. UQ Racings complete car model is based on the diagram shown in figure 4.3 and was constructed in Simulink by Francis Evans for ease of alteration and computation.

Figure 4.3 – Complete Car Model (numbers can be disregarded)
4.2 Dynamics of the Damper

Dampers provide a resistive force to counteract the release of stored energy in a spring by passing oil through small passages. The viscosity of the oil causes a pressure drop which in turn, produces a force acting on the piston. The forces produced in each stroke can be tailored to similarity and dependence or uniqueness and independence depending, on the construction of the damper.

4.2.1 Damper Motions

There are two basic motions in damper operation; compression, where the overall length of the shock absorber is decreasing and rebound, which is the opposite. As the damper compresses, oil in the body ahead of the piston becomes pressurized, while the motion of the piston causes the oil behind to become slightly depressurized. The volume of oil in the body is also reducing as a function of damper compression due to the volume of the main shaft entering the body. The volume of oil displaced by the main shaft is passed through the compression adjuster, while the oil in the body is passed through the piston, as illustrated in figure 4.4. For the damper to operate correctly, the pressure behind the piston must be sufficient to resist cavitation, requiring the compression adjuster to have similar resistance to the compression valve stack when compared on a basis of volumetric flowrate. Similarly, in rebound the pressure drop across the rebound adjuster must be overcome by the reservoir pressure to ensure cavitation free operation.
4.2.2 Cavitation

The occurrence of cavitation is caused by a flow condition resulting in the liquid pressure of the oil being dropped below its vapour pressure. The result is the localised appearance of gas bubbles and an inconsistent and unpredictable damping characteristic as shown in figure 4.5. Cavitation has also been shown to accelerate component wear. As the gas bubbles formed in low pressure regions pass into high pressures they collapse, imploding on themselves. In the shock absorber, this localised decrease in liquid pressure must be overcome by the gas pressure in the reservoir, to ensure cavitation free operation.

Figure 4.5 – Cavitation on Damper Trace
4.2.3 Compressibility

As the damper moves, hydrostatic forces generate stress in the liquid. Although usually assumed incompressible, strictly speaking, liquids are compressible to some extent. This compressibility can have effects on both viscosity and density. According to Dixon, oil viscosity may vary by $\sim 3\%$/MPa, due to the geometries commonly found in the flow passages, this has little effect as the static pressures drop significantly at the entrance. Dixon also states that the density of a pure liquid is effected by pressure changes, but the effect is quite small $\sim 0.04\%$/MPa. When a practical liquid is considered however, this effect may be greater, and is a common source of hysteresis in dampers. For the purposes of this thesis, compressibility effects will be assumed to be negligible.

4.2.4 Temperature

Dampers provide a resistive force by passing oil through small passages. While the velocity of the piston is constant, the resistive force will remain the same. The energy associated with creating the resistive force is dissipated as heat, which raises the temperature of the oil in the damper. Oils are inherently temperature sensitive, their viscosity varying greatly between $0^\circ$C and $120^\circ$C. Figure 4.6 shows how SAE grade 10 oil commonly used in dampers, varies over this temperature range. Given the size and packaging constraints imposed upon the suspension engineer in Formula SAE, it is not uncommon to see dampers mounted close to large heat sources such as engines, exhaust
systems or brake rotors. Given such large heat sources, the operating temperature of the dampers can reach 60°C or more at the rear of the vehicle, while at the front the dampers may operate much lower. For this reason, the temperature effects on damper behaviour cannot be ignored and will be included henceforth.

Figure 4.6 – Viscosity Variation With Temperature
4.3 Fluid Dynamics

This section outlines some of the fluid dynamic considerations to be taken into account when designing a damper. Consideration will be given to the effect of the Reynolds Number, Conservation of Mass and Bernoulli equations, entry and exit losses and finally Bernoulli Orifice Plate Theory. From this and previous considerations, a numerical representation of the shock absorber will be created and used to determine component designs.

4.3.1 Reynolds Number

The Reynolds Number is a dimensionless flow parameter used to describe the nature of the flow. Three possible flow states exist, laminar flow, transition and turbulent flow. The state the flow is in has varying effects on the forces produced in the damper, as turbulent flow for example is characterised by different flow equations to that of laminar flow. The accepted values for the three flow states are as follows;

- Laminar \( Re < 2000 \)
- Transition \( 2000 < Re < 4000 \)
- Turbulent \( Re > 4000 \)

Mathematically,

\[
Re = \frac{\rho V D}{\mu}
\]
where $\rho = \text{density (kg/m}^3\text{)}$

$V = \text{characteristic velocity (m/s)}$

$D = \text{characteristic length or diameter (m)}$

$\mu = \text{absolute viscosity (Ns/m}^2\text{)}$

The Reynolds Number usually determines the friction factor in both laminar and turbulent pipe flow. Friction factors are generally considered in pipe flows longer than those encountered in damper design, usually only a few diameters. For this reason the Reynolds Number will only be used here as a qualitative check, and it’s affects on friction factor will be assumed minor compared to entry and exit losses.

### 4.3.2 Bernoulli Equation & Conservation of Mass Law

The Conservation of Mass Law says that for a given control volume, the sum of the masses entering the volume must equal the sum of the masses exiting. Assuming a steady, incompressible flow, the Conservation of Mass Law can be expressed mathematically as,

$$\sum_i (V_i A_i)_{\text{out}} = \sum_i (V_i A_i)_{\text{in}}$$

where $V = \text{velocity (m/s)}$

$A = \text{area (m}^2\text{)}$
The Bernoulli Equation is an expression of energy conservation in a fluid flow, assuming constant density (Dixon, 1999). It is valid along any flow streamline provided losses are negligible. Mathematically, the Bernoulli equation can be expressed as,

\[ P_1 + \frac{1}{2} \rho u_1^2 + \rho g h_1 = P_2 + \frac{1}{2} \rho u_2^2 + \rho g h_2 + \Delta P_{St} \]

where \( P = \) pressure (Pa)
\( g = \) gravitational acceleration (m/s\(^2\))
\( h = \) height (m)
\( \Delta P_{St} = \) stagnation pressure change (Pa)

For small height changes such as in dampers, the Bernoulli equation can be simplified by eliminating the height term on both sides of the equation. The \( \Delta P_{St} \) term is added to the right hand side of the equation to account for minor losses, such as entry and exit losses. The equations in this section will form the basis for flow rate and consequently force estimations produced by the numerical model of the shock absorber.

### 4.3.3 Entry & Exit Losses

Entry and exit losses are part of a larger group of fluid flow effects known as “minor losses”. Other examples of minor losses include change of section and bends, but as such geometries are rarely found in damper design, this section will focus on entry and exit losses only.
Entry losses are caused by small vortices forming inside just inside the contraction of an oil passage. The result is a reduced flow area with an effective diameter, known as the ‘vena contracta’, which is less than that of the passage. Figure 4.7(a) shows how flow streamlines compress around abrupt entry geometry and the vortices that form. Note how vortex generation is reduced by providing a rounded entryway to the passage, figure 4.7(b).

Figure 4.7 – Entry Losses

Figure 4.8 shows loss coefficient values, K, for different inlet geometries. K is related to the change in stagnation pressure through the following relation,
Exit losses arise from the sudden expansion in section, relative to the flow direction. The sudden expansion causes vortices to form at the exit of the smaller diameter section as shown in figure 4.9. This flow condition will occur at the exit of the needle type adjusters. Figure 4.10 shows loss coefficients for varying exit geometry. Exit loss coefficients are related to a change in stagnation pressure similarly to entry loss coefficients.
The exit of the oil ports in the piston are usually covered by the valve stack, only opening when the pressure difference across the piston assembly overcomes the valve stack preload. The valve stack is restrained centrally at a certain diameter, known as the...
bending diameter. Oil pressure is directed at certain points around the valve, causing the valves to deflect, opening the port to some effective area. The proximity of the valve to the port and the geometry of the port will govern the exit loss coefficient in this flow situation. Modelling the complex flow relation around the valve stack would require either extensive CFD to capture the three dimensional nature of the flow or experimental testing to generate accurate loss coefficients. For this reason, the loss coefficient for flow in this region will be assumed to be unity.

4.3.4 Bernoulli Obstruction Theory

A second method for calculating the flow through the piston would be to consider it as a thin-plate orifice, to which Bernoulli obstruction theory could be applied. Initially, this method would seem more valid than that proposed previously, being able to account for discrepancies between the model and the actual device with the use of a dimensionless parameter called the ‘discharge coefficient’. Concerns with using this method include the requirement the ratio $\beta = d/D$ on which the discharge coefficient depends must be within the range $0.2 – 0.8$. While this may be achievable for some designs, the common damper sees ports positioned differently to the geometry given on pg 399 in White. A further requirement that would be difficult to meet is that of the thickness of the plate to not exceed $0.05*D$. This would create impractically thin pistons which would deform under the pressures being generated within the damper, severely affecting damper performance if seal contact is lost. For the above reasons, no further investigation of this method sought.
5. Numerical Modelling

Numerical models serve the purpose of aiding in the design process by allowing the designer to estimate different performance characteristics of the design. They also allow theory to be matched to physical results where actual testing can be conducted. The numerical models used in this project include the half and full car simulink models created by Francis Evans in early 2004, before this project started and a numerical model of the damper which I created specifically for this project.

5.1 Simulink Models

For the purpose of this project, the half and full car models have been used to estimate the damper characteristics under different operating conditions. Images of the simulink models can be found in Appendix B. The models make the assumption the there is minimal compliance in the suspension components other than the spring, and that Coulomb friction forces in the suspension joints are small enough to be ignored. Also, no aerodynamic forces have been considered at this stage.

5.1.1 Motion Studies

The dynamic behaviour of the vehicle working as a system of many different components, yields a wide variety of responses over its wide range of possible motions. For the purpose of simplicity, only the major motions of the vehicles operation have been
considered. The models have been used to estimate the force that the dampers are required to produce in the following situations,

- Acceleration/Braking
- High Speed Corner Entry
- Low Speed Corner Entry
- Bumps

Results from these motion studies will be compared to data recorded from the car during testing and used to as design targets for the new shock absorber.

5.1.2 Results

The following results have been calculated using suspension parameters from the 2004 vehicle, which is similar in design to the 2003 vehicle. A damping ratio of $\xi = 0.7$ has been used in the calculations based on discussions with industry professionals. The simulink models output traces similar to that shown in figure 5.1
Figure 5.1 – Simulink Outputs

The results from each of the motion studies listed above have been collated in table 5.1

<table>
<thead>
<tr>
<th>Situation</th>
<th>Speed (mm/s)</th>
<th>Force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acceleration/Braking</td>
<td>50</td>
<td>125</td>
</tr>
<tr>
<td>High Speed Corner Entry</td>
<td>12.7</td>
<td>36</td>
</tr>
<tr>
<td>Low Speed Corner Entry</td>
<td>50</td>
<td>125</td>
</tr>
<tr>
<td>Bumps</td>
<td>280</td>
<td>560</td>
</tr>
</tbody>
</table>

Table 5.1 – Motion Study Results

Table 5.1 concurs with literature statements about the operating ranges of dampers. For situations involving chassis control, such as corner entry, low damping speeds and moderate forces dominate. For situations involving wheel control, like hitting a sharp rise in the track, high piston speeds and high forces are required. Based on the information
from the motion studies, a target of 125N at 50mm/s, with a transition region above
50mm/s and a linear force-velocity characteristic across the dampers range of operation,
has been set for the new damper design. These numbers represent those to be achieved at
the middle of the adjustment range of the damper.

5.2 Damper Model

To aid in the design of the shock absorber, a numerical model has been created to
estimate its performance characteristics. As previously mentioned, probably the most
useful data output form to consider as a race car engineer is the force-velocity plot. The
engineer can quickly establish what force is being produced by the damper at a particular
speed, and hence the numerical model of the shock absorber is structured to output results
in this form.

5.2.1 Aim

The aim of the numerical model is to produce a force-velocity plot, such that different
design geometries can be evaluated, and how viscosity changes which are proportional to
temperature, affect the forces produced by the damper.
5.2.2 Assumptions

To numerically model the performance of the shock absorber, a number of assumptions have been made concerning the nature of the fluid flows within the damper and the mechanics of the damper.

- The oil within the damper is assumed to be incompressible over the normal range of damper operation
- The damper is assumed to have cavitation free operation over the normal range of operation
- The mechanics of the damper are assumed to produce negligible friction forces
- The flow is also assumed to be frictionless
- The flow is assumed to be steady
- The Bernoulli equation can be applied accurately to a passage with an small length on diameter ratio

5.2.3 Simplifications

Simplifications have been made to limit the complexity of the model. The simplifications have been made such that the model is still able to give a reasonable estimate of the forces and geometries required, however some degree of calibration may be required before a complete set of dampers is to be produced.
To compensate for the presence of the needle valve in the low speed adjuster, a ‘bleed port diameter’ is specified in the model. By altering the diameter of the port, the effective flow area is changed and hence the force is altered.

To accommodate for the variability in valve stacks and preload, pressures at which the valves are in the state of fully open or closed are specified. The valve is assumed to increase the effective area of the port in a linear fashion between these two points. If the pressure at which the valve is fully closed equals zero, the valve stack has no preload.

5.2.4 Limitations

The numerical model presented in this project has certain limitations which need to be highlighted.

The current model is applicable to dampers constructed with cylindrical port geometries. For this to be changed, the area calculations throughout the program must be edited.

The model produces the force-velocity characteristic by calculating flowrates and pressures generated by initial piston speed, in discrete steps, plotting to form a curve, and hence is not time based. Without this time dependence, the heat that is generated cannot be estimated and hence, temperature effects on the oil and gas in the damper can only be modelled for a single operating temperature. Comparisons of the effects can be made over multiple plots.
5.2.5 Results

The force-velocity output from the dampers numerical model is shown in figure 5.2.

![Damping Force vs Piston Velocity graph](image)

Figure 5.2 – Damper Model Output

This model shows good correlation to the expected output as outlined in Dixon, and shown in figure 5.3. While the expected output in Dixon shows a pressure-flowrate characteristic, both of these quantities are directly related to force and shaft velocity respectively.
Figure 5.2 clearly shows how the damper transitions from low speed, through the transition region and then into high speed damping. One inconsistency that was noted was that depending on the values used for the valve stack opening and closing pressures, the change to and from the transition region become rather abrupt. This changeover point would most likely be smoother in the actual damper.
6. **DATA ACQUISITION & ANALYSIS**

To accurately predict the size of components required in the new design, loading data was required. The data will then be analysed to specify a load case for the design of the new damper. Data was recorded both on the track using the team’s data logger and software and with the dampers out of the car, using an SPA Dynamometer at Fulcrum Suspensions at Moorooka.

6.1 **Dynamometer Testing**

To test the hypothesis that mountain bike dampers are not suitable for the purposes of Formula SAE, dynamometer testing was conducted on the Risse dampers used by the team. All four dampers were tested and their results compared to previously recorded by the 2003 team (Burt, 2003).

Inspection of the force-velocity graphs recorded in May 2004 showed that after approximately 50 hours driving, the damper oil and seals had degraded to the point where a noticeable difference in force production at the same settings. One of the dampers was also losing fluid around the compression adjuster at the full soft setting, indicating a blown seal. On the whole, the dampers had softened, each setting recording a slightly lower force at the same velocity. This showed that for the operating conditions imposed by Formula SAE a high quality, lightweight, synthetic oil would be beneficial as it would have better resistance to breaking down chemically, providing better wear resistance. The
dampers were fitted with new seal kits and filled with Motul Synthetic Fork Oil with a 5w viscosity rating.

Re-testing of the dampers confirmed the feedback from drivers, indicating the car seemed to perform the best with the dampers towards the lower end of their adjustment. Figure 6.1 shows an example of one of the shock absorbers force-velocity characteristic where the damping targets of 125N at 50mm/s are met on only the second click off minimum, hence the dampers have virtually no ability to go softer if required. Figure 6.1 also shows how the shock absorber provides good adjustment at high speed, concurring with the adjustment required for their intended application. As the Risse dampers offer only compression and rebound adjustment, any change to settings also affects both the high and low speed ranges of the damper. The compromise chosen by the team this year was to focus on achieving the damping target as specified above at low speed, at the expense of roadholding ability at high piston velocities.
Another point which must be highlighted is the variation in results between the four dampers. All four dampers produced different forces at the same setting. Inspection of the internals during fitment of the seal kit revealed nothing untoward about the dampers, each consisting of the same components. This leads to the conclusion that the manufacturing tolerances that are enforced during production of these dampers is larger than what is required for dampers on race vehicles. Again, this is most likely a function of their intended application which requires the use of the dampers as single units, rather than in pairs or quartets as in a racecar application. This degree of variability between the dampers makes adjustment to track surface conditions difficult, requiring the user to
memorise the different characteristics of each damper. This operation should take only
seconds as equal incremental increases or decreases should be available on the adjusters.

6.2 Track Data Acquisition

To gain an appreciation of how the car is using the dampers, data logging must be
carried out while the vehicle is in motion. This data can be used to corroborate or
contradict the design specifications predicted by the numerical models of the car.

Budget constraints and a tight project schedule have meant that comprehensive data
logging has not yet been possible on the 2004 car. Data recorded late last year from the
2003 car will be used as a supplement. While not specifically conducted for the purpose
of this thesis project, the data logging process was conducted as I partook in the process.
The lack of reliable strain gauge data has also forced the estimation of forces by other
means.

To acquire the necessary data, 0-5v, 360 degree, rotational potentiometers were installed
on the suspension rockers, as in figure 6.2. Due to the size of the components involved, a
gearing of 1:1 was chosen. As the swept angle of the rocker is quite small in normal
operation, approximately 15 degrees, electrical amplification of the signal was provided
in the data logger. This enabled the team to detect finer movement of the dampers due to
larger changes in resistance. Logging of the data was completed using four channels
simultaneously, logged at 120Hz to avoid any aliasing affects. Prior to commencing
recording, each channel was tuned such that the potentiometers were in the middle of their range as seen by the data logger.

![Potentiometers On The Car](image)

**Figure 6.2 – Potentiometers On The Car**

Labview software developed by Larry Weng, was used to present the recorded data. The software takes the text file output from the data logger and converts it to a shock speed histogram. The shock speed histogram gives an indication of how long the shock absorber is spending in the compression and rebound stages of its stroke, and also an indication of what shaft speeds are being reached. An example of the shock speed histogram is shown in figure 6.3. Using the adjusters on the shock absorbers, the dampers can be adjusted such that equal time is being spent in compression and rebound to avoid
such effects as jacking down, where the rebound force is too high causing the car to not completely return to its ride height before the next bump is encountered. Jacking down would otherwise be described as being overdamped in rebound.

For each adjustment a new set of histograms was recorded along with the adjustments made to the shock absorber. This can be used in conjunction with the data recorded from the dynamometer and the raw data stream to estimate the load applied to the shaft. This load would be calculated as follows;

- For a particular damper setting, harder equalling more internal stress on components, the raw data stream is recorded and the histograms produced.
The raw data stream is then analysed, particular attention being paid to spike changes in recorded voltage.

The rise time of the largest spike is measured, knowing the data logging frequency and then compared to the highest speed on the shock speed histogram.

As the largest spikes occur over bumps in the road as opposed to vehicle manoeuvres, and the unsprung mass related to that particular channel is known, the application of Newton’s Laws generates the force input to the shaft as follows;

\[ v = u + at \]
\[ F = m\ddot{x} + c\dot{x} + kx \]

rearrange to give;

\[ F = \frac{m(v-u)}{t} + F_{\text{Damper}} + k\int_0^t v \]

where \( m = \) unsprung mass (kg)

\( v = \) final velocity (m/s)

\( u = \) initial velocity (m/s)

\( t = \) rise time (seconds)

\( F_{\text{Damper}} = \) damper force from dyno graphs (N)

\( k = \) spring rate (N/m)

The following values have been taken from recorded data to give an estimate of the load seen at the shaft;

- \( m = 11 \text{ kg} \)
• $v = 0.5 \text{ m/s}$
• $u = 0 \text{ m/s}$
• $t = 0.1 \text{ seconds}$
• $F_{\text{Damper}} = 535 \text{ N}$
• $k = 40910 \text{ N/m (250 lb/in)}$

The force input to the main shaft as calculated from these numbers is approximately 2636 N. To maintain a constant force of 535 N, the piston velocity must remain constant, such that the force above the piston, $F_{PC}$ and the damping force, $F_{\text{Damper}} (F_{PC} - F_{PE})$, equal the input force, $F_D$, otherwise acceleration would occur. Hence, the main shaft acts as a pinned column with an axial buckling load of twice the input force, see figure 6.4. Extra frictional and atmospheric forces shown in figure 6.4 have been neglected in the analysis due to relatively small magnitudes.

Figure 6.4 – Damper force balance

The shock speed histograms also show the dampers recording velocities in the low speed range of the dampers for a large portion of the sample. This validates the design
specification predicted by the numerical model, emphasising the importance of platform control in this type of vehicle. We have now achieved a design targets on which to base the new design which are as follows;

- Maximum input force of 2650 N
- 125 N damping force at 50mm/s
- Transition region beginning at >50mm/s
- Linear force production over the range of operation of the damper
7. **Detailed Design**

This section will explore the design of a new damper, based on the loadings and performance characteristics identified in previous chapters. General design considerations will be considered first, followed by actual component designs.

### 7.1 Design Considerations

To form accurate conclusions as to the feasibility of the production of a customised set of dampers, certain design issues must first be taken into account. The factors which require consideration include manufacturability, cost, durability, heat dissipation, assembly and disassembly procedures and sealing.

#### 7.1.1 Manufacturability

For the design of the new damper to be feasible, the design must be such that it can be manufactured, preferably in house at the university. As expressed previously, dampers require exacting tolerances to be adhered to if quality items are to be produced. The mechanical engineering workshop has the ability to CNC machine parts to excellent accuracy, such that I believe it would be possible to manufacture a set of dampers with the current tooling.
7.1.2 Cost

The overall cost of the dampers can be reduced if careful consideration is given to the component designs. One area where potential savings exist over purchased dampers is in assembly, with students being able to assemble to units when the components have been manufactured. An actual costing analysis of the damper production will be performed after the design has been presented.

In Formula SAE competition, teams are required to complete a cost report based on the competition rules. To summarize, purchased items must be costed at recommended retail price, regardless if the team received a discount from the supplier. For a manufactured item however, the cost of the item includes the raw cost of the material, the machining operations included and the labour to machine and assemble the component. If the team were to manufacture its own set of dampers, significant savings could be made to the final cost of the car, a figure worth 30/100 points for the cost event.

7.1.3 Durability

Dampers need to be designed with durability in mind as they from the compliant link between the suspension and the chassis. As dampers are usually one of the most expensive items on the vehicle, it is beneficial to be able to re-use them. To be able to re-use the dampers, they should be designed such that major components do not wear to the point where replacement is necessary. This may mean increasing the weight of some
components to extend their fatigue life and exerting higher tolerances on machined parts, both of which increase the cost of the damper.

7.1.4 Heat Dissipation

Dampers produce a resistive force by passing oil through narrow passages. As time passes, frictional forces within the fluid and damper mechanisms generate heat which raises the temperature of the oil. Short term temperature variations will affect the viscosity of the damper oil, in some cases drastically altering the performance of the damper. Long term thermal cycling of oil eventually degrades its performance as its chemical properties change, thus good heat dissipation prolongs the life of the damper, requiring less frequent maintenance. Heat dissipation away from dampers is usually left to the vehicle designer, who must provide adequate airflow around the unit.

7.1.5 Assembly / Disassembly Considerations

As the damper consists of many smaller components, due consideration must be given as to how the damper is going to be assembled or disassembled. Most components are circular by nature and hence threads are prolific. Accessing these threads, by virtue of being able to apply enough torque to tighten or loosen them, must be considered.
7.1.6 Sealing

Dampers generate resistive forces by generate large internal pressures. To contain the contents of the damper under these pressures, adequate sealing must be provided. Static seals usually consist of rubber O-rings fitting into machined groves with specific dimensions as to provide sufficient ‘squish’ to form a seal. Another type of seal often found in dampers is the sliding seal. Sliding seals are used around the piston, the main shaft and possibly in the external reservoir. These sliding seals usually preform dual functions, providing both a sealing surface and axial support for the particular component.

7.2 Component Design

The components for the new damper design will be discussed in this section. Only the major components will be discussed. Due to concerns about the accuracy of the numerical model, the damper components will remain in the concept stage, without final dimensions. All 3D CAD is available and will be relatively easy to adjust once the numerical model has been calibrated.

7.2.1 Piston

Figure 7.1 shows the proposed design of the piston for the damper. The piston has been designed such that the valve stack does not impede the oil flow when the piston is in the opposite half of its stroke. This flow restriction was found in the Risse dampers,
providing a flow path which is difficult to model due to difficult to estimate loss coefficients. The diameter of the piston has also increased over the Risse, growing from 24mm to 36mm OD. The increased diameter provides better low speed damping response through greater surface area and more uniform pressure distribution over the piston face. The larger diameter piston also allows for the fitment of larger needle adjusters in the main shaft, by increasing the mainshaft diameter, and enables the removal of restriction to the high speed oil flow path through the piston. Cylindrical ports of diameter 3mm were predicted by the model in Appendix C, to give adequate high speed performance and linear force characteristics. Chamfers have been placed at the entrance to each port to reduce the entry losses associated with a sharp contraction. The piston incorporates a Teflon sliding seal around its perimeter to seal the two chambers and provide axial alignment. 7075-T351 aluminium has been chosen as the material from which to manufacture the piston, due to its enhanced yield strength, providing good durability.

Figure 7.1 – Piston Design
7.2.2 Mainshaft

Figure 7.2 shows the proposed design of the main shaft. The main shaft is designed to incorporate both the low and high speed rebound adjusters. The main shaft diameter was calculated using the load case specified previously, using Euler buckling formulae for an axially loaded, pinned column. A safety factor of two has been incorporated into the calculation to account for the occasional collision with a pothole or alike. The calculation is as follows;

\[
P_{cr} = \frac{\pi^2 EI}{L^2}
\]

\[
I_{xx} = \frac{m}{8} \left(d_o^4 - d_i^4\right)
\]

This formula prescribes a 12mm OD and 7mm ID. 4340 steel has been chosen for the main shaft based on its increased mechanical strength over aluminium and fatigue life concerns. The main shaft will require indexing upon assembly to align the ports for the rebound adjuster.
7.2.3 Valves & High Speed Adjusters

Figure 7.3 shows the proposed valving arrangement. This arrangement allows the preload of the valve stack to be adjusted by turning a dial on the outside of the damper. Altering the preload alters the point at which the damper enters the transition region, hence delaying the onset of the high speed damping curve. The high speed damping curve itself cannot be adjusted. The valving arrangement shown works by the port covering shim deflecting to push a slider upon which the valve stack acts. The valve stack can be flexed towards the slider, adding preload to the valve.
To accurately size the shims that make up the valve stack, first one must consider how the oil jets act on the valve. The oil jets act concentrated at three axi-symmetric locations around the valve. As the oil flowrate requirements increase, the pressure acting on the valve from the port causes increased localised deflection than that of the neighbouring region as shown in Figure 7.4. Conversely, the shims acting on the slider see a uniform load distributed around the valve at some radius, causing uniform deflection. As modelling the discontinuities associated with the jet flows, the assumption will be made that these shims experience uniform deflection, similar to the others. The shims which constitute the valves will be modelled using the Classical Plate Equation from the Engineering Fundamentals Website – [www.efunda.com](http://www.efunda.com), listed in Appendix D.
calculations of shim thickness for the required opening and closing pressures lead to effective shim thicknesses of 1.2mm using a sheet spring steel. Shim thicknesses are equal in compression and rebound to facilitate equal adjustment quantities.

![Figure 7.4 – Single Valve Deflection](image)

### 7.2.4 Low Speed Adjusters

Figure 7.5 shows the proposed design for the low speed adjusters. The adjusters have a parabolic needle for fine adjustment control. A spring loaded ball valve is also included to aid in damper sealing response during rapid directional changes. Oil flows enter through the two ports on the side of the shaft, flowing around the needle and forcing the ball valve open and leaving through the exit port. The port areas have been calculated such that the area of the two smaller ports exceed that of the larger port by 10% to compensate for losses due to the vena contracta in the smaller port. The components in the low speed adjusters are machined from 7075-T351 aluminium in an effort to reduce the weight of the damper. The spring and ball valve are made from 4340 steel.
7.2.5 Body & Reservoir

The body and reservoir casings are composed of multiple pieces as illustrated in figure 7.6. The cylinders for the body and reservoir are constructed from seamless extruded aluminium tube for better heat dissipation. The internals of the tubes require a quality machined finish, preferably honing, to minimise the wear in the sliding seals. Tubes should be anodised to further improve their wear resistance.

The caps which seal the body and reservoir should be machined from 7075-T351 aluminium in an attempt to preserve the integrity of the threads during assembly and disassembly without increasing the weight of the components. The top cap on the damper body also houses a spherical mounting bearing. This cap must transmit the forces generated by the oil to the chassis through this bearing, and the force flow paths have been designed accordingly.
Overall, the damper is slightly longer and rounder at 265mm and 60mm, spring included, compared to the Risse at 190mm and 50mm. The expected damping performance and adjustability gains far outweigh the increased size, however. Packaging flexibility has been maintained through the use of a hydraulic hose between the reservoir and body of the damper.

Figure 7.6 – Complete Damper

7.3 Cost Analysis

To ensure the feasibility of actually producing the dampers, a cost analysis will be detailed henceforth. For brevity the analysis is tabulated in table 2. Labour is costed on a hypothetical mechanical workshop staff member whose annual salary is $50,000 and works 48, 36hr weeks each year, equating to approximately $30/hr. Purchased component
pricing and external manufacturing costs are based on team experience with the local manufacturing industry. Damper costing will be completed for one unit, and the cost multiplied to account for a full set.

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
<th>Cost ($)</th>
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<tbody>
<tr>
<td>Body</td>
<td>Material</td>
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</tr>
<tr>
<td></td>
<td>Cut, bore, thread</td>
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<tr>
<td></td>
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<tr>
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</tr>
<tr>
<td></td>
<td>CNC Milling - Profiling</td>
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</tr>
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<td>Body Lower Cap</td>
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<td>Needle Valves (2)</td>
<td>Material</td>
<td>10.00</td>
</tr>
<tr>
<td></td>
<td>CNC Turning - Face, turn, thread, part</td>
<td>10.00</td>
</tr>
<tr>
<td>Springs</td>
<td>Purchase</td>
<td>25.00</td>
</tr>
<tr>
<td>Ball Bearings</td>
<td>Purchase</td>
<td>10.00</td>
</tr>
<tr>
<td>Speedflow Fittings and Hose</td>
<td>Purchase</td>
<td>40.00</td>
</tr>
<tr>
<td>Seals</td>
<td>Purchase</td>
<td>20.00</td>
</tr>
<tr>
<td>Rod End</td>
<td>Purchase</td>
<td>30.00</td>
</tr>
<tr>
<td>Spherical Bearing</td>
<td>Purchase</td>
<td>20.00</td>
</tr>
</tbody>
</table>

400.00
Table 2 – Estimated Damper Manufacturing Cost

Without actually manufacturing the items, the above analysis serves only as an estimate. The manufacturing process relies heavily on the CNC capabilities of the mechanical engineering workshop for accurate, reproduction of components. Without this, the process should not be attempted as slight differences in tolerances between dampers can vary the performance between the units. This estimate places the cost of these dampers in a similar range to that of the Risse dampers, while providing greater adjustability and damper characteristics more suited to a Formula SAE style vehicle.
8. **Recommendations & Conclusions**

The design of the new dampers is such that manufacturing a set would be technically feasible with the universities current tooling. The proposed damper is slightly larger than the Risse, however is expected to provide superior damping control. A cost estimate, based on experience of dealing with local industry suggests that a high quality damper, with adjustment more suited to Formula SAE style vehicles, would be possible for about the cost of a new low-end damper.

While the design and manufacture of a new set of dampers is technically feasible, further design work must first be undertaken. If a team were to manufacture a set, I would advise the manufacture of a piston and a test rig initially, such that the numerical model can be calibrated to the actual design. From here, alterations can be made prior to manufacturing an entire unit. Also, without dependable data acquisition, numerical modelling of the vehicles behaviour serves as an engineering approximation and a starting point for vehicle setup. Consistent driving and lap times will provide the means of car setup from there.

Suspension design is so critical to the performance of any racing vehicle, that its parameters drive the design of most others. This requires the component designs to be finalised early in the design phase. As damper performance is critical to the transient balance of the car, the dampers characteristics and quality must be known before it can be included in the design. I would recommend that if this were to be undertaken that
provision be made to run other dampers in the case that problems occur, such that the schedule of the project is not delayed.
BIBLIOGRAPHY


Smith C., 1996, *Drive to Win*, Carroll Smith Consulting Inc, Palos Verdes Estates, Ca USA.


Staniforth A., 1999, *Competition Car Suspension*, Haynes North America Inc., Ca, USA

5.7.4 Endurance Course Specifications & Speeds

Course speeds can be estimated by the following course specifications. Average speed should be 48 km/hr (29.8 mph) to 57 km/hr (35.4 mph) with top speeds of approximately 105 km/hr (65.2 mph).

**Straights:** No longer than 77.0 m (252.6 feet) with hairpins at both ends (or) no longer than 61.0 m (200.1 feet) with wide turns on the ends. There will be passing zones at several locations.

**Constant Turns:** 30.0 m (98.4 feet) to 54.0 m (177.2 feet) diameter.

**Hairpin Turns:** Minimum of 9.0 m (29.5 feet) outside diameter (of the turn).

**Slaloms:** Cones in a straight line with 9.0 m (29.5 feet) to 15.0 m (49.2 feet) spacing.

**Miscellaneous:** Chicanes, multiple turns, decreasing radius turns, etc. The minimum track width will be 4.5 m (14.76 feet).
Half Car Model
Wholecar Model
APPENDIX C

%Program to display valve flowrates

clear all;

bpd = 0.003; %Bleed port diameter (m)
ppd = 0.003; %Primary port diameter (m)
pistd = 0.036; %Piston diameter (m)
mainshaft = 0.012; %Mainshaft diameter (m)
kvisc = 1.5e-5; %Kinematic oil viscosity (m^2/s) at 20 degrees C
density = 800; %Oil density (kg/m^3)
Cd = 0.7; %Discharge coefficient
n = 3; %Number of primary ports

Abp = pi*bpd^2/4;
Am = n*pi*ppd^2/4;
EPA = pi*(pistd.^2-mainshaft.^2)/4; %Effective Piston Area (m^2)
Pvc = 120000; %Function based on shim stack determining the change to transitory damping
Pvo = 3000000; %Function based on shim stack determining the change to quadratic damping with valve fully open
Kin = 0.5; %Entry loss coefficient
Kout = 1; %Exit loss coefficient

for V = 0:0.001:0.5;
    Q = V*pi*(pistd.^2-bpd.^2)/4;
    U_est = Q/(Cd*pi*bpd.^2/4);
    P_est = 0.5*density*(U_est).^2+(Kin+Kout)*0.5*density*(U_est).^2;

    subplot (3,1,1);
    plot(V,P_est)
    hold on
    grid on
    xlabel('Piston Velocity (m/s)', 'FontSize', 12)
    ylabel('Pressure Estimate (Pa)', 'FontSize', 12)

    if P_est <= Pvc
        U = Q/(Cd*Abp);
        Pdiff = 0.5*density*(U).^2+(Kin+Kout)*0.5*density*(U).^2;
        Fnet = Pdiff*EPA;
        Re = U*bpd/kvisc;
    end
if $P_{est} > P_{vc} \& P_{est} < P_{vo}$
    \[ Af = \frac{(P_{est}-P_{vc})}{(P_{vo}-P_{vc})}; \]
    \[ U = \frac{Q}{(Cd*(Abp + Af*Am))}; \]
    \[ P_{diff} = 0.5*density*(U)^2+(Kin+Kout)*0.5*density*(U)^2; \]
    \[ F_{net} = P_{diff}*EPA; \]
    \[ Re = U*(bpd+Af*n*ppd)/kvisc; \]
end

if $P_{est} >= P_{vo}$
    \[ U = \frac{Q}{(Cd*(Abp+Am))}; \]
    \[ P_{diff} = 0.5*density*(U)^2+(Kin+Kout)*0.5*density*(U)^2; \]
    \[ F_{net} = P_{diff}*EPA; \]
    \[ Re = U*(bpd+n*ppd)/kvisc; \]
end

%%subplot (3,1,2);
plot(V,Fnet)
hold on
%grid on
xlabel('Piston Velocity (m/s)', 'FontSize', 12)
ylabel('Damping Force (N)', 'FontSize', 12)

%%subplot (3,1,3);
%plot(V,Re)
%hold on
%grid on
%xlabel('Piston Velocity (m/s)', 'FontSize', 12)
ylabel('Reynold's Number', 'FontSize', 12)
end
% Program to calculate shim thickness based on opening and closing pressures

clear all;

density = 7850;       % density of material in shim (kg/m^3)
E = 200;            % Young's modulus for shim material (GPa)
v = 0.3;            % Poisson's ratio
Rl = 0.0085;        % Loading radius of shim (m)
Ro = 0.014;         % Outer radius of shim (m)
Ri = 0.004;         % Inner radius of shim (m)
w = 0.003;          % Max shim deflection at edge (m)

Pr = 3000000;        % Valve fully open pressure (Pa)

B = Ro/Ri;
p = Rl/Ro;
c2 = 0.25*(1-B^2*(1 + log(B)));
c3 = B/4*((B^2 + 1)*log(1/B) + B^2 - 1);
c8 = 0.5*(1+v+(1-v)*B^2);
c9 = B*(((1+v)/2)*log(1/B)+((1-v)/4)*(1-B^2));
l3 = (p/4)*((p^2+1)*log(1/p)+p^2-1);
l9 = p*(((1+v)/2)*log(1/p)+((1-v)/4)*(1-p^2));

h = ((12*(1-v^2)*p*Ro^3)/(E*w))*((c2/c8)*(c9*Rl/Ri-l9)-c3*Rl/Ri+l3)