University of Southern Queensland
Faculty of Engineering and Surveying

Investigation of a Variable Ride Height Suspension for an Automobile

A dissertation submitted by

Mr Joshua Graeme Walton

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towards the degree of

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ABSTRACT

This dissertation investigates a new way of varying the ride height for a passenger vehicle. It follows the design process from the first step of checking history, background and current development of suspension. Next it looks at building the concept of a new air spring which attempts to maintain a relatively constant pressure while increasing the area over which this pressure is acting. Material and manufacture process selection are conducted for the new concept and a process of vibration analysis for an automobile is undertaken.

This investigation has been successful in developing a design for a new height adjustable suspension however the scope of this study did not allow for adequate testing to prove its viability and allow the design to be progressed into manufacture. Despite this lack of testing, the system in theory will provide for a more comfortable ride and a much more versatile vehicle. This new system of suspension will be able to perform in a wide variety of applications and environments without length setup or adjustment.

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Joshua Graeme Walton

Student Number: 0050026275

Signature

Date

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NOMENCLATURE AND ACRONYMS

The following abbreviations have been used throughout the test and bibliography:

FBD	Free body diagram
DOF	Degree of Freedom
m_x	Mass in kilograms
I_x	Second moment of inertia in kilograms per metre squared
k_x	Spring constant in newtons per metre
c_x	Damping ratio in newtons per metre squared
ϕ	Angle in degrees
z_x	Vertical height in metres

CHAPTER 1

INTRODUCTION

1.1 Outline of the study

This project is focused on modelling and analysing a new variable height suspension system for a passenger car. The project intends to undertake detailed static and dynamic analysis of a new variable height suspension system, model the suspension system and propose a complete design for a particular automobile. This is entirely theoretical and analysis will be computer based due to the time constraints placed on this project.

1.2 Introduction

The cars of today tend to be only made for one purpose. It would be near impossible to find a car that not only caters for a smooth ride over rough roads but can also minimise roll when cornering tightly on smooth roads. Variable ride height suspensions are by no means a new idea, but don't offer the versatility being pursued in the outcome of this project. Limitations in current systems come about by not catering for a change in spring rate as the car's height is being altered. The result is a less than optimum solution in a smaller range of running conditions.

Australian's in general are required to travel massive distances every year over roads that vary greatly in quality. For anyone who enjoys the luxury of a low riding car and the associated handling benefits, they are forced to sacrifice comfort when driving on rough roads. Innovation is required to produce a system that will be able to function in all conditions and for all applications.

This project is based entirely on theoretical concepts and mathematical analysis. The solution obtained cannot be proven by physical construction and testing due to the time constraints applied to this project. These time constraints may mean this project cannot successfully produce a system that satisfies the aims of this project. However more important than the solution itself is the process involved in deciding whether or not a successful solution has be found. This project will define the process needed to analyse a suspension system and can therefore be used to continue this study or be applied to other studies involving four point suspension on motor vehicles.

Because of the personalised nature of each individual vehicle suspension, this project will only investigate the new system for one model of vehicle. For this project that vehicle will be a 1996 Series I, VS Commodore due to the availability of this vehicle for gaining data to be analysed. Despite the fact only one application will be studied, it will be easy for the same principles and system identified in this project to be applied to any other vehicle. Once the system has been created and testing procedures established, the personalisation of the system for each individual vehicle will be a natural step forward.

At the outset, time will be spent investigating current technology to prevent wasted time spent on covering already developed areas of this research and development. Then with a well rounded view of current innovation, the design and testing of a system to satisfy the aims of this project can be completed with a much stronger sense of direction and in a shorter space of time.

Developing a complete solution requires a wide variety of vibration analyses on the system to be in question. The behaviour and response of the car as it is acted on by the road affects the drivers comfort and ability to navigate different types of roads at varying speeds. This project will seek to simulate the extremes to which suspension in Australia might be made subject to, and furthermore the affects passed onto the driver in each situation. Results for the new suspension method will be compared and contrasted to a set of control results established for a standard suspension system in use today.

It is important that the analysis of vehicle suspension and vibration be sound in its consideration of all factors and variables to ensure sufficient accuracy for the results to be projected directly into the real world. Following the conclusions of this project there may be a desire to continue the development of this idea and produce physical prototypes. Therefore despite this projects immediate theoretical basis, it will be possible for the solution to be taken and put directly into service.

1.3 The Problem

Cars have undergone rapid evolution over the 20th century and during the entire process suspension has been critical to the cars ride and handling. Suspension governs the way in which a car holds on to the road, takes corners and how much road variation is transmitted to the occupants.

Take almost any car that is manufactured today and you will find that it is only suitable for driving the way in which it was intended by the manufacturer. For instance a base model family car like a Commodore or a Camry is made to ensure a comfortable ride at relatively low speeds. If a car like this were to be driven fast in cornering manoeuvres or rapidly accelerated/decelerated, the driver would notice the car become unstable and begin to pitch and roll. This reaction is a direct result of the suspensions limited functioning capacity purposely intended by the designers. Next take a sports car with low profile tyres and stiff suspension. A car like this is well suited to smooth roads and fast driving but take it outside the city or on an unsealed road and the driver comfort and the cars life span declines rapidly. The general rule in making cars is that they are made for one purpose only and using it beyond that purpose is detrimental to the safety, comfort and life span of the vehicle. My aim is to produce a new way of suspending a car that will allow it to multitask. I want to design a new strut that adjusts between a sports-like ride and a soft cushioning ride. By fitting this strut I want the driver to have the ability to modify the cars height and handling while driving to accommodate such a radical transition as going from racing on smooth track to driving on unsealed roads.

The new struts primary purpose will be to meet the performance demands of users across the board by increasing functionality. Its development will not be constrained by an increase in cost or greater use of power however these factors will rank as highly important throughout the design process. This project is conducted with the recognition that this new strut is by no means an essential improvement but rather a luxury and an after market upgrade. Its viability will depend on the degree to which it succeeds to meet its intended design criteria and at what cost this comes.

1.4 Research Objectives

This project aims at creating a versatile method of suspension capable of adjusting its resting height and spring constant. This suspension must permit the car on which it is installed to enjoy improved, sustainable and successful use over a greater variety of road conditions. The suspension must provide comfort and safety for the driver by dissipating vibrations from the worst of road conditions. In meeting these aims, the specific objectives below are to be pursued:

- Review literature relating to the development and current advancements in suspension technology, particularly in the area of adjustable suspension.
- Develop a method of achieving car height and spring rate adjustment.
- Produce a design concept for the new suspension system.

- Determine and implement a way of modelling the new system of suspension for analysis purposes

- Develop assessment criteria that can be used to determine the relative success or failure of the suspension system in meeting the project aims.

- Run a full analysis using the chosen method of model taking into account different driving variables such as road condition, car load, and car height.

- Optimise the design concept using the gained results and finalise the design by producing detailed drawings.

1.5 Conclusions: Chapter 1

This dissertation aims to develop a new versatile method of adjustable suspension for an automobile. This research will hopefully identify ways of suspending a car that had not yet been considered. Modelling methods used within this project will prove useful for any study with a focus on vibration analysis of automobiles. The final design produced by this dissertation will be ready for prototyping and physical testing with a view to moving the design closer to a production.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

The movie 'Hitch'(2005) contains a line spoken by Alex Hitchins:

"You can't know where you're going until you know where you've been".

The context in which this statement was said bears no relevance to this project, but the statement itself does. Without a proper review into the literature associated with this problem there is no way of knowing whether the solution developed is simply retracing an already defined path in the history of suspension. This chapter will assist in understanding the background to the problem identified in this dissertation.

2.2 Development of Suspension

Suspension has and always will be one of the most defining aspects of a cars handling, comfort, and safety. The development and progression of the motor car over the last century draws strong parallels to the development of the suspension on which it rides. With cars becoming faster, and a growing expectation for increased comfort and safety, suspension has had to evolve and cater for all these needs.

The Sumerians were the earliest recorded civilisation and yet all those thousands of years ago vehicles were being used and the inherent need for suspension was born. From the ox cart used by the Sumerians came more advanced carriages and chariots. While none of these vehicles at this stage had incorporated suspension they did contain a certain degree of well thought out engineering. The chariots for instance were made as light as possible for increased performance. The wheels were so light that they were removed when the chariot was not in use so that they could not squash into an oval shape. The desire for greater performance and innovation was evident but it would not be until the 8th century that suspension would be born.

The first attempt at lessening the jolts transmitted from the primitive roads of medieval time came in the form of an ox drawn carriage suspended with steel chain and rested upon straw-covered baskets. The result was a terribly unstable ride but the general theory carried on into subsequent designs. By the 15th

century the chains were replaced with leather straps on coaches and then in the 17th century the metal spring came into existence.

The first steel spring was a flat plate and was used on carriages by the French. This led to the leaf spring being created. The leaf spring utilised multiple layers of springs which actually worked to dampen some of the jolts as a result of the friction between each layer. With the advent of this leaf spring the ‘eight springer’ was created in 1804 by a man named Obadiah Elliot. The most revolutionary aspect of this new vehicle was that each of the four wheels was fixed to the carriage via two leaf springs opposed to each other. In the past horse drawn vehicles required a heavy under-carriage below the suspended carriage, where as now with this new design the carriage could be fixed directly to the axles via springs. This meant lighter vehicles capable of greater speed and improved safety.

The coil spring was first patented in 1763 by R. Twedell but didn’t make its entrance into the automobile industry until Daimler used them on their twin cylinder in the late 1800’s. Other means of suspending a car included torsion bars and even early attempts at air suspension. It wasn’t until 1934 that a great deal of car manufacturers started using the coil spring extensively.

Until the late 1800’s springs were mounted on cars with out any deliberate attempt to prevent the continual oscillation that comes from compressing and releasing a spring. A man by the name of J.M.M. Truffault was the first to incorporate a friction device on his motorbike in 1898. This device was later modified and fitted to an Oldsmobile becoming the first automobile shock absorber. It worked simply with two levers hinged together with a rubber pad at the interface and a bolt that could be tightened or loosened. This type of shock absorber could be defined as a friction type absorber. Following this design came the hydraulic shock and the air shock which still have their place in the engineering of today’s cars.

Air suspension was first attempted in 1909 but the system leaked proving it useless. Firestone released their air suspension in 1933 and became the first to do so successfully. The four springs were replaced with air filled bellows supplied by a small compressor. Unfortunately this system proved rather expensive in comparison to conventional springs and even today the use of air springs requires more setup and more cost.

Suspension has always faced a dilemma in the fact that it seeks to create the softest ride and yet in order to damp the oscillations, the shock absorbers actually cancel out the softening action of the springs. As a result springs and the shock absorbers actually work against each other. In effect there is no perfect solution to suspension just a great deal of middle ground where an optimum setting can be achieved between hard and soft suspension.

As long as roads have bumps and ripples, a car will always require a form of suspension and even today development continues in a never ceasing battle to be better, to innovate and to improve.

2.3 Height Adjustable Suspension

Motor vehicles cater for multitudes of people all demanding performance in a vast array of situations. It is not surprising then that standard production line cars sometimes don't provide the versatility or performance required by an owner. One way in which these aspects of a vehicle can be improved is by including or installing a method of adjusting the ride height. By adjusting the ride height of a vehicle its aerodynamics, economy, handling, vibration response and carrying capacity can all be affected for the better. While very few cars are produced standard with height adjustable suspension, it can be a useful device that some choose to fit after market.

There are three basic forms of height adjustable suspension. The first utilises a standard coil spring although the spring itself may sometimes be far from standard. Systems using coil springs generally require manual adjustment to affect a change in height. Other means of height adjustment utilise either air or a combination of hydraulic fluid and air known as a hydropneumatic suspension.

Coil springs can be used to vary a vehicles ride height by moving the lower stop of the spring up and down. The spring remains unchanged in the process and the height is modified by effectively mounting the car at a different point up along the spring. The images shown in Figure 2.3.1 are coil height adjustable suspensions commercially marketed by Tien, Koni and Suzuki. Adjusting this type of suspension is done manually and requires time and effort in partially disassembling the vehicle in order to access the adjusting screw. The springs used in many of these forms of suspension contain variations in pitch and coil diameter to attempt to account for the handling requirements of the different height settings. However the spring rate cannot be changed as the car is lowered and hence extra stiffening of the suspension is sourced from adjustable dampers. This solution to height adjustment is most suited to users who intend to race their vehicle. Others who are requiring greater versatility from their vehicle may find this system requires too much time and effort to achieve that versatility. Above all other positives of this system stands the cost. While this may not benefit all users requiring adjustable suspension it is a simple and cost effective method.

Figure 2.3.1 – Coil spring height adjustable suspension.

Air suspension as mentioned in the previous section has been used on vehicles at varying frequencies and with mixed success. Air systems can also be used as a means of achieving height adjustment in the vehicle. By using a series of air springs similar to the ones shown in Figure 2.3.2 and combining it with a compressor, controller, valving and a dryer, height variation can be automated substantially. With this method the height can be adjusted from inside the vehicle at any time by increasing and decreasing the pressure. It has an added advantage in that the controller can also perform the task of levelling the car if height sensors are placed in each spring. Levelling is valuable when heavy loads must be carried and large deflections of ordinary springs are not wanted. Air suspension has long been the trusted springing method for heavy vehicles and has very tentatively begin edging its way into smaller vehicles. Existing air springs do have the drawback that they cannot simply replace a coil spring in all their behavioural properties. While the properties of air suit heavy vehicle transport they are not as easily put to work underneath a light automobile. There lack of popularity within smaller vehicles is also due to their elevated cost when compared to a coil spring. These systems are a more costly solution than the coil springs however their versatility and ease of adjustment makes their cost worthwhile.

Figure 2.3.2 – Air springs.

The final option for height adjustment is with a hydropneumatic system which combines air/gas and hydraulic fluid, separated by a diaphragm, to spring the vehicle. This system of suspension is typified by the use of spherical reservoirs positioned at the top of each strut on the vehicle. A section view of these reservoirs is shown in Figure 2.3.3 and identifies the two halves of the system. The top half of the system is a gas at pressure and this is separated from a hydraulic fluid by a centrally located diaphragm. Instead of a standard spring and damper making up the strut, a simple piston or syringe is positioned where the normal strut would be positioned. As the suspension is compressed the piston forces hydraulic fluid into the spherical reservoir through a hole which serves as the damper of the system. The size of this hole can be varied to achieve the required damping rate. As the hydraulic fluid is forced into the sphere it compresses the gas in the top section which then acts as a spring and absorbs the force applied by the fluid. Once the suspension is no longer being compressed, the added pressure in the gas forces the fluid to return into the piston.

While the initial theory of this system seems simple enough it cannot run effectively without interconnections between all four corners of the car. The hydraulic fluid is pressurized via a hydraulic pump driven directly from the engine. This pump can also be used in braking and power steering and hence is not just another device required to run a luxury system. From this pump lines are run to apply pressure in the hydraulic fluid and adjust the height of the vehicle. Also numerous connections between opposing struts forward and aft are needed to assist in levelling the car and also help create a stabilising effect similar to that of a stabilising bar. The system must be made quite complex to ensure that the hydraulic fluid is allowed to flow between struts but only when needed. For instance if the front left wheel hits a bump the fluid should be able to flow across to the right front wheel and apply a lifting force to that wheel in order to level the car and detract from the abruptness of the impact. However if this is an open circuit and the fluid is simply allowed to pass at will between the two spheres then during continuous slow cornering the car would gradually fall to the outside of the turn as hydraulic fluid is forced from the outside strut across to the inside. Therefore numerous valves and additional spheres are used within the circuit to make the hydraulic fluid perform similar tasks to mechanical components in a conventional suspension system.

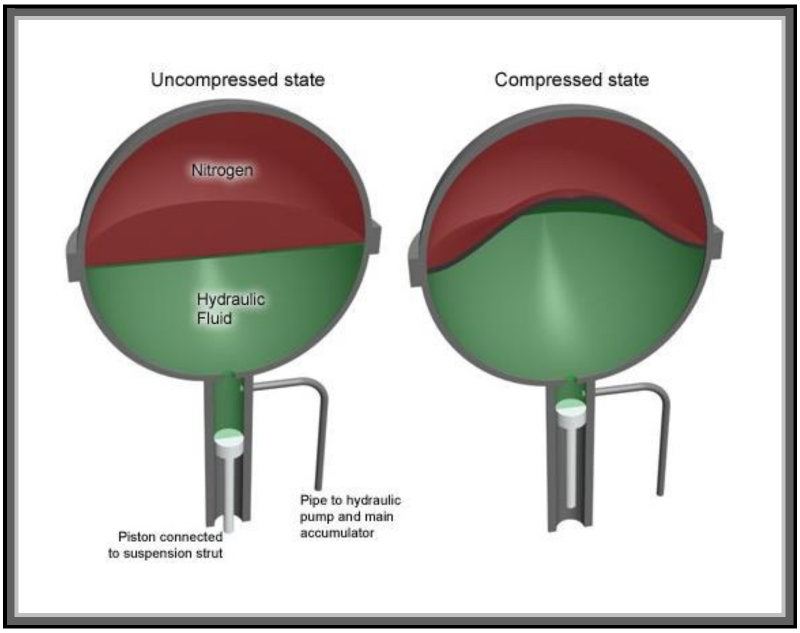

Figure 2.3.3 – Hydropneumatic suspension.

The system of hydropneumatic suspension is one of substantial complexity but does provide a very comfortable ride. One of the drawbacks of this design is that in time there may be issues with hydraulic leaks and possibly pump failures that could be messy and costly to fix.

Height adjustable suspension is performed successfully in a number of ways. It seems all these methods have their advantages and disadvantages and none of them can claim to be the most desirable solution. It seems in all these systems that complexity or limited functionality restricts their versatility and improvement in this field is definitely capable.

2.4 Motor Vehicle Geometry

By definition the suspension of a car includes not only the strut but also the arm(s) by which the strut and wheel are attached to the car. Early types of suspension suffered road holding problems whenever the cars height changed over bumps or around corners. This was caused by the suspension arm altering the wheels camber and hence reducing the contact area of the tyre. Modern day suspensions use arm configurations that maintain the camber as the cars height above the road changes. This means that changing the height of the car with a new strut will not adversely affect the contact area of the tyre because the height variation intended in this investigation will be within the manufacturers own limits of normal suspension movement. Many variations of suspension arms exist however because these hold no relevance to the design of this new spring, they need not be explored further.

2.5 Dynamics of a Car

This project aims at discovering an effective alternate means of suspending a car. In order to determine the effectiveness of any new suspension system there must first be a method available to test and compare the suspension to real world and existing suspensions. It would be impossible to evaluate this new method for all models of cars and still remain within the scope of this project. A more realistic approach is to select one type of car, then model the suspension to only suit that motor vehicle. Then by running tests on both the existing suspension and the new suspension without changing the geometry or basic constraints of the car, a result for effectiveness can be achieved.

The vehicle chosen to be used in this project as seen in Figure 2.5.1 is a 1995 Holden Commodore VS Series 1 Vacationer. This is the vehicle I own and would have used for physical testing had time within this project permitted. By choosing this vehicle it is easy for me to analyse, understand and model its behaviour based on my own observations.

Figure 2.5.1 – Holden Commodore VS Series 1 Vacationer.

The suspension beneath this model of Holden Commodore is made up of:

- Front
  - … McPherson strut
  - … Wet sleeve shock absorber
  - … Direct acting stabiliser
  - … Progressive rate coil springs

- Rear
  - … Five Link Live Axle
  - … Trailing arms
  - … Panhard rod
  - … Progressive rate coil springs
  - … Stabiliser bar

The setup on this commodore is the most common form of suspension found on modern cars today. The Macpherson strut shown in Figure 2.5.2 was invented by an engineer at Fiat but was named after Earl S. Macpherson who developed the strut. This suspension can be used on both front and back of vehicles but is most commonly found on the front. It consists of a wheel hub, a link locating the bottom of the hub, and a main upright housing the coil spring and shock absorber. The Macpherson strut is a simple and cheap form of suspension which has ensured its popularity. Another positive is the small amount of space it consumes leaving more room in the engine bay. It has a few draw backs because of its geometry and subsequent tendency to alter the wheels camber when the height is changed. Also because the main upright is fixed to the body of the car it transmits much more vibration and road noise to the body of the car than other forms of suspension.

Figure 2.5.2 – McPherson strut suspension.

The rear axle of this Commodore is solid and fixed at five points. A similar configuration for is shown in Figure 2.5.3. This axle makes up one of two main forms of rear suspension with independent rear suspension being the other popular choice. Four of the fixing points along the solid axle are connected to trailing arms that sweep back off the body of the car to locate the axle forward and back and allow it to swing up and down. The fifth fixing point connects to a Panhard rod which stops the axle from moving side to side. An alternative to a Panhard rod is a watt's linkage.

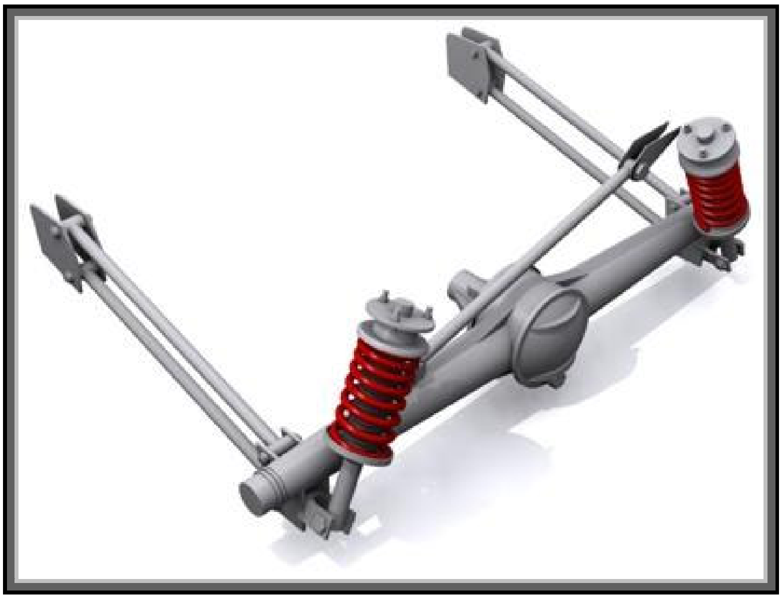

Figure 2.5.3 – Solid five point rear axle with Panhard rod.

The progressive rate coil springs are almost impossible to model accurately without an equation being supplied by the manufacturer. An attempt was made to obtain the spring rates from Holden but they could not be procured. Fortunately data for a Holden Commodore was found within the thesis work of Lars Svedung which he himself gained directly from the Holden Motor Company. The spring rate is said to vary from 19-23 N/mm and can be approximated with a constant value of 21.6 N/mm. A graph shown in Figure 2.5.4 shows data supplied by the company for the spring rate.

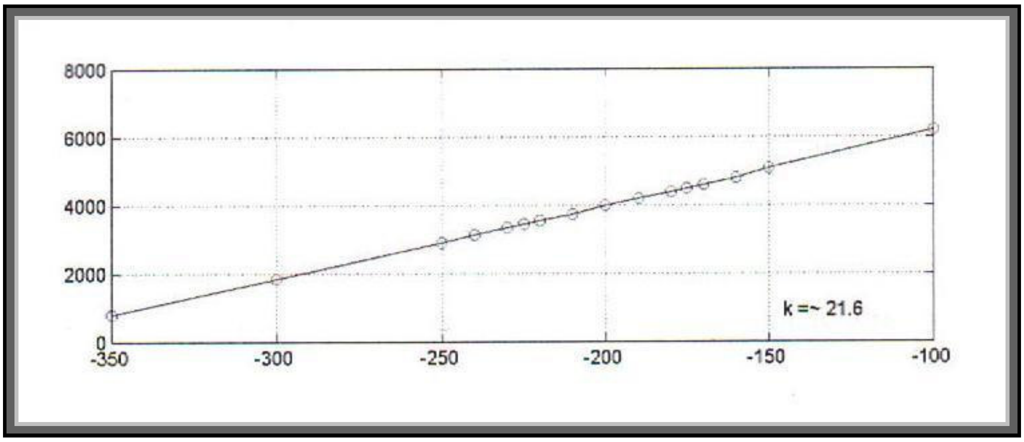

Figure 2.5.4 – Company supplied spring data.

Along with data for the variable spring rate came values for the damping coefficients of Holden's wet sleeve shock absorber. A plot in Figure 2.5.5 shows experimental values of the damping force. As is stated in the figure, the damping coefficient can be approximated with a two standard values. The first value is for compression and is equal to 0.2 N/(m/s) while the second value for extension is 1.4 N/(m/s).

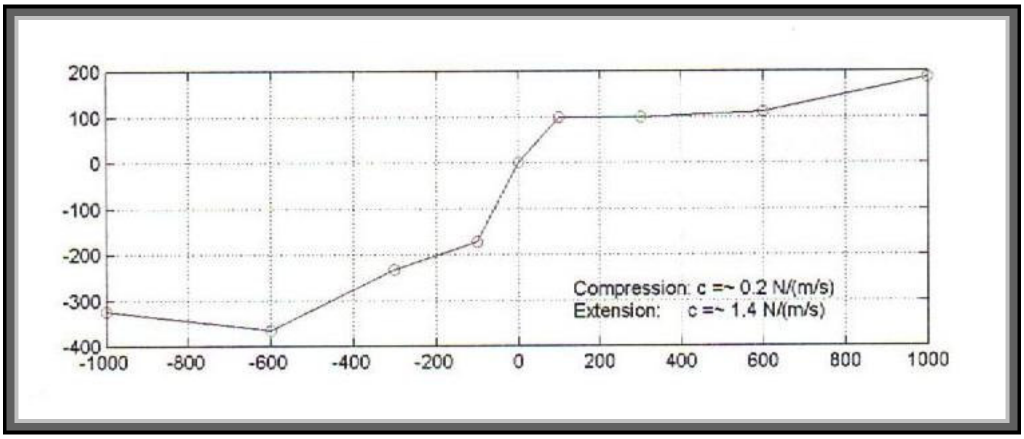

Figure 2.5.5 – Company supplied damper data.

Both front and rear suspensions on the Commodore are linked laterally across the car via a stabilising or anti-roll bar. Stabilisers help the car remain level while cornering or whilst the car is subject to bumps on only one side of the vehicle. A typical stabilising bar can be seen in Figure 2.5.6 linking both sides of a double wish bone suspension. A stabilising bar works by applying opposing force to the coil spring on the opposite side of the car to which the suspension is being forced upwards. The opposing force prevents the unloaded side of the car from rising up and further de-levelling the vehicle.

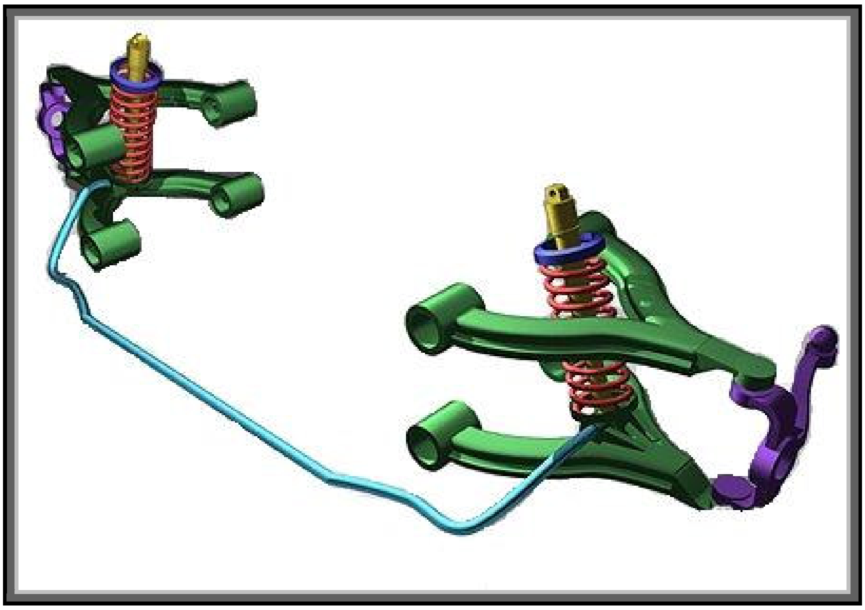

Figure 2.5.6 – Stabilising bar.

In the mechanics of the suspension of this Holden Commodore all these components work together and are critical to the performance and handling of the car. Understanding the physical makeup of this suspension can provide the basis for a sound theoretical investigation and analysis of this suspension.

2.6 Conclusions: Chapter 2

The field of automotive engineering plays such a major role in the fast paced world of today. Suspension alone can be seen as a field that has and continues to be developed and changed to produce new and exciting solutions to our every increasing list of demands. One of these demands being our need for greater versatility in the area of suspension has been met through adjustable coils, air springs and hydropneumatic solutions. These systems then combined with linkages and fixtures form a dynamic situation with countless possible combinations and behaviours. While no ‘right’ way of suspending a car may ever be established it is encouraging to see engineering’s continued push to improve and refine.

CHAPTER 3

CONCEPT DEVELOPMENT AND DESIGN

3.1 Introduction

Every product, invention or innovation began with a single idea. At some point in time, in some place, in one persons head, an idea came to life and sparked a process of development. No product we use in this modern age ever came into being without time and effort spent in turning an idea into a viable reality. Even the biggest or most complex projects started with an idea and were slowly unfolded to build an accurate picture of how the idea might be brought into existence. This chapter will follow the process of development for the variable ride height suspension from its initial idea through until the aims and objectives have been achieved with a viable solution.

3.2 The First Idea

Many new ideas are born from a perceived need and in this case the same is true. Existing suspension systems successfully provide the ability to adjust car ride height but do not in the same act succeed in varying the spring rate. As well as this the height adjustment on existing units is not a simple operation but instead involves time and effort. The initiation of the idea for this investigation began with the desire to create an adjustable spring that would change its spring constant by itself as its open height was varied. (See Figure 3.2.1) By adjusting the spring rate with the open height, the closed reaction force of the spring can be kept the same and hence prevent a car from bottoming out.

The starting point for innovation was considering whether existing designs could be furthered to include better automation in their adjustment capabilities. For instance in systems where an adjustable mechanical screw is used to move the lower spring mount further down the strut and effectively lower the vehicle. The adjusting screw could be made to operate via electrical motor or servo activated remotely from the cabin. This function combined with the existing remote shock adjustment would allow the entire system to be adjusted easily. However the size of motor or servo needed to move the spring base would be much too large to be deemed viable for cost, space and power consumption reasons. Other methods of adjusting a steel spring were not considered with any weight due to the complexity of the design that would be needed.

The first concept to be given significant thought was the idea of using an air cylinder. The attractive principle of an air cylinder is its ability to change initial volume. Then no matter what size of initial volume left in the cylinder, the slide would never be capable of bottoming out due to the inverse relationship between

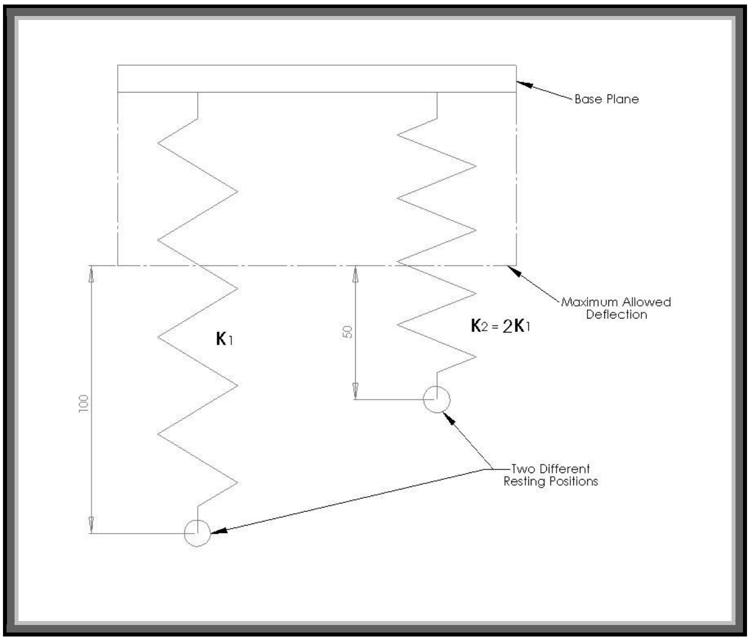

Figure 3.2.1 – Illustrating the varying spring constant.

pressure and area. Take any cylinder of any diameter with a slide. The cavity is sealed and the pressure in the cylinder is atmospheric. So we know that as this cylinder is allowed to compress, the amount of the air inside does not change. As the slide compresses the air, the pressure doubles every time the volume halves. Therefore it is virtually impossible for the slide to reach the bottom of the cylinder because the pressure would be infinite.

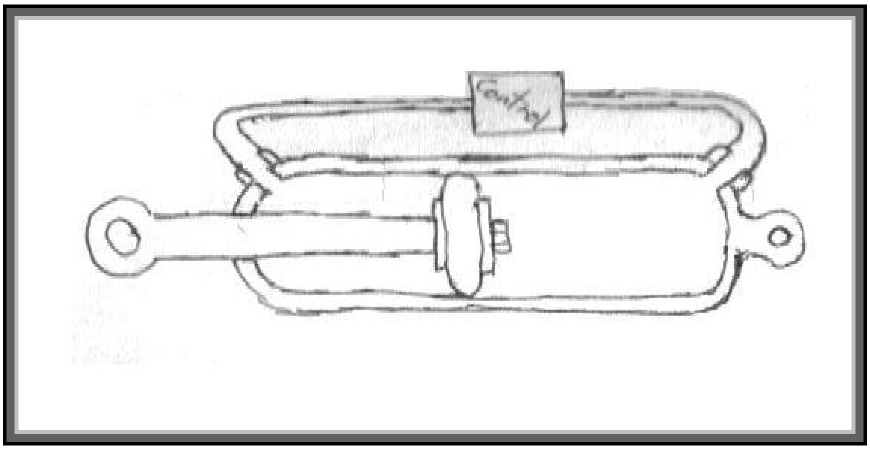

Figure 3.2.2 – The very first concept.

Again take the same cylinder but position the slide at the middle of the cylinder and at this point make the pressure atmospheric. From this point seal the amount of air in the cylinder and begin compressing the volume. As the slide moves down the pressure increases similarly to the first situation except the pressures are scaled over a shorter distance. In this situation the slide will again never reach the bottom of the cylinder because the pressure would have to be infinite. This principle is illustrated in Figure 3.2.2 as a closed circuit that could simply circulate air from the top of the cylinder to the bottom and hence change the position at which the slide was comfortable to be at rest.

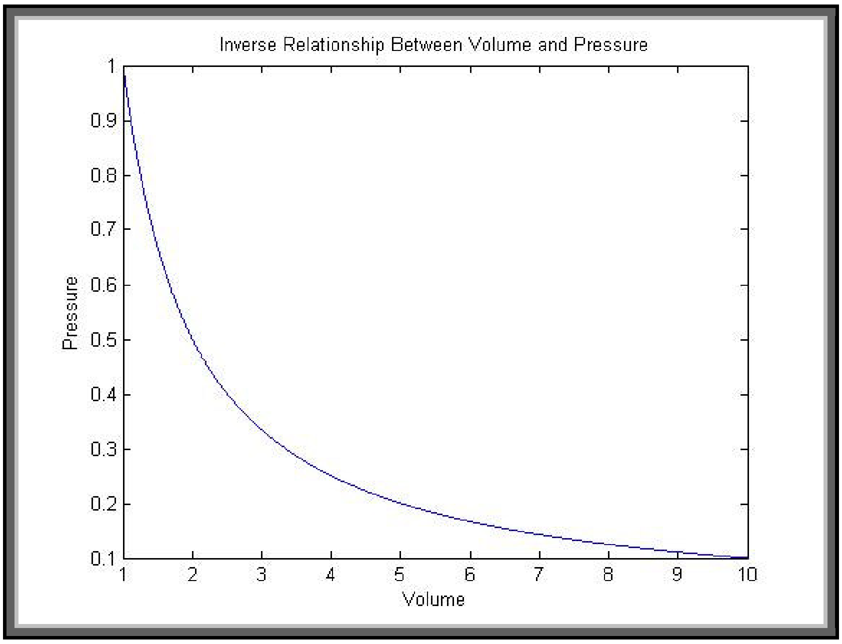

Figure 3.2.3 –The inverse relationship between volume and pressure.

This idea seemed attractive initially due to the fact that by simply altering the mass of air in the cylinder the spring height could change and the slide would never suffer from bottoming out. Unfortunately the problem with this idea came about by the same gas property that made the proposition attractive in the first place. Given an inverse increase in pressure as the volume decreased as shown in Figure 3.2.3, the pressure rose increasingly faster and did not provide an adequate approximation to a spring. A spring generally has a fairly constant ratio between deflection and opposing force which makes the absorbsion of irregularities seem smooth. When this constant ratio is replaced by an inverse one, the irregularities are absorbed far too quickly and larger forces are passed through into the mounting of the spring. In the case of an automobile the spring mounting becomes the chassis and the seat of the occupants which results in a rough ride. Due to this factor the idea of a cylinder could not be further explored.

3.3 The Right Theory

Achieving great adjustment in this suspension had to come about by the use of air. Air provides an easy means of adjustment by simply modifying pressure or volume through hoses, valves and pumps. Despite the inability of an air cylinder to perform the required job, the use of air began to be explored further with an aim to convert the inverse relationship of the air and pressure into a linear relationship between some other two variables.

A simple look at the gas equation (3.1) proves that unless this device can alter the density or temperature rapidly enough to account for the changes in volume then no linear relationship can be found. The gas equation states that the pressure in a vessel will be equal to the product of the density, gas constant and temperature all divided by volume of the vessel.

$$P = \frac{\rho \cdot R \cdot T}{V} \tag{3.1}$$

Instead of manipulating the gas a new method of adjusting the force of the spring had to be found. By analysing the basic formula for pressure (3.2) an option was discovered. This equation states the pressure is equal to force divided by area.

$$P = \frac{F}{A} \tag{3.2}$$

By transferring the area across to the other side of the equation it becomes an equation for force and states that force is equal to the pressure multiplied by the area.

$$F = P \cdot A \tag{3.3}$$

When comparing this with the general spring force equation (3.4) which states that the force is equal to the spring constant multiplied by the deflection of the spring a similarity is observed.

$$F = k \cdot x \tag{3.4}$$

By choosing to hold the pressure constant as (**k**) and only vary the area linearly as a function of height (**z**), the equation will be transformed into a spring equation with a linear spring constant.

$$F = P \cdot A\left(z\right) \tag{3.5}$$

The assumptions made to achieve this spring equation do work nicely but now need justifying rationally. For instance maintaining pressure may not be possible over the entire range of compaction involved with a linear spring. Also increasing the area linearly with respect to height variations may be a difficult feat to achieve. Less pressure variation can be achieved by not altering the volume of a chamber significantly during operation. This means however that the chamber volume needs to be larger in order to decrease the percentage volume made up by the cavity in the spring. Having realised zero pressure variation to be an impossible goal, the assumption was made that during complete compaction of the spring the volume change may be able to be kept to half. Therefore according to equation (3.5) the assumed pressure increase over the full length of operation would be twofold. If the linear area increase could be achieved, the force to vertical displacement graph (Figure 3.3.1) would approximate a straight line and hence a linear spring constant. From this graph the inverse relationship can be seen to have its affect, however the graph remains flat enough to allow an approximate straight line of best fit. This graph proves the theory and made possible the continued development of this idea.

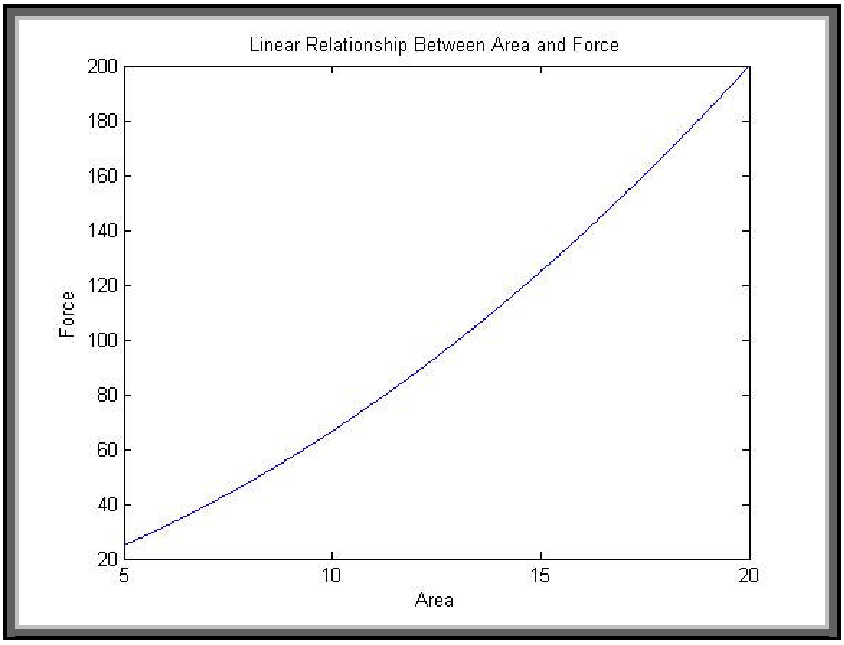

Figure 3.3.1 –The linear relationship approximation between area and force.

Traditionally the air bag suspensions in use today are of a cylindrical construction. To achieve an area increase during compaction of the spring a cylindrical bag was considered. With the area growing externally from an initial radius $\mathbf{r_1}$ by an amount $\mathbf{\Delta r}$ the relationship for linking area to radius is found to be:

$$\Delta A = \pi \cdot \left[\left(2 \cdot r_1 \cdot \Delta r \right) + \Delta r^2 \right] \tag{3.6}$$

This is not a linear relationship thanks to the squared terms, however by attempting an increase in area both internally and externally from a circle of radius $\mathbf{r_1}$ a new relationship can be found to be:

$$\Delta A = 4 \cdot \pi \cdot r_1 \cdot \Delta r \tag{3.7}$$

This relationship is linear and means if the system to be designed can increase area internally and externally it will have a good approximation to a standard spring constant.

3.4 Concept Development

While developing the theory for this investigation a parallel process of developing multiple concept ideas ran with it. These existed as mental pictures and sketches right from the inception of this investigation. Some of these concepts were given thought and created before the theory was fully understood, while others were developed after realising the true goal of this design. As a result most of the initial concepts hold no technical value in the build up of theory and understanding for this investigation. However whether technically relevant or irrelevant to the final solution, all these concepts help complete the picture of the mental process of this design.

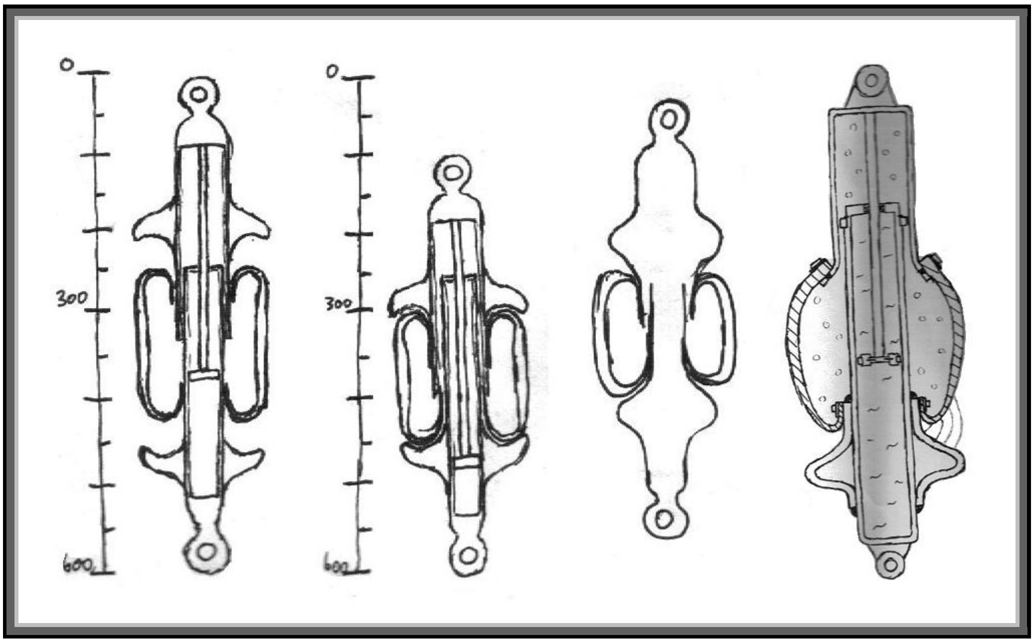

Figure 3.4.1 –Early concepts for an air strut.

Some of the earliest concepts as seen in Figure 3.4.1 were based around creating some way of ensuring the air bag could prevent the car from bottoming out during operation. These ideas allowed the normal operation of the of the air bag as it rolls itself up and down a vertical column but created sudden stops where the air bag be squashed against and not allowed to roll any further. The increased area acted on by the bag would prevent further vertical deflection of the strut. In operation this method was thought to do the job by allowing the operator to lower the vehicle to any height and still remain assured that the car could not reach the limit of its travel. In a situation where the car might be used for track racing the vehicle could be lowered right down to the stop and enjoy a vast stiffening of the suspension as a result. The chances of this system accommodating the huge variations in required ride height, handling and road conditions without causing discomfort to the passengers seemed very slim. The idea was abandoned as the thought process was developed further.

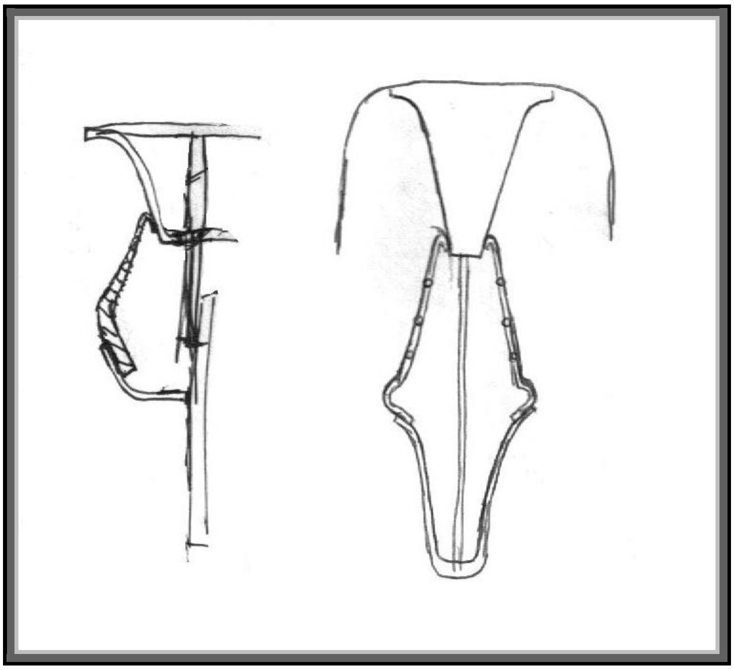

Figure 3.4.2 –Attempts at achieving an area growth via vertical displacement.

From this point on the concepts revolved around achieving an increase in area over the whole stroke of the strut. Simple concepts of this line of thought are illustrated in Figure 3.4.2 and show the use of conical shaped formers and rubber bladders that facilitate the area increase. At this point in the design the need to increase the area externally as well as internally was not realised. Problems with these concepts also existed and began with the formers and bag being unattached to the centre column over a long distance. This would mean the strut could be prone to collapse as the bag and former buckle away from the central shaft.

Another problem existed in the large bag section and its ability to hold its shape under high levels of pressure. In true operation the shape of the bag could not be expected to look as idealist as shown in these sketches. Finally these designs sought the need to seal the central shaft at some point along a sliding surface to prevent air leaks. Considering the number of times the seal would have to slide up and down the shaft in its lifetime it appeared completely impractical to assume the strut would sustain an air tight seal for even a short period of time.

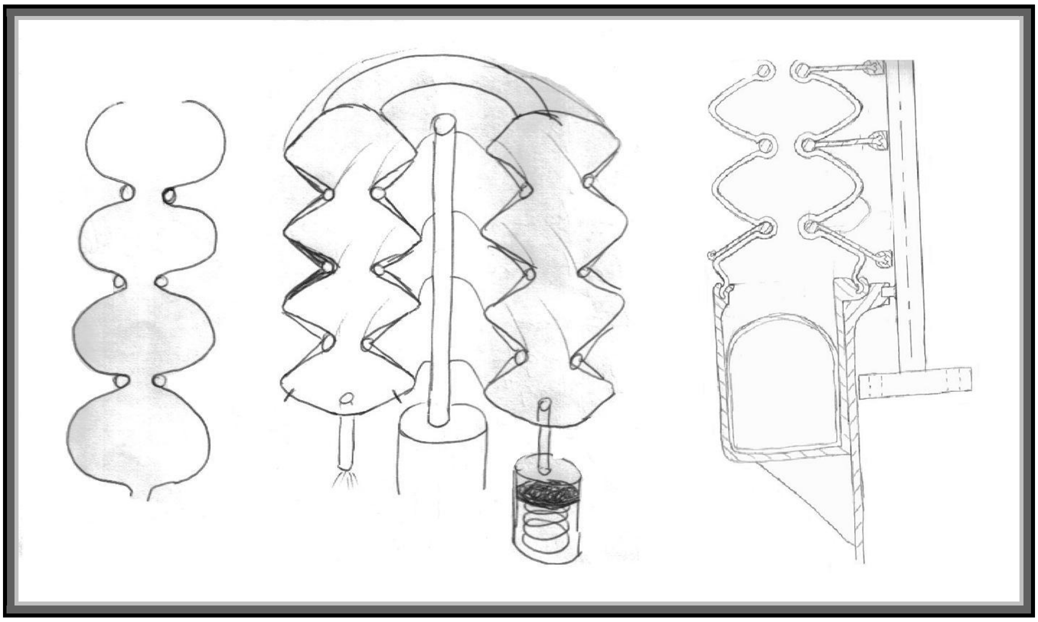

Figure 3.4.3 –Area growth internally and externally.

It was at this point in the design stage that the need to increase the area internally and externally was realised. Not only did this cater for the need to increase the area linearly in relation to the height but it was also considered that a completely air tight seal could be achieved with this new method. On top of these two advantages was another positive of being able to support the bag on the central shaft at the breaks in the sections over its entire length.

By using a reservoir comprised of an internal and external flexible layer as shown in Figure 3.4.3, any sliding seal could be removed and the central shaft could be run up the middle of the reservoir without having to be directed through into the air cavity. At this stage it was also deemed necessary to split the flexible reservoir into smaller sections defined by some type of steel former holding the reservoir in shape and allowing each individual section to squash together during compaction and hence gain an increase in area. Without the former the reservoir would simply balloon outward and inward and cease to function. Due to the space constrictions these smaller sections were needed to prevent the bag ballooning out and taking up too much space in the wheel well of the vehicle. These initial concepts for achieving internal and external area growth used a thick wire ring to simply restrict the bags expansion at certain positions. The rubber not in direct contact with the rings would then only be affected by pressure and could naturally balloon out but only to a limit imposed by the size of the sections. As the strut is compacted the individual sections would squash down on top of each other and create an increase in area varying with height. It was decided that in this design the area increase could not be controlled well enough. This was evident by considering the way in which the rubber would tend to balloon and how that would affect the sections squashing together. There were concerns that the area would not increase at a constant rate and create an unpredictable reaction force from the strut.

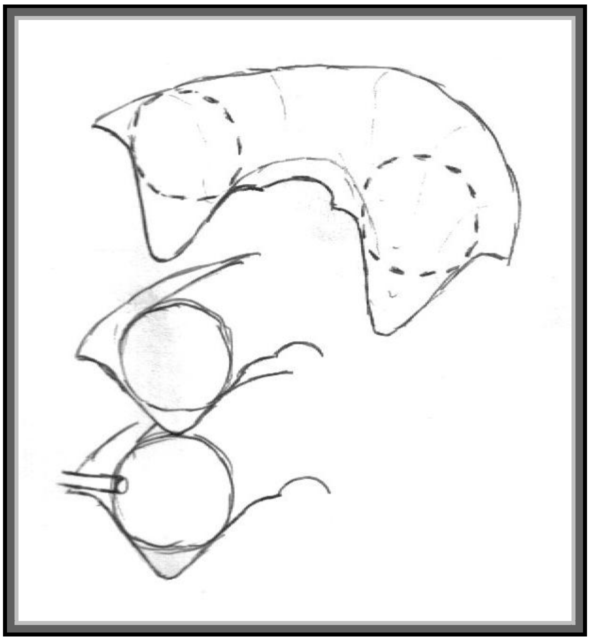

Figure 3.4.4 –Attempt at volume segregation.

Another idea explored on the way to a final solution was using individual air reservoirs in the form of donuts. These small reservoirs would be stacked on top of each other along with a sheet metal former which would separate each donut. As the strut was compressed the donuts would deform around the former and supply the area increase needed. The main disadvantage of this idea was the practicality of linking each of the individual reservoirs adequately enough to allow rapid air flow which is required in this design. Without the air in each section being capable of being transferred to an accumulator quickly, the pressure would spike before the chamber could be drained of excess air and the jolts of the road would not be absorbed successfully. In the Figure 3.4.4 this system is illustrated along with a bleed off line running into one of the donuts.

This would not be easy to protect during operation and may be a cause of concern for reliability.

The positive taken from the design in Figure 3.4.4 is the shape of the formers. Previous concepts displayed in Figure 3.4.3 have positive attributes associated with the single reservoir and having a path for rapid air flow down through the gap between the internal flexible layer and the external flexible layer. Both these concepts have the advantage of locating the bag on the central rod and allowing the bag to be designed to fit within any size of cavity. These positives were major contributors in advancing the design towards completion and to a point where all the objectives and aims could be met.

3.5 The Final Concept

The last step in the production of a solution to this problem came about through a combination of ideas developed in many of the initial concepts. The final concept is shown in Figure 3.5.1 and is made up of three basic components. The first of these is the rubber bag or reservoir which is shaped to have internal and external ribs. Helping keep these ribs defined under pressure and also provide a surface for the bag to be squashed upon during compaction are internal and external steel rings. These rings are sloped to achieve approximately ninety degrees of angle between the exposed surface of the ring and the reservoir and provide a mirror to the reservoir with respect to the horizontal plane. This angle is thought to help the reservoir be laid out upon the rings uniformly during compaction due to the way the bags shape will follow the rings when rolled down them. By keeping all angles with in the structure identical the spring can be compressed to a very small percentage of its open height.

Positive attributes of this design include the central cavity to allow for the shaft of a shock absorber to be inserted. This shaft will also permit the central rings to be located and not allowed to wander or buckle out from under the load. The segmented design means this air spring can be made to be any length by simply stacking a greater number of sections on top of one another. The design utilises no sliding seals and air flow through the whole spring is unimpeded. The angle of the rings and shape of the bag have sought to do away with having to take into consideration the bulging of the reservoir. Instead using this design the pressure inside the bag and its resulting shape is manipulated to create a controlled reaction force dependent only on the height to which the spring has been compressed. Although the outside rings are not located like the central ones, the ability they have to move around is very little and the stability of the spring is ensured by the limited travel of the central discs.

Figure 3.5.1 – The final concept expanded and compressed.

The open bottom end of the reservoir is intended to be attached to a base which acts as a cavity for the air to rush to when the spring is compressed. By providing a cavity for the storage of air, the pressure of the spring can be kept relatively constant throughout the process. From the beginning this cavity was intended to be positioned directly below the spring and hence allow there to be no difficulty with transporting air or keeping sufficient seals. However as Figure 3.5.2 shows, the idea was then changed to make use of a remote cavity or accumulator due to the space and height restrictions of the area of service. This remote cavity would be linked to the base of the spring via large diameter hosing to allow unrestricted flow back and forth between the two components. If a cavity was positioned directly below the spring it would subtract too much height and travel from the spring.

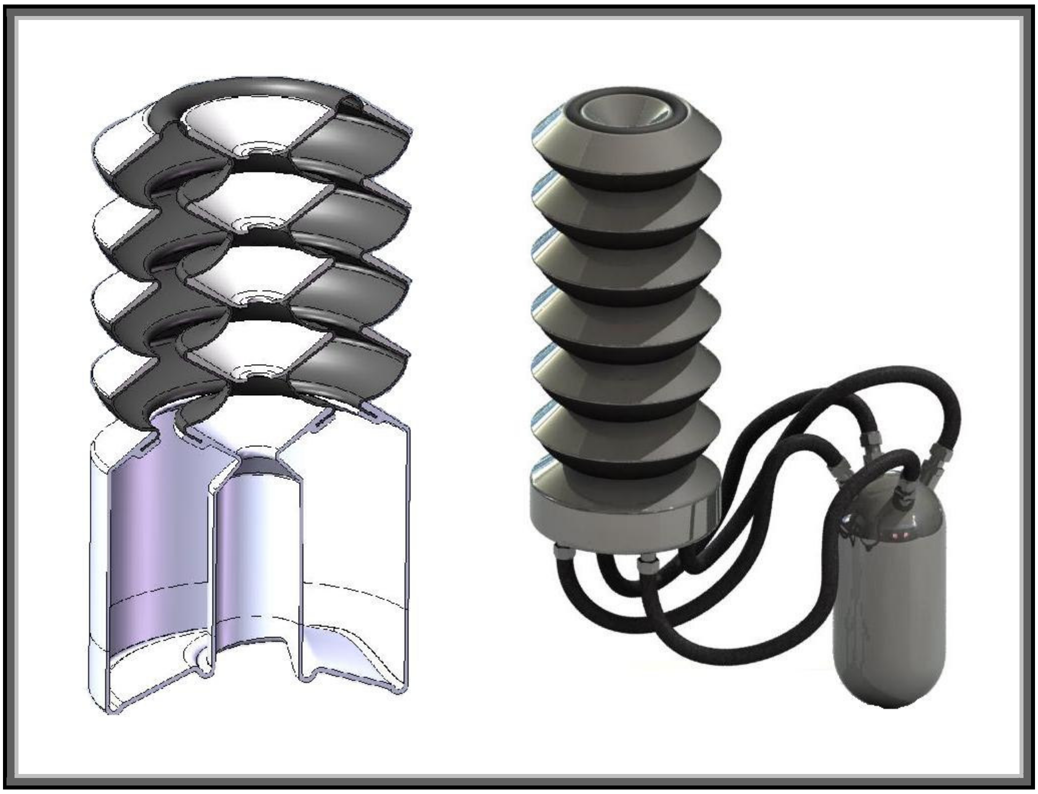

Figure 3.5.2 – Two cavity illustrations.

This concept still requires some thought to bring it to a stage where it can be manufactured but it is successful at this stage in the way it meets the objectives of this investigation. The sloping rings on the outside will prevent any foreign material from gathering on or in the spring. Also the shape of this final solution stands it in good stead for being able to replace existing steel springs of any shape or size. From this point on the process of concept development ends and the design can be carried forth into more processes that pave the way for this system to be put into production.

3.6 Adjusting the Concept for Manufacture

While the theory of how this spring will operate is logical, the design depicted in these concepts does not make sense from a manufacturing standpoint. This concept needs modifications that do not alter the way in which it operates but make it possible to manufacture and assemble.

The most crucial aspect of the design is ensuring the flexible rubber reservoir is able to be produced. During conceptualising this rubber bag was considered to be a single piece. While this is ideal for sealing purposes it is not possible. The internal steel rings cannot be inserted into their positions with the shape of the bag as it is now. To solve this problem there are two possible solutions. Firstly the internal rings could be integrated into the rubber bag from the time of manufacture. If this was at all possible it would certainly require a difficult and complex manufacturing process. The second solution is to split the flexible rubber bag into smaller sections as shown in Figure 3.6.1 and these will then become joined onto the rings during assembly. These smaller sections would be identical and could mean less setup for tooling but the sealing of the sections would be critical to making this design work. If the rubber were joined to the rings via a crimping action and vulcanising, the seal should stand the test of time. The picture shown in Figure 3.6.1 shows the crimping action on the rings as having been closed. The rings would be manufactured with the clamping jaw open so the rubber sections can be inserted and vulcanised to the inside of the jaw and then the jaw can be squashed closed over the rubber.

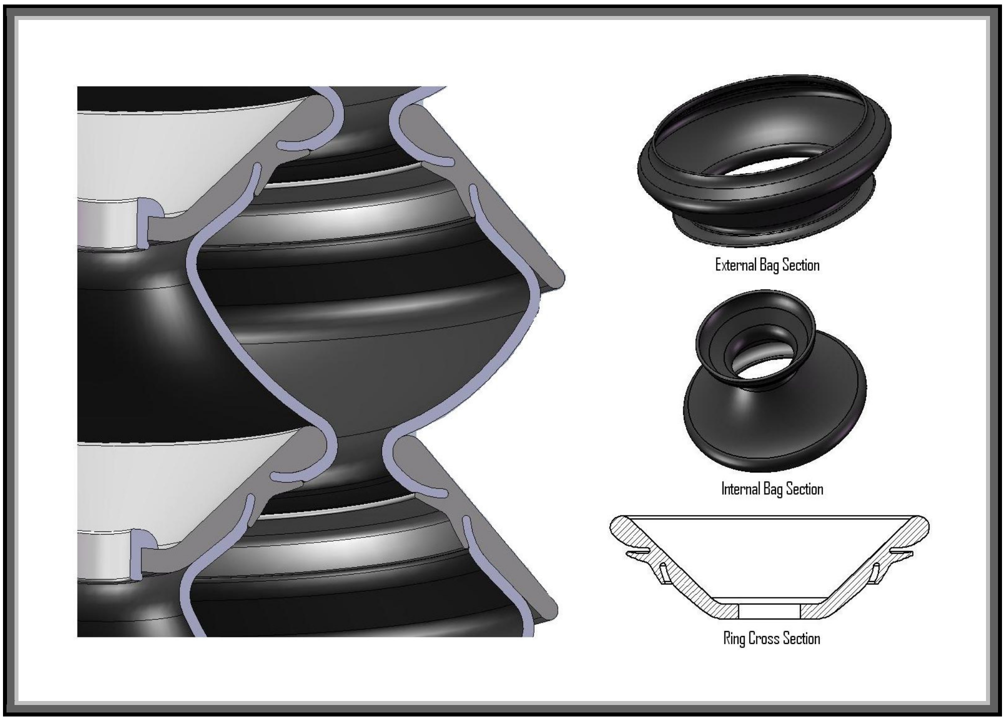

Figure 3.6.1 – Individual joined sections of flexible rubber.

All components of this final concept now perform their task and are entirely capable of being made successfully. The conceptualising has not taken this design very far into the real of manufacture but has produced a visual idea that is capable of successful manufacture. While this concept and its components are by no means simple and straight forward to manufacture, they do perform their job which is to satisfy the objectives of this investigation. The following chapter will continue the task of defining how this system will be built by undertaking the manufacturing process and materials selection.

CHAPTER 4

MANUFACTURE

4.1 Introduction

The design established in the previous chapter has been developed to accomplish the functional requirements of the problem. This chapter will carry on the design process and define the manufacturing processes involved in effectively, economically and simply making the theoretical design into a physical reality. Multiple options for manufacture will be explored before selecting the most suitable process for this specific situation. If necessary the design may undergo changes to allow for a more favourable manufacturing cycle.

4.2 Component Separation

In order to define manufacturing processes and materials for the design it must be split into its separate parts that can be manufactured individually and then assembled. The overall design for the spring has been split up into the following components.

- Flexible Bag Sections
- Shaping Rings
- Upper mount
- Lower mount

The sections that follow from here will examine each of these parts individually to determine the most appropriate material and manufacturing processes.

4.3 Flexible Bag Sections

As seen in the previous chapter, the bag is made up of numerous sections of flexible material shaped to form ribs that must be joined and sealed to the rigid rings creating one long compactable spring. These flexible pieces will be subject to continual distortion and tension as the bag is compacted. As well as this the bag will be exposed to forces of nature that will seek to minimise its life span. The finished product must be able to:

- Withstand the tension it experiences from the internal pressure.
- Flex but not stretch when in operation.

- Resist degrading when exposed to UV light.
- Be attached and sealed to other materials.
- Be cost effective.

The shapes of these flexible sections pose extra difficulty in manufacturing. While selecting a suitable material may seem a simple task, the way in which that material may be worked into the shape require complicates the matter. The solution to this manufacturing problem must be found with a combined look into materials and which respective forming operations are suitable for both the material and the finished product being sought.

A quick glance at any air suspension system from their inception up until this day will prove the fact that rubber is the common material used in flexible bag situations. There are a number of reasons for its prolific use. Rubber can be moulded, extruded and formed into many different shapes and then cured to maintain that initial shape. Rubber can be reinforced with fibres and has excellent ability to bond to various different fibre materials. Rubber can also be bonded or in effect glued to hard surfaces. The stiffness, strength and stability of the rubber may be manipulated to suit a specific service application by varying the constituents that make up the rubber. Rubbers ability to serve in a situation where sealing and flexibility are essential makes it the primary candidate for use in flexible bag suspensions. Selecting the right type of rubber for this particular realm of service will require a well rounded understanding of rubber in general.

4.3.1 Rubber Technology

Rubber was originally sourced from trees until the 1900's when innovation and the introduction of the motor vehicle saw the need and consequently the first supply of synthetic rubber. At present very little natural rubber is used due to the lack of supply and its unreliable properties. Instead synthetic rubber offers consistency and predictability as well as a variety of properties depending on where the rubber might be used.

Rubber is a polymer of isoprene and is made by firstly producing monomers and then putting them through a process of polymerisation to form rubber. The chemical definition and composition of rubber is known and can be found in the book 'Rubber Technology and Manufacture' (1971). Deeper insight into the chemical aspects of rubber is beyond the scope of this study and will not be explored further here. Bearing greater relevance to this study is the physical properties of rubber and having a broad understanding of rubber composition and its affect on properties.

Rubber by itself is does not provide a very useful or practical material. Very early on in the history of rubber there was not even an understanding of curing and vulcanising rubber. This meant rubber was always tacky and had very little strength and needed to be coated in powders to cover the sticky surface of the rubber. It was perhaps by accident and because of this method of coating that vulcanising was discovered to occur when the rubber was coated with sulphur and heated. Instead of the material being sticky it transformed into an elastic and tough material that could not be dissolved in solvents. From this discovery rubber became highly useful and continued to prove its value as the development of additives further improved its properties.

Rubber additives consist of accelerators, fillers, plasticisers, softeners and extenders. Accelerators assist in the rapid curing of rubber. Natural rubber tends to cure more quickly than synthetic rubber so accelerators are generally added to the synthetic rubber to bring its curing time in line with natural rubber. Fillers cover a wide range of additives and can be included in the rubber mix for many purposes such as:

- Reinforcing
- Colouring
- Adding Bulk
- Texturing
- Modifying rates of friction and wear
- Creating UV Stability
- Stiffening
- Increasing Durability
- Heat Resistance

The most common filler is Carbon Black which is basically fine carbon particles and helps create a structure within the rubber and consequently increase its strength. The difference in strength brought about by including carbon black is a tenfold increase. Different grades and types of carbon can be added to attain various properties. Table 4.1 gives a general overview and comparison of the various forms of rubber available.

PROPERTIES		NATURAL RUBBER	SBR	BUTYL	EPDM	NBR	SILICONE	NEOPRENE	HYPALON
Tensile Strength (PSI)	Pure Gum	Over 3000	Below 1000	Over 1500	Over 3000	Below 1000	Below 1500	Over 3000	Over 2510
	Black Loaded	Over 3000	Over 2000	Over 2000	Over 3000	Over 2000		Over 3000	Over 3000
Hardness Range	(Shore A)	30 - 90	40 - 90	40 - 75	40 - 90	40 - 96	40 - 85	40 - 95	40 - 95
Specific Gravity	(Base Material)	0.93	0.94	0.92	0.85	1.00		1.23	1.28
Adhesion to Metal		Excellent	Excellent	Good	Good	Excellent		Excellent	Excellent
Tear Resistance		Good	Fair	Good	Good	Fair	Poor	Good	Fair
Abrasion Resistance		Excellent	Good	Good	Good	Good	Poor	Excellent	Excellent
Compression Set		Good	Good	Fair	Good	Good	Fair	Fair	Fair
ReBound	Cold	Excellent	Good	Bad	Good	Good	Excellent	Very Good	Good
	Hot	Excellent	Good	Very Good	Good	Good	Excellent	Very Good	Good
Dielectric Strength		Excellent	Excellent	Excellent	Excellent	Poor	Good	Good	Excellent
Electrical Insulation		Good	Good	Good	Excellent	Poor	Excellent	Fair	Good
Permeability to Gases		Fair	Fair	Very Low	Poor	Fair	Fair	Low	Very Low
Acid Resistance		Fair	Fair	Excellent	Good	Good	Excellent	Good	Very Good
SOLVENT RESISTANCE	Aliphatics	Poor	Poor	Poor	Poor	Excellent	Poor	Good	Good
	Aromatics	Poor	Poor	Poor	Poor	Good	Poor	Fair	Fair
	Ketones	Good	Good	Good	Good	Poor	Fair	Poor	Poor
	Lacquer	Poor	Poor	Poor	Poor	Fair	Poor	Poor	Poor
RESISTANCE TO	Swell in Lubricating Oil	Poor	Poor	Poor	Poor	Very Good	Fair	Good	Good
	Oil & Gasoline	Poor	Poor	Poor	Poor	Excellent	Fair	Good	Good
	Animal & Vegetable Oils	Poor	Poor	Excellent	Good	Excellent	Fair	Good	Good
	Water Absorption	Very Good	Good	Very Good	Good	Fair	Good	Good	Very Good
	Oxidation	Good	Good	Excellent	Excellent	Good	Excellent	Excellent	Excellent
	Ozone	Fair	Fair	Excellent	Excellent	Fair	Excellent	Excellent	Outstanding
	Sunlight Aging	Poor	Poor	Very Good	Excellent	Poor	Excellent	Very Good	Outstanding
	Heat Aging	Good	Very Good	Excellent	Excellent	Excellent	Outstanding	Excellent	Excellent
	Flame	Poor	Poor	Poor	Poor	Poor	Fair	Good	Good
	Heat	Good	Excellent	Excellent	Excellent	Excellent	Excellent	Excellent	Excellent
	Cold	Excellent	Excellent	Good	Excellent	Good	Excellent	Good	Good

Table 4.1 – Properties of varying forms of rubber.

Once the rubber mixture has been determined it can then be spread over a textile of some kind to further increase its usefulness. In the motor vehicle industry this aspect of rubber technology plays a large part in enabling the rubber to perform its job properly. Steel and soft fibre belts in tyres and fabric sandwiched sheets in air bags are two examples of this method of reinforcing being used. Without the textiles holding the rubber together while under pressure it would very easily split and fail. The degree of benefit associated with the textile reinforcing will depend on three aspects of the textile. Firstly there will be an intermolecular attraction between the fibre and the polymer which will vary in strength and be governed by the choice of material making up the textile. Secondly the geometric structure of the thread or yarn will affect the mechanical adhesion between the rubber and fabric. A greater amount of thread ends will provide a better adhesion but may cause a decrease in strength because of the discontinuation of the thread within the textile. The last factor affecting the adhesion is the weave of the fabric. A more open weave will create more opportunity for the rubber to penetrate into the textile while a closed weave may decrease the quality of adhesion by limiting penetration. While modifying the properties of rubber with additives is a simple task performed by mixing, the use of textiles requires manufacturing processes to incorporate the rubber into the textile so that they cannot be easily stripped from each other during their lifespan.

Methods of manufacturing rubber components include the following processes:

- Extruding
  … Results in long lengths of various shapes. Some shapes are simple while others can be complex extrusions for use in specific applications like window trim.
- Calendering
  … Creates rubber in sheet form by passing it through series of rollers. This method can be used to incorporate textiles into the flat rubber sheet by rolling textile and rubber between the same rollers.
- Solvent Dispersion
  … Spreads rubber over textile by scraping a rubber dough across a sheet with a knife edge then evaporating the solvent and curing the left over rubber.
- Moulding
  … Used to form items of required shapes in moulds similar to metal casting.
- Bonding
  … Attaching rubber to foreign material such as metals to form composite products. This is usually performed with a process of vulcanisation.

4.3.2 Material Selection

Having acquired an understanding of the way rubber has evolved, is composed and is produced, the choice must be made as to which grade of rubber should be used for this specific application. The comparisons made in figure…give a very good overview of properties relevant to the application intended for its service. When comparing the materials a number of different forms of rubber seemed fitting. Natural rubber was ruled out immediately due to its unreliability and variability. Of the other rubbers, the properties found to have greatest weighting in this decision in order of importance were:

- Permeability to Gases
  … Needed to ensure that no air from within the bag will be likely to leak through the rubber and cause a decrease in car height.
- Tensile Strength
  … Needed to ensure the bag can withstand high pressures.

- Adhesion to Metal
  … Needed to ensure adequate hold and seal between the bag sections and the rigid rings. This part of the design is most susceptible to failure and must be treated with care.
- Resistance to Chemicals and Harsh Environment
  … Needed to ensure the rubber is stable when exposed to oil, fuel, sunlight, water and heat.
- Abrasion Resistance
  … Needed to ensure the bag is not likely to wear itself away in the action of compressing and expanding during its lifetime.

Assessing the permeability to gases helped narrow the choice of rubber to three possibilities including Butyl, Neoprene, and Hypalon. Further consideration of the tensile strength ruled out Butyl rubber and left only two options. Of these two forms of rubber the decision between the two was quite difficult. However Neoprene was found to be the most suitable material for due to its greater resilience or rebound. In an application where the rubber will continually be flexed and placed under varying stress it is important the rubber is not affected adversely by this motion but instead is able to retain its original form. Neoprene also offers excellent adhesion to metal, excellent resistance to all environmental considerations that may be of concern and finally it has excellent abrasion resistance.

The material selection is not complete without choosing appropriate textile reinforcement. Early on in the development of rubber the only reinforcing used was cotton because of its easy adhesion to the rubber. Now with the advent of adhesives, materials that do not naturally provide great adhesion to rubber can be made to function well as reinforcing. As a result of this synthetic fibres such as rayon, nylon and polyester are common reinforcing materials. From Table 6.16 found in Rubber Technology and Manufacture (1971) and reproduced below as Table 4.2, the available range of industrial reinforcing can be seen. This table defines polyester as the strongest form of reinforcing. However in the columns closer to the right of the table it can be seen that polyester does not offer the best strength to weight ratio. Instead Polyvinylalcohol seems to provide the best strength qualities due to its lower density and high level of strength. For these reasons polyvinylalcohol reinforcing has been chosen for the flexible sections of rubber. This reinforcing will ensure the longest possible lifespan and durability of the rubber material.

Material	Density (g/cm^3)	Strength (Kgf/mm^2)	Specific Strength		Modulus (kgf/mm^2)	Specific Modulus	
			(cm x 10^6)	(g/den.)		(cm x 10^6)	(g/den.)
Rubber (tread)	1.1	2	0.19	0.2	0.7	0.062	0.1
Silk	1.25	60	4.6	5	350	27.5	30
Cotton	1.54	70	4.6	5	1000	62.5	70
Rayon (dry)	1.52	70	4.6	5	1900	12.5	140
Nylon	1.14	90	8.1	9	500	38	40
Polyester	1.40	110	8.1	9	1100	76	90
Polyvinylalcohol	1.28	100	8.1	9	1600	125	120
Polypropylene	0.92	70	8.1	9	700	76	85
Glass	2.56	350	13.7	15	7500	280	320
Steel	7.8	280	3.8	4	21000	280	300
Carbon Fibre	1.95	210	11.2	12	42000	2150	2400

Table 4.2 – Mechanical properties of industrial fibres.

4.3.3 Manufacturing Process Selection

The sections of flexible bag pose an interesting manufacturing challenge. The odd shape and the circular nature will mean that custom tooling will need to be produced.

To gain the correct shape of these sections with the reinforcing sandwiched in the middle of rubber layers, the textile will firstly need to be custom woven into the required shape. If a standard fabric in sheet form was used, it would need to be cut and rejoined in multiple sections and would mean a sacrifice in strength. The best option is to manufacture three different pieces of woven fabric, for the inside sections, the outside sections and the top piece to join the inside rings to the outside rings. Custom woven sections will have continuous fibre and can be run through a series of shaped rollers for coating with rubber.

Calendering will not work as a means of coating the rubber because of the profile of the sections. If two custom rollers were made to sandwich the textile and press rubber into the fabric the rollers would create varying friction factors as they rolled together which would give varying degrees of adhesion. The best option for coating the fabric is via spreading. The profiled fabric will be spun by two sets of rollers creating a tight section of the fabric where a knife can be run against a flat surface to spread solvent thinned neoprene rubber across the fabric. This process will need to be done once for the inner coating and once for the outer coating. Following this the section of composite material can be heated and cured for a short period of time.

While this whole process will be extremely costly to set up, the benefit of ensuring the flexible sections are manufactured correctly is well worthwhile. Assuming this system is viable and begins mass production, the cost of tooling will become far outweighed by the efficiency and reliability of the process.

4.4 Rigid Rings

This component of the mechanism plays a number of important roles in the functioning of the system. The rings primary purpose is to hold the shape of the bag and by the angle of its faces must define the rate of area increase as the bag is compressed. The rings also join together the small sections of flexible bag to produce one continuous cavity and must therefore give adequate sealing on the sections of flexible bag. In doing these things the rings should also:

- Minimise or negate any wear of the bag.
- Be capable of withstanding degradation in harsh environments.
- Have adequate strength within the expected range of loading conditions.
- Achieve all these objectives with as little cost as possible.

4.4.1 Material Selection

The way in which the rigid rings have been designed mean they must join the flexible bag sections by clamping and sealing the free ends of rubber material. The rings must do this by grabbing the free ends inside a formed jaw that runs the full circumference of the disk. This clamping action means the rings should be made from a material that can be manufactured in some way to create the open jaw ready for assembly. Then once the rubber has been inserted into the jaw the material must be able to be formed, yield and allow the jaw to be closed. The closing action of the jaw must not work harden the jaw to the point where it has become weak. The material used for constructing these rings should be carefully selected to fulfil the mechanical requirements while also maximising cost effectiveness. In order to select the most appropriate material the following tables have been extracted from various texts to allow a comparison to be made between common steels, metals and alloys.

Material	Density	Cost/tonne	Relative	Cost/m^3	Relative
	kg/m3	$/tonne	$/tonne	$/m^3	$/m^3
Carbon Steel	7820	1375	1	10752	1,0
Alloy Steels	7820	2075	1,51	16225	1,5
Cast Iron	7225	2075	1,51	14990	1,4
Stainless Steel	7780	11125	8,1	86552	8,0
Aluminium/alloys	2700	5550	4,0	14985	1,4
Copper /Alloys	8900	13875	10,1	123487	11,5
Zinc alloys	7100	5550	4,0	39405	3,7
Magnesium /alloys	1800	10000	7,3	18000	1,8
Titanium /alloys	4500	42500	30,9	191250	17,4
Nickel alloys	8900	45000	32,7	400500	36,8

Table 4.3 – Costing for various types of steel, metal and alloy.

The table above should not be trusted as an accurate source of market price. Metal prices are always changing and would need to be confirmed just prior to purchase so as to achieve an accurate price estimate. However this table does provide a good price level comparison which should be accurate given any fluctuations in market price.

It is not necessary to choose the cheapest material for these rigid rings. However it is more desirable to have a lower costing material if all the other design parameters are met by the material. Costs do not generally increase directly in relation to physical properties of a material. A situation may exist where two materials with similar physical properties are priced very differently. Ideally a balance should be struck to achieve the best “bang for you buck”. This saying means that a material should not be chosen based solely on either cost or physical properties. Instead the two should be played against one another to ensure a great deal of extra money is not being spent for little or no increase in physical properties.

Metal	Fresh Water	Sea Water	Air
	Static/Turb	Static/Turb	City/Indust
Grey Cast Iron- Plain or Low Alloy	4/3	4/3	3
Cast Iron Ni_Resist. (14% Ni, 7% Cu, 2% Cr bal Fe)	5/5	5/5	4
Ductile Iron Ni_Resist. (24 % Ni, bal Fe)	5/5	5/5	4
Mild Steel- Low Alloy steels	4/3	4/2	3
Stainless Steel Ferritic (17% Cro)	4/6	1/4	3
Stainless Steel Austenitic (18% Cro ,8% Ni)	6/6	2/5	4
Stainless Steel Austenitic (18% Cro 12% Ni, 2.5% Mo)	6/6	3/5	6
Stainless Steel Austenitic (20% Cro 29% Ni, 2.5% Mo,3.5% Cu)	6/6	4/6	6
Copper Nickel alloys (Up to 30% Ni)	6/6	6/6	5
Nickel Commercial (99% Ni)	3/5	6/6	4
Aluminium Brass	6/6	4/5	5
Bronze (88% Cu,5% Sn,5% Ni,2% Zn)	6/6	5/5	5
Aluminium Alloys	4/5	0-5/4	5
Titanium	6/6	6/6	6

Table 4.4 – Corrosion resistance of various steels, metals and alloys

The table shown here gives a rating for how affective various metals are at resisting corrosion in specific environments. The scale of ratings begins at zero and designates the metal is unsuitable for a specific application. The best rating is six and designates a metal is excellent for service in the indicated environment.

The metal components of this air spring can be made subject to all three of the environments included in Table 4.4. Values of corrosion for a static and turbulent fluid are given in the table however for this process of material selection the corrosive fluid is assumed to be static. Reasonable corrosive resistance in these solid rings is a requirement. Selecting a material that can provide absolute corrosive resistance would ensure greatest longetivity of the system but may not be possible due to costs.

Material	Cast Iron	Carbon Steel	Alloy Steel	Stainless Steel	Aluminium and Alloys	Copper and Alloys	Zinc and Alloys	Magnesium and Alloys	Titanium and Alloys	Nickel and Alloys	Refractory Metals
Sand Casting											
Investment Casting											
Die Casting											
Impact Extrusion											
Cold Heading											
Closed Die Forging											
Powder Metal Processing											
Hot Extrusion											
Rotary Swaging											
Machining (from stock)											
ECM											
EDM											
Sheet Metal (stamp/bend)											

Table 4.5 – Compatibility between processes and materials

This table shows manufacturing processes that can be performed on various common types of materials. White squares indicate that the process is readily performed on a material. Grey squares indicate the process is not commonly used on a specific material but does not rule out its use. Finally the black squares indicate the process is unsuitable for the material designated.

The solid rings for this air spring will need to eventually become a mass produced item. Not only will the system be produced on a production line but each spring assembly will require multiples of these rings. For this purpose the manufacturing processes chosen may require significant tooling and this can be justified in the long term.

Material	Yield Stress (MPa)	Ultimate Stress (MPa)	Elongation (%)
Carbon Steel	350	520	30
Alloy Steels	290	415	24
Cast Iron	350	450	15
Stainless Steel	200	500	40
Aluminium/alloys	90	150	12
Copper /Alloys	350	551	50
Zinc alloys	200	300	7
Magnesium /alloys	120	200	7
Titanium /alloys	140	240	54
Nickel alloys	860	1000	20

Table 4.6 – Strength values for various types of steel, metal and alloy.

This table shows the ultimate or tensile stress, yield stress and percentage elongation before failure for some common metals. The strength required by these rings is unknown at present but can be calculated easily given that:

- Maximum load per strut is 500 kg.
- The maximum load per strut is applied to the smallest spring area. (i.e. the spring is fully extended)

- The stress in the rings can be approximated by evaluating the hoop stress as if the rings were a part of a cylindrical pressure vessel. This assumption will not be entirely accurate so the stress will be increased by a factor two. The increase is necessary because the rings are not closed pressure vessels and because of this they cannot be expected to have the same rigidity or strength.

Now the smallest area on which the spring can apply its pressure is when it is fully extended and the bag has not been rolled out upon the rings. At this point area **A** is equal to the area contained by the external radius $\mathbf{r_o}$ minus the area contained by the internal radius $\mathbf{r_i}$.

$$r_o = 56.5\,mm = 0.0565\,m$$
$$r_i = 38.5\,mm = 0.0385\,m$$

$$\therefore A = \left(\pi \cdot r_o^2\right) - \left(\pi \cdot r_i^2\right) \tag{4.1}$$

$$A = 5371\,mm^2 = 0.005371\,m^2$$

The pressure within the spring can be calculated with equation (4.2):

$$P = \frac{F}{A} \tag{4.2}$$

$$\therefore P = \frac{5000\,N}{0.005371\,m^2}$$

$$P = 931\,kPa$$

Now the formula for calculating hoop stress is:

$$\sigma_h = \frac{P \cdot r}{t} \tag{4.3}$$

Where **P** is the pressure of the cylindrical vessel, **r** is the radius of the cylinder, and **t** is the thickness of the wall. The radius and thickness are taken off the final design for the rigid rings and the values are measured at the outermost point of the external ring. This position has been assumed to be susceptible to the most stress.

$$r = 83\,mm = 0.083\,m$$
$$t = 4\,mm = 0.004\,m$$

$$\therefore \sigma_h = \frac{931000 \cdot 0.083}{0.004}$$

$$\therefore \sigma_h = 19.3\, MPa$$

After taking into account the double factor for inaccuracies of the hoop stress model and also apply a factor of safety of three, the required yield stress of this rigid ring material is found to be:

$$\sigma_{yield} = \sigma_h \cdot \left(2\, times\, inaccuracy\, factor\right) \cdot \left(3\, times\, safety\, factor\right) \quad (4.4)$$

$$\sigma_{yield} = 115.8\, MPa$$

Therefore when using the Table 4.6 the smallest allowable yield stress for selecting a material is 115.8 Mega Pascals.

All the tables above can now be compared and contrasted with each other to select the most appropriate material. These tables define the most important aspects for our material selection and cover a wide range of possibly appropriate material.

The material chosen after a process of elimination is alloy steel. The reason for this choice is heavily weighted on the price. A step up from alloy steel to a stainless steel will see the price increase five fold. Also in many cases for stainless steels the corrosion resistance in static salt water is not significantly better than alloy steel. The strength of alloy steel is very sufficient for this application and it can be put through every manufacturing process we might be inclined to use. Other steels that were considered were the stainless steels and copper alloys. These two materials were compatible with all the manufacturing processes we desired and copper alloys in particular had exceptional corrosive resistance. However both these materials would have required much more cost and were therefore deemed unacceptable. Although alloy steel does not have perfect corrosion resistance it should provide enough resistance to ensure a long life in this application and prevent wear of the flexible rubber bag. When this system is physically tested the corrosion resistance can be observed and the material can be upgraded to stainless steel or copper alloy if needed. For now though the cost effectiveness of the alloy steel make it the best option for the rigid rings.

4.4.2 Manufacturing Process Selection

The rigid rings provide a difficult problem to solve as far as manufacturing processes are involved. The rings are rather small and intricate and need to be finished smoothly. On top of this the shape of them as shown in Figure 4.4.2.1 means that simply machining them out of a billet will create far too much waste. The figure shows two sets of jaws which represent both the manufactured shape (solid line) and the squashed jaw position (dashed line) after the rubber has been inserted. Upon consideration it is evident that no single process can be used to manufacture these rings. Some combination of processes will be needed to bring these rings to a point where they meet the desired specifications for this design. If there are to be multiple processes used it seems obvious that the first process will be used to form the basic shape while the second process will create the required surface finish as well as the detail.

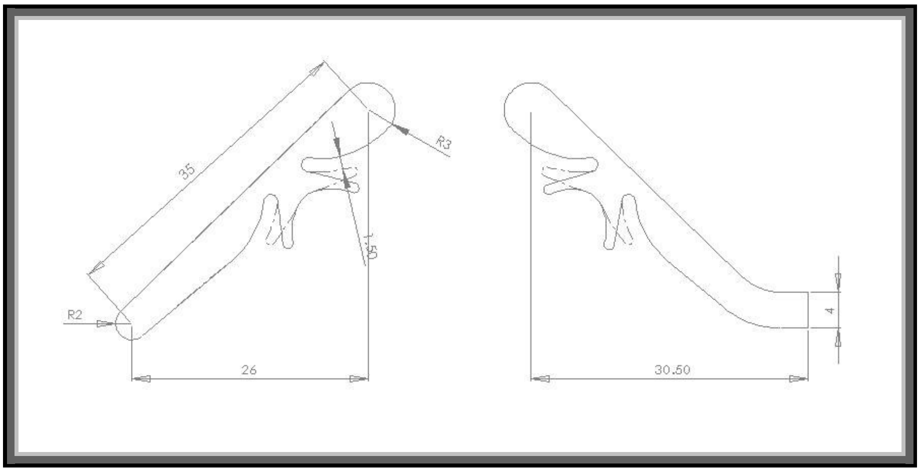

Figure 4.4.2.1 – Revolved profile of both internal and external rigid rings.

The first process can be any number of manufacturing processes. The roughed out shape hopefully obtained from the first process is shown in Figure 4.4.2.2. The dotted lines in this figure represent the area to be removed in the second process. The rings could be formed via casting; however investment casting which is costly would be the only casting method capable of achieving a smooth surface finish. Given that simple machining processes are out the only other attractive solution form roughing out these parts is hot forging. This process will enable a smooth finish over most of the part and can create the exact shape we need with great repeatability. Hot forging this part will involve starting with a disc of hot material that can be sat in the bottom of a two part mould. The top section of mould is then dropped to meet the first and stamp the hot material between the two dies until all cavities of the mould have been filled. Hot forging will leave a flash around the inside and outside of the part due to excess material that is not able to fit in the mould once it has been closed. This flash is necessary to ensure the mould is completely filled.

Once the hot material has been stamped for the first time and the shape has been made the flash can be removed roughly in a second stamping process. Further removal of flash and cleaning will need to be performed in the second process.

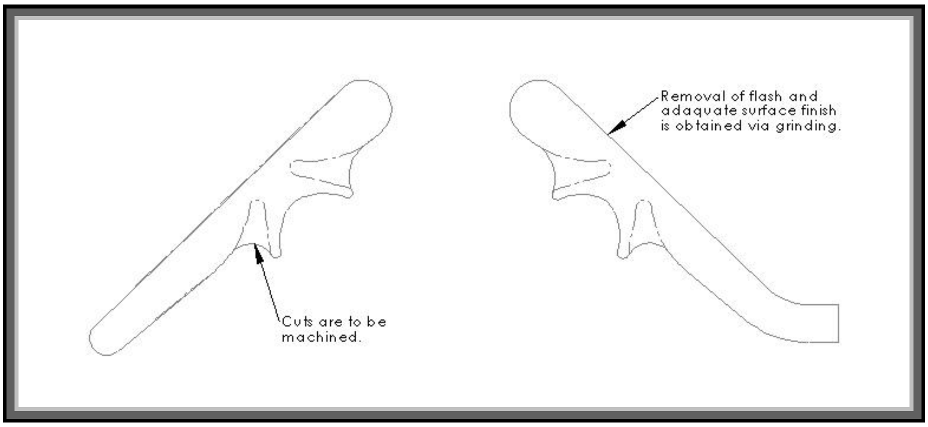

Figure 4.4.2.2 – Revolved profile of rough shape.

Once forging of these rings have been completed the jaws must be cut into the rough shape and the whole part must be cleaned and smoothed. Given these rings are circular the most simple option for completing this step of manufacture is turning these features into the rings. Both internal and external rings may be held in a machine centre while the necessary cuts are made with a tooling point. Once the cuts have been made then the whole disk can be smoothed with a fine grinding operation also performed in a machine centre.

The processes selected to manufacture the rigid rings have been selected out of necessity. The rings have been designed to perform a task and could not be modified in any way to allow a more straight forward manufacturing process. While in many cases a part may be designed with manufacturing in mind this was not the case with these rings. Despite that fact the manufacturing processes assigned to these disks should allow good repeatability and cost effectiveness when put into a production line.

4.5 Upper Mount

Part of this design is allowing the spring to be fitted into a position where an existing spring is already located. For this to be done it needs to be fixed top and bottom to an already existing mounting point. The top mounting point for this spring is a flat surface where the spring can bolt onto at the top of the wheel arch.

The final design of the top mount is shown in Figure 4.5.1 and has been created from two pieces that will be stitch welded together. The material for the upper mount will be alloy steel to match the material used in making the rigid rings. As with the rigid rings, the lower piece of this mound will be created by hot forging a billet of material and then grinding the surface smooth. The upper section of this mount can be manufactured by stamping a disc of flat sheet that has been laser cut. The stamping will require a former to be manufactured but this process need not be completed with any great form of accuracy. The only concern for the upper section is that the holes are consistently positioned at the right pitch circle diameter.

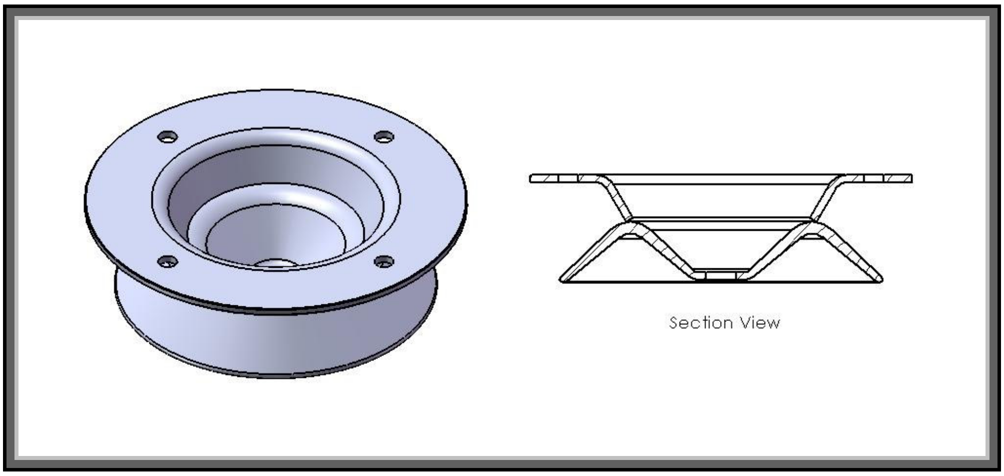

Figure 4.5.1 – Upper mount.

The hole in the middle of the forged component makes provision for the shock absorber shaft to protrude through this plate and also be attached to the car.

This top mount is not attached to the rest of the spring but simply sits on the uppermost point of the flexible bag. This loose method of attaching the top mount is acceptable because the shock absorber and its shaft will not allow the lower sections of the spring to fall away or move sideways in relation to the top mount.

4.6 Lower Mount

The bottom mount of the spring poses a much more challenging concept. This part of the spring has one more job than the upper mount which is to provide an outlet for the air in the cavity to be transported to an accumulator. In order to perform this task the lower mount has been designed as a cavity with two openings. One of these openings is a large circular ring that permits the flexible bag to be joined onto the base and provide free flow of air from the bag into the lower mount. The other opening consists of multiple holes that allow the lower mount to be connected to an accumulator mounted remotely from the spring.

Manufacturing this item can be done in a number of ways. One idea is to cast the lower mount. A concept for the casting is shown in Figure 4.6.1. The reason this concept is not one whole casting but instead includes a capping plate is to allow access from the underside of the casting to fix the flexible bag in place. Once the flexible bag has been secured to the casting then the capping plate is attached and a perfect seal is obtained. The large threaded holes in the capping plate provide a place for hose fittings to be attached to the bottom of the base. These hoses will link the base to the accumulator.

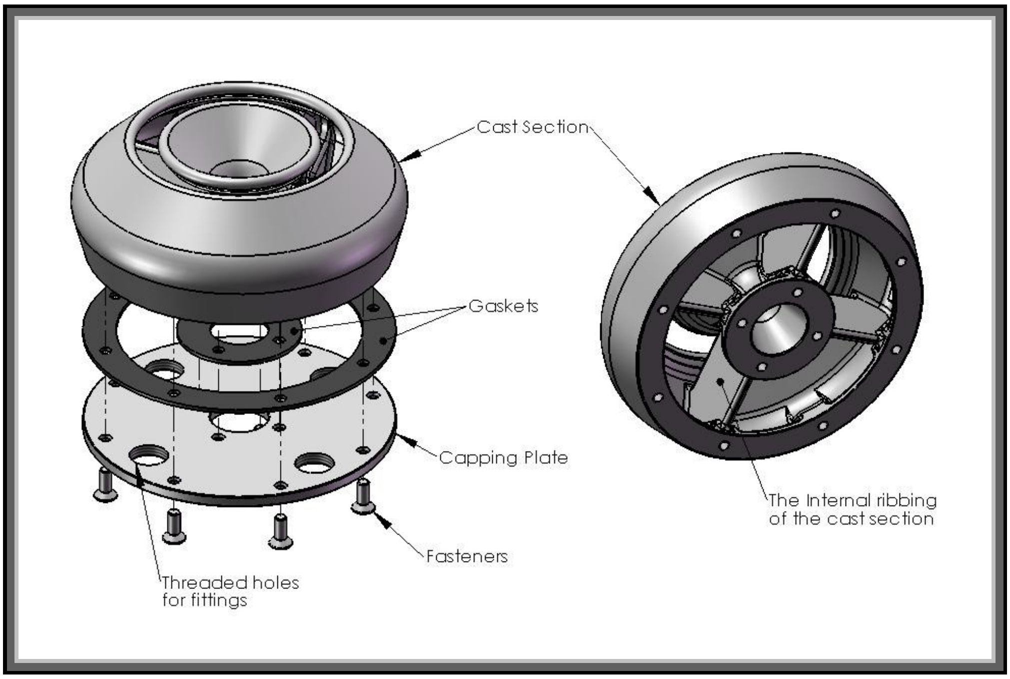

Figure 4.6.1 – Lower mount manufactured by casting.

While casting the lower mount can certainly be done successfully it may not be the most economical way to produce this component. Casting such a complex part will be costly and time consuming and then once it has been cast it will still require some machining. A better option for manufacturing this item is to make use of the rigid rings that are already being manufactured and place them within a welded assembly to achieve the same result. The lower mount manufactured by welding is shown in Figure 4.6.2. The welded assembly is comprised of standard rings at the top of the mount. These rings are then welded to a rolled piece of flat creating a cylinder for the outside and a short piece of heavy walled tube creating a cylinder for the inside. At the base of the mount is a cut, drilled and tapped piece of plate that joins the inner and outer cylinders and has gusseting strengthening the transition.

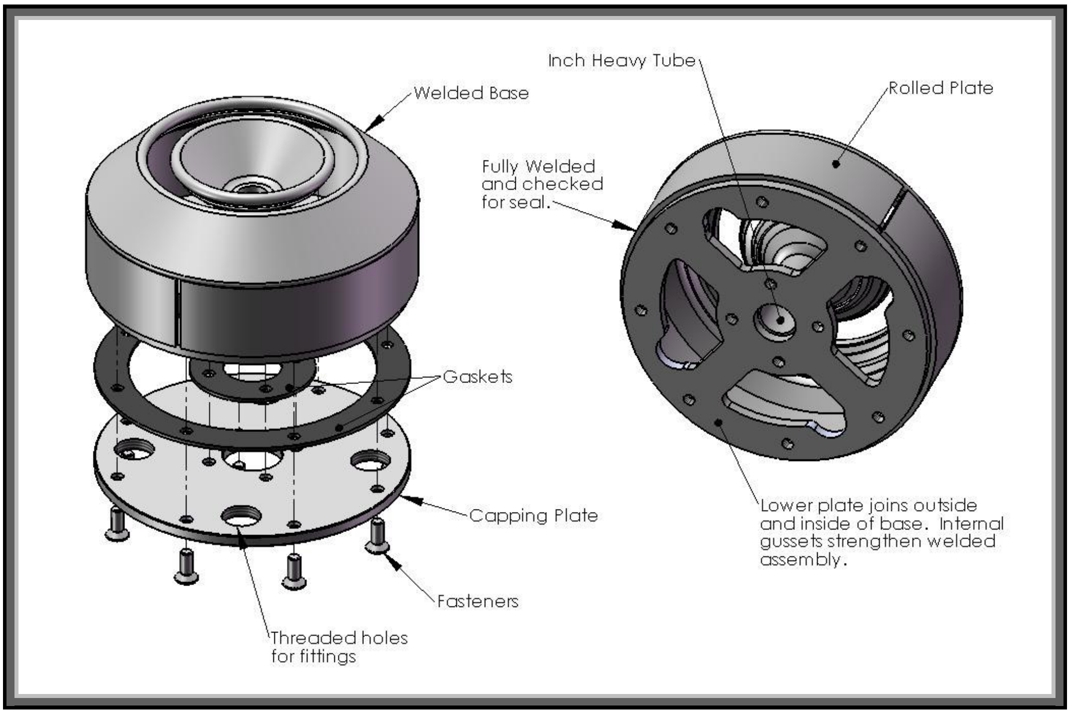

Figure 4.6.2 – Lower mount manufactured by welding.

This method of manufacturing the lower mount seems much more straight forward to make and far less costly. By using this method the tooling created for the rigid rings is serving a purpose in this area of the design as well. Also any additional components needed are very simple to manufacture. For all these reasons the base will be welded.

4.7 Conclusions: Chapter 4

Manufacturing and materials selection has been conducted for the major components of this design. The air spring will be manufactured as four distinct parts then assembled.

The flexible rubber sections are to be manufactured from neoprene rubber with polyvinylalcohol reinforcing fibre by a process of spreading. The rigid rings will be manufactured from alloy steel by firstly hot forging the rough shape and then machining and grinding to a finished product in a machine centre. All steel components will be made from the same alloy steel used in these rigid rings. The top mount will be comprised of a hot forged then grinded top ring welded to a pressed laser cut plate. Finally the lower mount will be comprised of standard rings and various plates and tube sections to form a welded assembly.

Assembly will involve joining the flexible rubber sections by firstly vulcanising the rubber to the inside of the jaws found on each rigid ring. Once the vulcanising has been completed the jaws will be crimped shut to provide extra hold on the flexible bag sections.

The information contained in this chapter can now be used to implement prototype creation and lead into physical testing.

CHAPTER 5

VIBRATION ANALYSIS

5.1 Introduction

The effectiveness of any suspension is decided by how successfully it carries a vehicle on the road when subject to unexpected and continually variable conditions. The suspension will determine what kind of reaction a vehicle will have when driven over roads with specific forms of undulation. A vehicle primarily designed for sealed roads and fast driving will generally not be equipped with a suspension that will allow comfortable driving on unsealed roads. In the same manner an off road vehicle as a generalisation will not perform well at high speeds when taken around corners. In order to determine what kind of performance can be expected from a particular form of suspension it is wise to conduct a vibration analysis. When this analysis is performed before manufacture the vibration response of the system can be estimated to ensure the desired road holding and handling performance is met.

In the case of this new form of suspension it is important that its fundamental success is ensured before any physical testing is done and too much money is wasted on the idea. For an analysis of this kind on a new form of suspension it would be desirable to have a set of standard results to compare and contrast the new data with. Therefore a vibration analysis will be simulated on the existing suspension and results recorded before any analysis is conducted on the new suspension.

5.2 Evaluation Method

When conducting any form of testing, there needs to be criteria for determining the success or failure during the testing procedure. These criteria cannot be developed once the tests have been run because in most cases the success criteria will determine the measurements taken throughout the test. Without a criteria defining success there is no telling what the design parameters are and whether they have been met and to what extent. For instance if a formula one racing team were to consider an FJ Holden as a possibility for their vehicle in next years competition and set about testing its performance, how would they determine it unfit? They may have the idea that if the car can run and drive it must be fit to be used in this competition. However if this was the case they would be terribly mistaken! There are two ways they can go about testing the vehicle. Firstly they can evaluate to aims of the competition which would be maximum speed, handling achieved within a set or rules and guidelines. Given this criteria they can then set about testing and developing the car to evaluate and improve its current speed. Secondly they can compare their vehicle to others in the competition. While their team may be able to squeeze every last ounce of speed and manoeuvrability from this FJ, it will never compare to the speed of a real formula one race car. Therefore in evaluating success it is also important to have some means of comparing a new design to existing ones.

The purpose of suspension on any vehicle is to smooth out an undulating road and in doing so improve:

- Driver comfort
- Stability
- Safety
- Carrying ability
- The life of the vehicle

The main aspect that will be studied in this analysis will be driver comfort. Determining a vehicles overall stability and safety would require a more complex model than the one used in this study. This study will only take into account straight line motion and while this does not come close to mimicking a real car it all its forms of operation, the scope of this study does not allow time for a deeper analysis. This new form of suspension is destined to be used in a wide variety of road conditions contingent on its success in those situations. Therefore by comparing this new suspension with an existing one and firstly assessing the aspects of vibration that affect comfort, the overall success can be roughly measured. In all likeliness if the suspension can perform within comfortable limits for various road conditions then it has a good chance of also providing the stability and safety required for its complete success.

Throughout the development of suspension there have been a number of generic guidelines developed to assist in manufacturing a successful system of suspension. According to Braun (2002) these guidelines can be summarised in the following four points:

- In view of the human comfort, the suspension systems should be designed to achieve low vertical mode natural frequency of the sprung mass. A limiting value for this natural frequency is approximately 1 Hz to ensure adequate rattle space and it should not be greater than 1.5 Hz for passenger vehicles.

- Forces due to wheel motions and unbalance are transmitted to the sprung mass through the suspension. The unsprung mass (wheel hop) natural frequency therefore should lie outside the frequency range of vibration to which the human body is most sensitive. This implies that the vertical mode frequency of the unsprung mass should not be less than 8 Hz. On the other hand, larger frequencies will demand stiff tire affecting ride quality. A practical value for unsprung mass natural frequency is around 10 Hz.

- Pitch and bounce frequencies should be close together. The pitch motion of the sprung mass enhances the bounce motion of the body at a location away from the c.g., such as the driver location in a long wheelbase truck or bus. The bounce natural frequency less than 1.2 times the pitch frequency gives good results.

- When a vehicle goes over a bump, the front axle is subjected to the impact occurring at τ_d(L/V) seconds before the rear axle. This will excite the pitch resonance, which is more annoying than the vertical motion. Designing rear suspension with slightly larger ride rate than the front will introduce higher frequency of oscillation for the rear than the front. This will convert the pitch motion to a bounce motion within half a cycle after the bump is passed. Based on common operating speeds, V, and wheelbase, L, the rear suspension may be assigned 20-30 percent larger ride rate, or the c.g. should be closer to the rear axle then the front.

These considerations in suspension manufacture have been found to be most important and during simulation it will be these factors that we use to measure success.

5.3 Theoretical Modelling Methods

A dynamic analysis on an automobile can be performed in numerous ways depending on the desired complexity of the model and solution. The difficulty with modelling a system like car suspension is dealing with increasing numbers of degrees of freedom (DOF). The complexity of the model is decided by the number of DOF's to include within the model.

The most complex of these models requires seven degrees of freedom to define its motion. This model known as a full-car model assigns a DOF for each of the four wheels and then three DOF's for the body of the car. The body is defined with one DOF assigned to the vertical displacement of the centre of gravity and the other two DOF's assigned to the pitch and roll. This model is most accurate and can be used for dynamic analysis of countless situations including:

- Variations in road texture across the vehicle
- Variations in centre of gravity
- Cornering
- Accelerating and decelerating

This model is most suited for an accurate and in depth analysis but its complex solution places it outside the scope of this study.

The simplest form of model for a system of suspension is performed with only two DOF's. This model is known as a quarter-car model because it takes into account only one strut of the vehicle. The DOF's in this model are assigned to the wheel and the body of the car connected to the wheel via the strut. This type of model is limited in its ability to assess a suspensions success. It may serve a purpose during initial feasibility testing but as an isolated strut it ceases to provide an accurate example of the cars behaviour at the mercy of the spring. So much of the vibration response comes as a result of the interaction between each of the four wheels of the car and isolating one can cause misleading results.

A form of simulation that takes a middle ground between both of the previous methods is the half-car model. This way of simulating a car requires four DOF's. This model can be seen as a 2D side on view of a car with the wheels, the vertical height of the c.g. and the pitch accounting for each DOF. While this model is not as accurate or realistic as the full-car model it allows for a much simpler analysis due to the fewer DOF's involved. Not only this but because the front and rear wheels are considered together the pitching of the car can be simulated. Even though the roll is not considered in this model the pitching is considered a far more important aspect of the car when determining the vibration response on a straight stretch of road. Once this method of simulation has been mastered the step up to modelling with seven DOF would be much easier to understand and make successful.

The theoretical model being used in this study is the half-car or four DOF model. This was chosen because of its much simpler nature in comparison to the larger model. The larger model only really proves its worth once different road inputs are used for both sides of the car and when lateral accelerations are applied. Within the scope of this investigation the simpler model can return accurate and informative results that satisfy our aims and objectives.

5.4 Creating a Half-Car Model

The process of testing a new suspension was undertaken with the use of a four DOF, half-car model. Before any simulation or testing can be conducted the model must first be created and formulated to a point where values associated with the suspension can be entered into the model and vibration response data can be returned. To begin this process the vehicle with all its springs, dampers and masses has be converted into a set of equations that define the relationships of relative motion between all these components. The first step in this process is creating a free body diagram (FBD) of the 2D car with four degrees of freedom. This FBD can be seen in Figure 5.4.1 and shows the critical positions as being the centre of the wheels, the centre of gravity for the car and the road position for both front and back wheel.

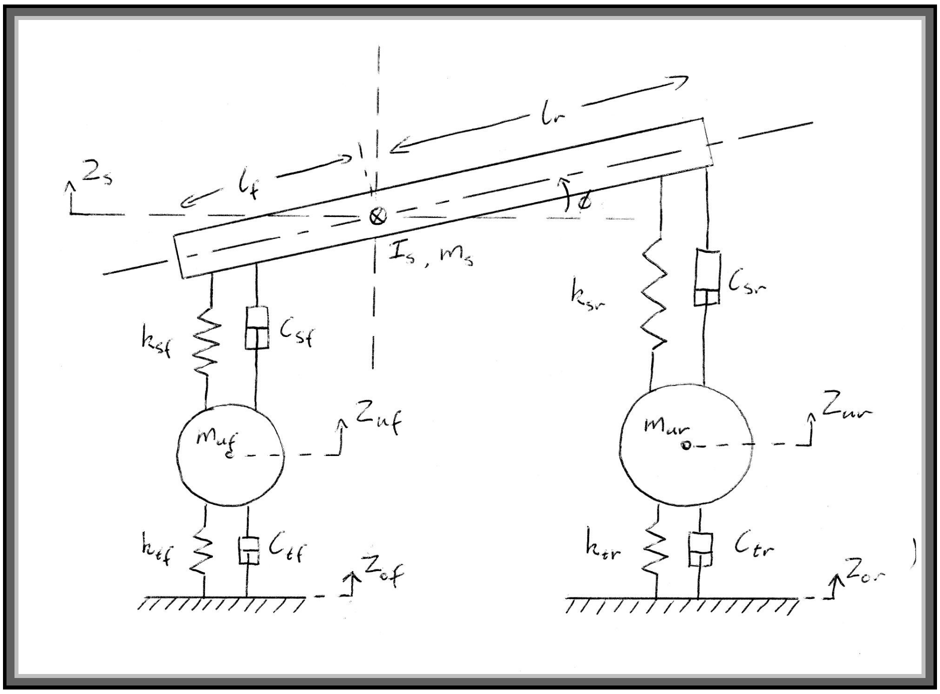

Figure 5.4.1 – Free body diagram for a half-car model.

The variables of interest to this investigation are listed below. These variables will facilitate the visualisation of the motion of each component of the vehicle model.

- z_s
- $\dot{z}_s$
- $\ddot{z}_s$
- ϕ
- $\dot{\phi}$
- $\ddot{\phi}$
- z_{uf}

- $\dot{z}_{uf}$
- $\ddot{z}_{uf}$
- z_{ur}
- $\dot{z}_{ur}$
- $\ddot{z}_{ur}$

From the FBD above, the equations of motion for each distinct part of the vehicle can be written. These equations take the form of the general equation of motion (5.1). These equations are derived by considering each part of the model individually and accounting for all the forces affecting its motion. The arrow directions in the FBD indicate

$$m \cdot \ddot{z} + c \cdot \dot{z} + k \cdot z = 0 \tag{5.1}$$

The equation of motion for the vertical motion of the car body (sprung mass) is found to be:

$$m_s \ddot{z}_s + k_{sf}\left[-z_s + \left(l_f \phi\right) + z_{uf}\right] + k_{sr}\left[-z_s - \left(l_r \phi\right) + z_{ur}\right]$$

$$+ c_{sf}\left[-\dot{z}_s + \left(l_f \dot{\phi}\right) + \dot{z}_{uf}\right] + c_{sr}\left[-\dot{z}_s - \left(l_r \dot{\phi}\right) + \dot{z}_{ur}\right] = 0 \tag{5.2}$$

The equation of motion for the rotation of the car body (sprung mass) is found to be:

$$I_s \ddot{\phi} + l_f k_{sf}\left[z_s - \left(l_f \phi\right) - z_{uf}\right] + l_r k_{sr}\left[-z_s - \left(l_r \phi\right) + z_{ur}\right]$$

$$+ l_f c_{sf}\left[\dot{z}_s - \left(l_f \dot{\phi}\right) - \dot{z}_{uf}\right] + l_r c_{sr}\left[-\dot{z}_s - \left(l_r \dot{\phi}\right) + \dot{z}_{ur}\right] = 0 \tag{5.3}$$

The equation of motion for the front wheel (front unsprung mass) is found to be:

$$m_{uf} \ddot{z}_{uf} + k_{sf}\left[z_s - \left(l_f \phi\right) - z_{uf}\right] + k_{tf}\left[-z_{uf} + z_{of}\right]$$

$$+ c_{sf}\left[\dot{z}_s - \left(l_f \dot{\phi}\right) - \dot{z}_{uf}\right] + c_{tf}\left[-\dot{z}_{uf} + \dot{z}_{of}\right] = 0 \tag{5.4}$$

The equation of motion for the rear wheel (rear unsprung mass) is found to be:

$$m_{ur}\,\ddot{z}_{ur} + k_{sr}\left[z_s + \left(l_r\,\phi\right) - z_{ur}\right] + k_{tr}\left[-z_{ur} + z_{or}\right]$$

$$+ c_{sr}\left[\dot{z}_s + \left(l_r\,\dot{\phi}\right) - \dot{z}_{ur}\right] + c_{tr}\left[-\dot{z}_{ur} + \dot{z}_{or}\right] = 0 \qquad (5.5)$$

In order to use these equations within matrices for the purpose of solving the dynamic problem they must first be rearranged into multiples of the variables of interest that were listed below the FBD.

From equation (5.2):

$$m_s\,\ddot{z}_s + z_s\left(-k_{sf} - k_{sr}\right) + \dot{z}_s\left(-c_{sf} - c_{sr}\right) + \phi\left(l_f\,k_{sf} - l_r\,k_{sr}\right)$$

$$+\dot{\phi}\left(l_f\,c_{sf} - l_r\,c_{sr}\right) + z_{uf}\,k_{sf} + \dot{z}_{uf}\,c_{sf} + z_{ur}\,k_{sr} + \dot{z}_{ur}\,c_{sr} = 0 \qquad (5.6)$$

From equation (5.3):

$$I_s\,\ddot{\phi} + z_s\left(l_f\,k_{sf} - l_r\,k_{sr}\right) + \dot{z}_s\left(l_f\,c_{sf} - l_r\,c_{sr}\right) + \phi\left(-l_f^2\,k_{sf} - l_r^2\,k_{sr}\right) + \dot{\phi}\left(-l_f^2\,c_{sf} - l_r^2\,c_{sr}\right)$$

$$-z_{uf}\left(l_f\,k_{sf}\right) - \dot{z}_{uf}\left(l_f\,c_{sf}\right) + z_{ur}\left(l_r\,k_{sr}\right) + \dot{z}_{ur}\left(l_r\,c_{sr}\right) = 0 \qquad (5.7)$$

From equation (5.4):

$$m_{uf}\,\ddot{z}_{uf} + z_s\,k_{sf} + \dot{z}_s\,c_{sf} - \phi\left(k_{sf}\,l_f\right) - \dot{\phi}\left(c_{sf}\,l_f\right) + z_{uf}\left(-k_{tf} - k_{sf}\right)$$

$$+\dot{z}_{uf}\left(-c_{tf} - c_{sf}\right) = -k_{tf}\,z_{of} - c_{tf}\,\dot{z}_{of} \qquad (5.8)$$

From equation (5.5):

$$m_{ur}\,\ddot{z}_{ur} + z_s\,k_{sr} + \dot{z}_s\,c_{sr} + \phi\left(k_{sr}\,l_r\right) + \dot{\phi}\left(c_{sr}\,l_r\right) + z_{ur}\left(-k_{tr} - k_{sr}\right)$$

$$+\dot{z}_{ur}\left(-c_{tr} - c_{sr}\right) = -k_{tr}\,z_{or} - c_{tr}\,\dot{z}_{or} \qquad (5.9)$$

Now these equations form the basis for our vehicle model and can now be used for two different forms of analysis covered in the following two sections.

Before the analysis can begin, the values for each of the unknowns in the preceding equations must be found. In the case of this analysis the values will be specific to a Holden Commodore as outlined in chapter two.

Taken from chapter two and supplied by the manufacturer are the spring and damping rates:

$$k_{sf} = k_{sr} = 21.6\ {}^{kN}\!/_{m}$$

$$c_{sf} = c_{sr} = 0.8\ {}^{N}\!/_{(m/s)}$$

According to the vehicle manual the maximum allowable loads for each of the axles measured on a weighbridge is:

$$Maximum\ Front\ Axle\ Load = 900kg$$

$$Maximum\ Rear\ Axle\ Load = 1025kg$$

Contained within these maximum axle loads are the unsprung masses. The values shown below for this analysis correspond to only one side of the car and therefore only half of the total unsprung mass of the vehicle.

$$m_{uf} = 30kg$$

$$m_{ur} = 60kg$$

Also from the vehicle manual are the dimensions of the car:

$$Wheelbase = 2.7m$$

$$Height = 1.45m$$

$$Length = 4.8m$$

By working with the axle loads and the wheelbase a simple force analysis yields the distances between the centre of gravity and the wheels:

$$l_f = 1.4m$$

$$l_r = 1.3m$$

The mass of the vehicle for this analysis can be derived from the axle loads and the unsprung masses:

$$m_s = \frac{(900kg + 1025kg)}{2} - m_{uf} - m_{ur} = 872kg$$

The second moment of inertia for the vehicle can be found by considering the side of view of the car as a rectangle and using equation (5.10).

$$I = \frac{1}{12} m \left(l^2 + h^2\right) \tag{5.10}$$

Where I is the second moment of inertia, m is the mass of the vehicle and the other two variables designate the length and height. Therefore the second moment of inertia for the car assuming that the centre of gravity sits half a metre from the ground is:

$$I_s = \frac{1}{12} m_s \left(l^2 + h^2\right) = 1674\, kg \cdot m^2$$

The value for a spring and damping constant to represent the tyres contact with the road must be determined approximately. As the tyre comes in contact with the road, the area over which it applies its pressure increases as shown in Figure 5.4.2. which results in a spring effect.

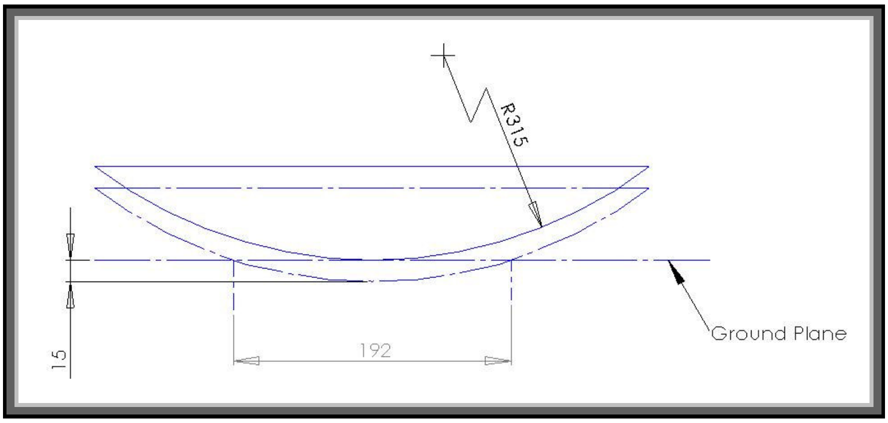

Figure 5.4.2 – Tyre contact with road.

Assuming a tyre pressure of 40PSI or 276kPa and drawing dimensions from Figure 5.4.2, then the spring constant for the tyres will be approximately:

$$k_{tf} = k_{tr} = 847.2\ {}^{kN}\!/_{m}$$

The damping constant for the tyres is not able to be accurately estimated and will simply be given an identical value to the dampers joining the wheels to the vehicle. During analysis the value will be modified if need be.

$$c_{tf} = c_{tr} = 0.8\ {}^{N}\!/_{(m/s)}$$

Having defined every variable associated with the theoretical model of the Holden Commodore the analysis for this vehicle is ready to proceed. A summary of all the determined values is given below:

- $m_s = 872kg$
- $I_s = 1674\ kg \cdot m^2$
- $m_{uf} = 30kg$
- $m_{ur} = 60kg$
- $k_{sf} = 21.6\ kN/m$
- $k_{sr} = 21.6\ kN/m$
- $c_{sf} = 0.8\ N/(m/s)$
- $c_{sr} = 0.8\ N/(m/s)$
- $k_{tf} = 847.2\ kN/m$
- $k_{tr} = 847.2\ kN/m$
- $c_{tf} = 0.8\ N/(m/s)$
- $c_{tr} = 0.8\ N/(m/s)$
- $l_f = 1.4m$
- $l_r = 1.3m$

5.5 Free Vibration Analysis

The first way in which the equations developed in section 5.4 can be used is to run a free vibration analysis to determine natural frequencies by making the variables z_{or} and $\dot{z}_{or}$ equal to zero. By doing this we can remove the forced vibration component of the model. Making these terms zero will make the right hand side of equations (5.8) and (5.9) equal to zero. The resulting equations are shown below:

$$m_{uf}\,\ddot{z}_{uf} + z_s\,k_{sf} + \dot{z}_s\,c_{sf} - \phi\left(k_{sf}\,l_f\right) - \dot{\phi}\left(c_{sf}\,l_f\right) + z_{uf}\left(-k_{tf} - k_{sf}\right)$$

$$+\,\dot{z}_{uf}\left(-c_{tf} - c_{sf}\right) = 0 \qquad (5.11)$$

$$m_{ur}\,\ddot{z}_{ur} + z_s\,k_{sr} + \dot{z}_s\,c_{sr} + \phi\left(k_{sr}\,l_r\right) + \dot{\phi}\left(c_{sr}\,l_r\right) + z_{ur}\left(-k_{tr} - k_{sr}\right)$$

$$+\,\dot{z}_{ur}\left(-c_{tr} - c_{sr}\right) = 0 \qquad (5.12)$$

Using equations (5.6), (5.7), (5.10), and (5.11), the mass (**M**), damper (**C**) and spring (**K**) matrices may be created for use in a matrix version of the equation of motion (5.12) shown below. These matrices are created by filling in 4x4 matrices with the multiples of the variables of interest. The remaining matrices in the equation below contain the variables of interest in 4x1 matrices.

$$\boldsymbol{M}\cdot\ddot{\boldsymbol{z}} + \boldsymbol{C}\cdot\dot{\boldsymbol{z}} + \boldsymbol{K}\cdot\boldsymbol{z} = 0 \qquad (5.13)$$

The matrices containing the variables of interest become:

$$\ddot{\boldsymbol{z}} = \begin{bmatrix}\ddot{z}_s\\ \ddot{\phi}\\ \ddot{z}_{uf}\\ \ddot{z}_{ur}\end{bmatrix} \qquad \dot{\boldsymbol{z}} = \begin{bmatrix}\dot{z}_s\\ \dot{\phi}\\ \dot{z}_{uf}\\ \dot{z}_{ur}\end{bmatrix} \qquad \boldsymbol{z} = \begin{bmatrix}z_s\\ \phi\\ z_{uf}\\ z_{ur}\end{bmatrix}$$

And the mass (**M**), damper (**C**) and spring (**K**) matrices according to equations become:

$$\boldsymbol{M} = \begin{bmatrix} m_s & 0 & 0 & 0\\ 0 & I_s & 0 & 0\\ 0 & 0 & m_{uf} & 0\\ 0 & 0 & 0 & m_{ur}\end{bmatrix}$$

$$\boldsymbol{C}=\begin{bmatrix} \left(-c_{sf}-c_{sr}\right) & \left(l_f\,c_{sf}-l_r\,c_{sr}\right) & c_{sf} & c_{sr} \\ \left(l_f\,c_{sf}-l_r\,c_{sr}\right) & \left(-l_f^2\,c_{sf}-l_r^2\,c_{sr}\right) & \left(-l_f\,c_{sf}\right) & \left(l_r\,c_{sr}\right) \\ c_{sf} & \left(-l_f\,c_{sf}\right) & \left(-c_{tf}-c_{sf}\right) & 0 \\ c_{sr} & \left(l_r\,c_{sr}\right) & 0 & \left(-c_{sr}-c_{tr}\right) \end{bmatrix}$$

$$\boldsymbol{K}=\begin{bmatrix} \left(-k_{sf}-k_{sr}\right) & \left(l_f\,k_{sf}-l_r\,k_{sr}\right) & k_{sf} & k_{sr} \\ \left(l_f\,k_{sf}-l_r\,k_{sr}\right) & \left(-l_f^2\,k_{sf}-l_r^2\,k_{sr}\right) & \left(-l_f\,k_{sf}\right) & \left(l_r\,k_{sr}\right) \\ k_{sf} & \left(-l_f\,k_{sf}\right) & \left(-k_{tf}-k_{sf}\right) & 0 \\ k_{sr} & \left(l_r\,k_{sr}\right) & 0 & \left(-k_{sr}-k_{tr}\right) \end{bmatrix}$$

According to Braun (2002), these three driving matrices for our model can be combined into one 8x8 dynamic matrix shown below:

$$Dynamic\,Matrix=\begin{bmatrix} -\boldsymbol{M}^{-1}\boldsymbol{C} & -\boldsymbol{M}^{-1}\boldsymbol{K} \\ \boldsymbol{I} & 0 \end{bmatrix}$$

Where $\boldsymbol{I}$ is a 4x4 identity matrix. The purpose of this can be seen when it is used within the following equation:

$$\begin{bmatrix} \ddot{z} \\ \dot{z} \end{bmatrix}=\begin{bmatrix} -\boldsymbol{M}^{-1}\boldsymbol{C} & -\boldsymbol{M}^{-1}\boldsymbol{K} \\ \boldsymbol{I} & 0 \end{bmatrix}\cdot\begin{bmatrix} \dot{z} \\ z \end{bmatrix} \tag{5.14}$$

Now according to Braun (2002) the eigenvalue solution of the dynamic matrix will yield four pairs of complex conjugates eigenvalues of the form:

$$s_i=\alpha_i\pm j\beta_i \tag{5.15}$$

From the values contained within the eigenvalue solutions three aspects of the suspension can be determined:

- Natural frequency of the *i*th mode:

$$\omega_{n_i}=\sqrt{\alpha_i^2+\beta_i^2} \tag{5.16}$$

- Damped natural frequency of the *i*th mode:

$$\omega_{d_i} = |\beta_i| \tag{5.17}$$

- Damping ratio of the *i*th mode:

$$\xi_i = \frac{-\alpha_i}{\omega_{n_i}} \tag{5.18}$$

Matlab was used for the process of creating the necessary matrices and performing the required calculations in order to return eigenvalue solutions for the suspension system of the Holden Commodore. The script required for these solutions can be found in Appendix A and the resulting eigenvalues are shown below.

$$eigenvalues = \begin{bmatrix} 69.4944 \\ 49.1739 \\ -69.4405 \\ -49.1467 \\ 6.4656 \\ 7.4664 \\ -6.4642 \\ -7.4646 \end{bmatrix}$$

These values don't fit with the textbook definition of the results. After an investigation and some fiddling the negative signs used to create the dynamic matrix were removed to give the following results:

$$eigenvalues = \begin{bmatrix} -0.03 + 170.18i \\ -0.03 - 170.18i \\ -0.01 + 120.34i \\ -0.01 - 120.34i \\ -0.0 + 6.93i \\ -0.0 - 6.93i \\ -0.0 + 8.00i \\ -0.0 - 8.00i \end{bmatrix}$$

This form of solution fits with the expected results spelled out by Braun (2002). The need for the modification to the dynamic matrix and the change in results is thought to be due to an error in the assumed direction at which the forces in the model were acting while the equations of motion were being developed. Despite this assumed error in the direction of forces for the equations of motion it appears this simple correction of the dynamic matrix has repaired the problem and these results can now be analysed.

The set of results for the eigenvalues contains eight complex conjugates while Braun (2002) states that the expected result should be only four sets of complex conjugates. For this reason the top four complex conjugates will be discarded due to the extraordinarily high frequency in comparison to the expected results outlined in section 5.2. Therefore taking the bottom four results and assuming they correspond to the vertical body motion, rotational body motion, front wheel vertical motion and rear wheel vertical motion respectively the following results for natural frequencies and damping ratio were found:

$$\omega_{n_{z_s}} = \omega_{d_{Z_s}} = \omega_{n_\phi} = \omega_{d_\phi} = 6.93$$

$$\xi_{z_s} = \xi_\phi = 0$$

$$\omega_{n_{z_{uf}}} = \omega_{d_{Z_{uf}}} = \omega_{n_{z_{ur}}} = \omega_{d_{z_{ur}}} = 8.00$$

$$\xi_{z_{uf}} = \xi_{z_{ur}} = 0$$

The results for the natural frequencies of the unsprung masses fall very close to the desired values. On the other hand the natural frequencies for the sprung mass in both its DOF's are much higher than anticipated. Also the damping ratio being found to be zero does not seem to be right. Despite these concerns over the accuracy of the results there is no could be no other results gained. The model was tweaked and changed but always returned the same results. It seems that the model is accurate according to Braun (2002) but doesn't seem to make terrific sense logically.

5.6 Forced Vibration Analysis

The second way in which the equations of 5.4 might be used in analysing the vibration of this suspension system is to use them in concurrence with a specific road function and analyse the vibration response. In order to do this analysis a road function defining z_{of} and z_{or} must be selected and this would then be substituted into equations (5.5) and (5.6). The by solving for the response of all critical points in the model, a visual of the vibration can be created.

A sinusoidal input will be simple to include in this formula and has been chosen as the shape of the road. The bumps in the road will be made 100mm top to bottom and the frequency of the bumps will be every two metres. The function defining this road with respect to time then becomes:

$$z_{of}(t) = A\sin(\omega t) \tag{5.18}$$

Where A is the amplitude in metres, ω is the angular velocity in radians per second and t is the time in seconds.

Assuming the car is travelling a 60km/h or 16.667m/s the values for the road function become:

$$A = 0.05m$$
$$\omega = 52.36\ rad/s$$

Therefore the road function for the front wheel becomes:

$$z_{of}(t) = 0.05\sin(52.36\,t) \tag{5.19}$$

Now the road function for the rear wheel must be delayed from the front wheel. This is simply a matter of adjusting the time input into the equation so as to delay the function by the time it takes the car to travel its wheelbase. The function for the rear wheel will be:

$$z_{or}(t) = A\sin\left(\omega(t - del)\right) \tag{5.20}$$

Where *del* is the time taken for the car to travel the length of its wheelbase. Therefore the delay needed for this equation is:

$$del = \left(l_f + l_r\right) \Big/ V = 2.7/16.667 = 0.162\ seconds \tag{5.21}$$

Now the equation for the road function at the rear wheel becomes:

$$z_{or}(t) = 0.05\sin\left(52.36(t - 0.162)\right) \tag{5.22}$$

The two equations for the road function at the front and rear wheel have been created. However these now both need differentiating to give the velocity of the vertical motion of the road as the car passes over the top. These velocities are also required to define the road movement in the model.

$$\dot{z}_{of}(t) = 2.168\cos(52.36\,t) \tag{5.23}$$

$$\dot{z}_{or}(t) = 2.168\cos\left(52.36(t - 0.162)\right) \tag{5.24}$$

This completes the definition of the road function for this simulation. In order to now run the simulation the equations (5.6) to (5.9) must be placed within a differential solver in Matlab. Using this software helps in developing accurate solutions to this model and also facilitates the conversion of raw data into a working image of the vehicle.

For this analysis the "ode113" function was used within Matlab as it is well suited to higher order differential equations. The equations developed in section 5.4 were modified to isolate the accelerations. By doing this these equations then became suitable for use inside this differential solver. Instead of only solving the differential equations and returning a set of graphs displaying the relative movements of the various components of the model, it was thought a good idea to create an image of the car in motion. For this reason the programming in Matlab existed within three distinct programs.

The first program "AnalysisDriver.m" is a driving program and is the only program required to be executed. The program begins by defining any variables needed within itself for later plotting results that are handed back to it from one of the other programs. It then defines a period of time for the simulation as well as the initial conditions for the model. Once these have been created it then calls to the solver "halfcarmodel.m" and gives it the time and initial conditions it has created. The solver creates any variables it needs, solves for the vibration response and then passes back the solution to the driving program. Once the driving program has received a solution back from the solver it then calls the third program "FourDOFplot.m" and hands it the solution. This program exists for the sole purpose of creating a visual picture of the vehicle motion. It pulls apart the results obtained from the solver and converts them into a full scale motion picture of the car (Figure 5.6.1) as it rides across the road. The simulation time is set to ten seconds and due to processing limitations the speed at which the visual can run is far less than real time. The visual will run until the exit button is pressed and following this control is passed back to the driving program where it finishes the cycle by plotting a series of graphs describing the motion of the various components of the car. These programs can be found in Appendix B.

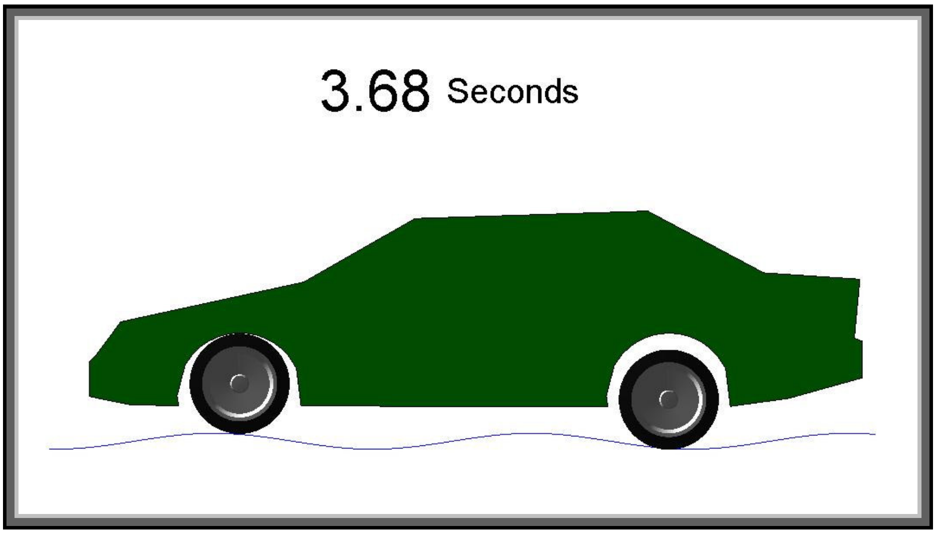

Figure 5.6.1 – Motion picture of car.

The computer simulation of this vehicle was intended to include the ability to have complete user entry of the vehicle specifications, road conditions and speed. However when programming the user interface it was difficult pass the variables between programs and added difficulty meant that a lack of time forced this option to be abandoned.

The results obtained from solving the model with the road function defined as in equations (5.19), (5.22), (5.23), and (5.24) are shown below in Figure 5.6.2 through to Figure 5.6.4. These results show a period at the beginning of the simulation where the results for velocity and acceleration are quite sporadic and then settle down after a few seconds. The initial position of the car has been set within the model so the wheels are sitting on the road at the start of the simulation. However the car is stationary when the analysis begins so the violent response in the velocity and acceleration at the beginning of the simulation are understandable due to the spike they must endure off the start. As the results show, these crazy results settle down as the dampers come into play and a more natural vibration can be seen.

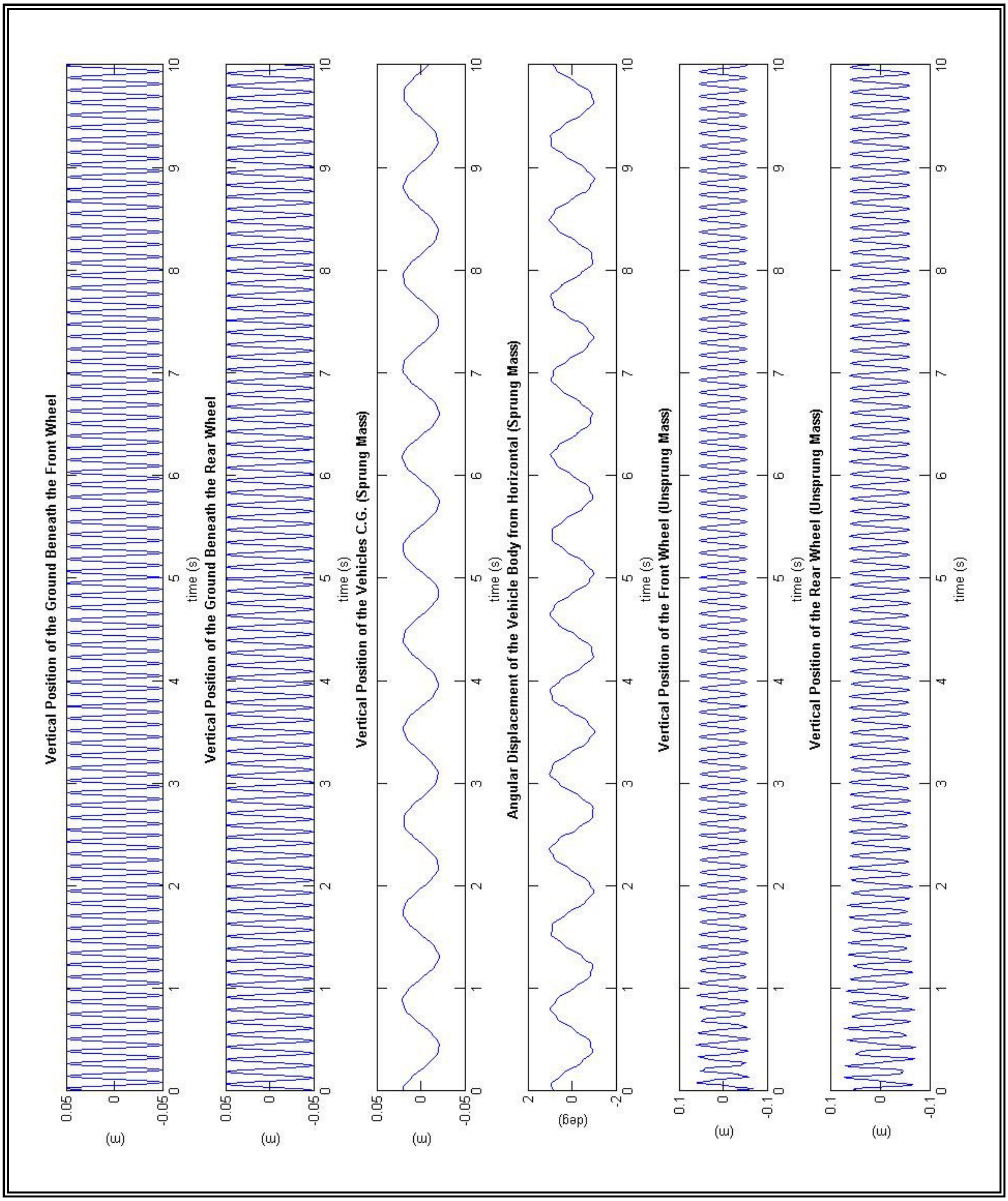

Figure 5.6.2 – Position results for forced vibration.

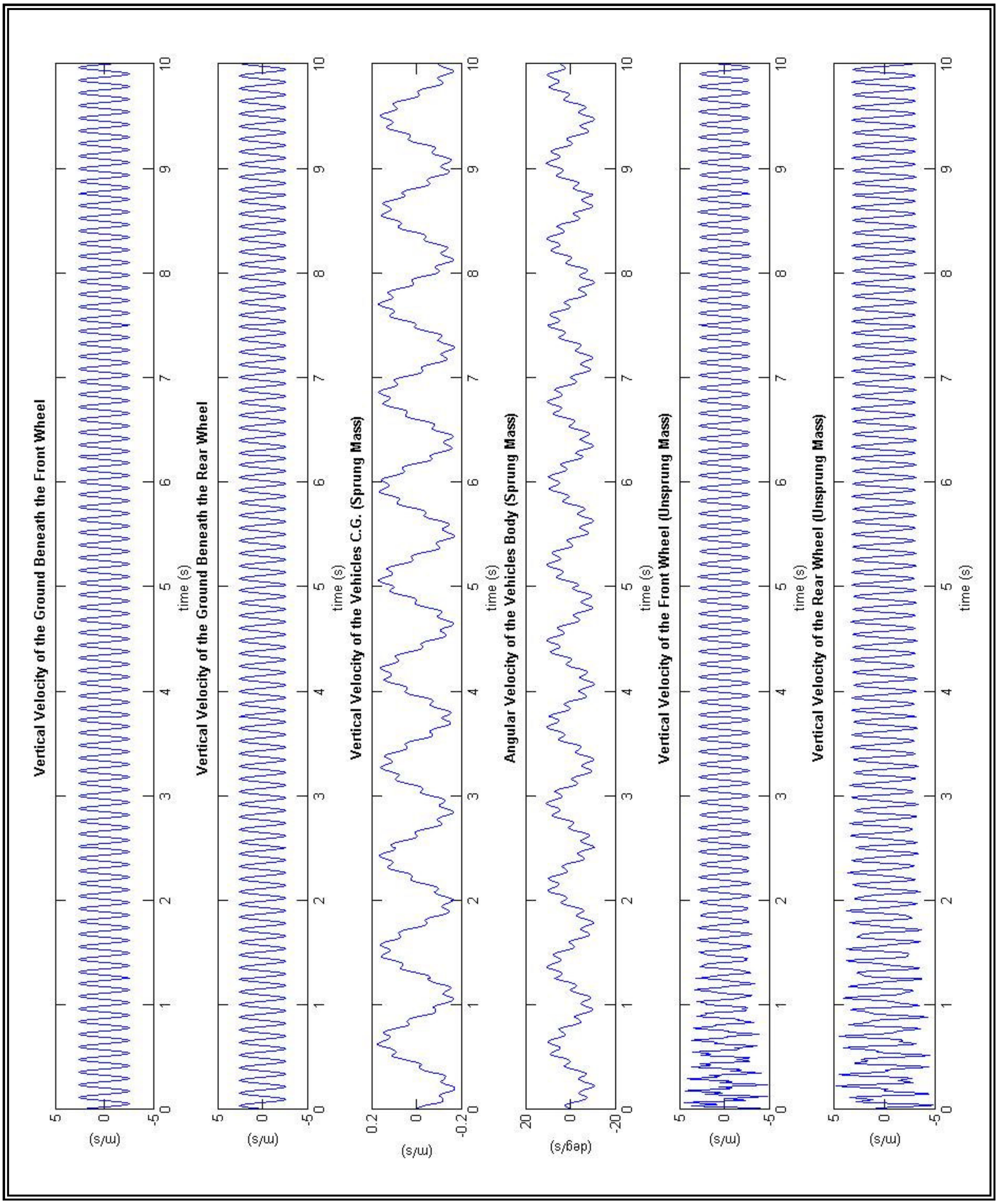

Figure 5.6.3 – Velocity results for forced vibration.

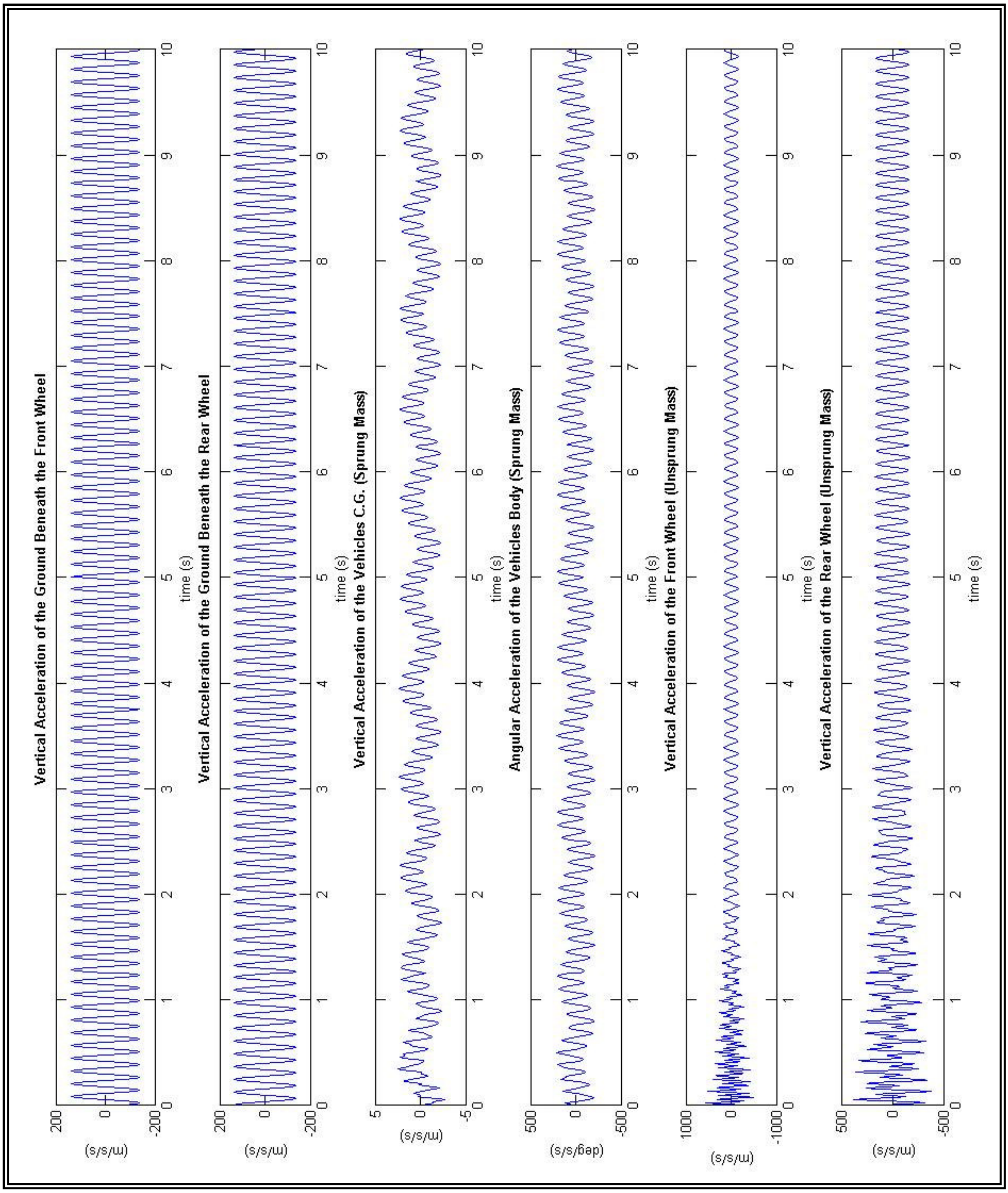

Figure 5.6.4 – Acceleration results for forced vibration.

These results provide a good insight into the vibration of a motor vehicle. The different trends in the position, velocity and acceleration along with the variation between components can be seen easily when comparing these plots. If this investigation had more time it would be possible to run multiple analyses on different road forms to see the vibration response in many more different situations.

While these results are interesting and seem relatively accurate, it is obvious that this model has a number of inconsistencies with the real world. Firstly the wheels are tied to the ground with a damper and spring. The result of this is a force that not only pushes the tire up off the ground but also pull the tire back down. In actual fact the model would be much more accurate if the tyre could be modelled with a spring that can only act when the tyre is in contact with the ground. Achieving this goes outside the scope for this project but is entirely possible with some clever programming in Matlab. Another problem with this model is the damping constant of the tyre not being known. If an accurate approximation for this constant could be found then the accuracy of this model could be given more weight.

This model could be improved in a number of ways. More user interface and automation could allow quick adjustments to road and vehicle properties. A capacity for acceleration and deceleration would also expand the model into a more realistic realm. Ultimately the model could be extended to a seven DOF system and account for all components within the vehicle. If all these improvements could be made then the program would become a useful tool in determining spring and damping rates to suit specific and varied road conditions.

5.7 Conclusions: Chapter 5

A half-car model has been created successfully and used in analysing forced and free vibration for a system of suspension belonging to a Holden Commodore. Natural frequencies were determined but contain unknown error and forced vibration response has been plotted and animated. The model developed in this chapter lays the framework for extending the vehicle model right through into a full-car, seven DOF system. Such a system would be capable of taking into consideration all aspects affecting a cars ride and handling. Within the scope of this investigation the model could not be extended past this point. Furthermore time constraints meant that multiple analyses could not be completed using this model. Only a single sinusoidal forced vibration analysis was conducted. Given more time the model can be set up to facilitate quick adjustments to vehicle and road properties. This model is certainly not perfect and cannot be claimed to mimic real life but it does provide an interesting visual to contemplate vibration response within suspension systems.

CHAPTER 6

CONCLUSIONS

6.1 Introduction

This investigation has surveyed a full overview of passenger vehicle suspension. In an attempt to develop a new improved method of height adjustable suspension this project has covered the entirety of suspension from its inception right through to modern day systems, manufacture and testing.

6.2 Discussion

This investigation has succeeded in developing a new concept for a height adjustable spring. The concept relies on pressurised air for its operation. Unlike existing air springs that function by compressing the air inside them in order to increase pressure and the resulting reaction force. This new system seeks to maintain a relatively constant pressure and adjust the area over which it acts in order to achieve an increase in opposing force as the spring is compressed. This design more closely resembles the characteristics of a genuine steel spring which has a linear relationship linking force and deflection. While a physical model of the spring was not able to be produced in this project, the theory of the design has been established and stands to be further explored.

This new concept provides a fresh avenue for exploring air springs. Its design does not limit its height and its diameter could be infinitely variable. Problems have always existed in manufacturing air spring components that are free of leaks but with this design the reservoir uses no sliding seals and can be easily kept air tight. The design of the ribbing creates a perfect situation for preventing sediment build up. The steep angles of the sides prevent any deposits that could cause wear.

The new spring has been assessed and developed for manufacture also. A process of selecting materials and manufacturing processes was conducted for the new concept. Where needed the design was rethought to ensure that manufacture and assembly was possible and that the life span of the components would be adequate for the task at hand.

A spring or system of suspension must be proven through a process of vibration analysis. This investigation saw the development of a half-car model for observing the vibration response of a system of suspension. The model was used to determine the response of an existing suspension on a Holden Commodore. Time constraints prevented the use of the model to critically appraise the new height adjustable system. However given that the model created can only take into account constant spring rates, the vibration analysis of the new suspension fitted to the commodore would have been no different to that of the standard suspension. Changing the model from one type of spring to another could only be done in the half-car model by changing the spring constant. With the new spring however it would be designed to match the spring constant of the standard spring underneath the Commodore at the standard ride height. If time had allowed it would have been interesting to assess the effectiveness of then raising the car in the model using the height adjustable spring and then checking its continued effectiveness at different heights. With air springs the best model for simulating the vibration response would be one that can account for a spring with a varying rate. It is a luxurious assumption to believe that the spring rate will remain constant over the full length of its travel. The model being used limits effective comparisons between springs due to its inability to take a variable rate spring.

Despite the model not being used to analyse the new height adjustable spring it has been a worthwhile process in developing a set of programs that can simulate a cars vibration response. Graphs and an animation have been created of the car undergoing a sinusoidal forced vibration on the road. Admittedly the model has inaccuracies that could not be removed and hence the results are not concurrent with a real life situation. However despite some inaccuracies this model lays the foundation for further expansion into vibration analysis and response.

6.3 Further Research and Recommendations

The scope of this project did not allow an in depth analysis on the new system of suspension. Complex programs could in the future be employed to run full FEA analysis on components to ensure their strength and viability. On top of this a more accurate vibration model should be employed to assess the air bags success. Depending on cost it may be more beneficial to construct a prototype for physical testing of the vibration response. Accurately modelling the air bag with all its components interacting may require too much effort and hence place a prototype, however costly, as a more desirable option.

Accurate estimations of natural frequencies induced in a car with the new air bag should be determined. Again the produce this accurate estimation a physical model my be required.

Assuming the air bag can be prototyped the next step would be to put it into service for a period of time to assess its effectiveness in a real life situation.

The model created and used within this project should be further developed so that the tyres are no longer tied down to the road. A more accurate approximation to a cars vibration would be possible if the tyre were able to separate itself from the road.

6.4 Summary of Chapter 6

This investigation into a new method of height adjustable suspension has identified new ways of suspending a car. The new method has not however been put through adequate testing at this stage to affirm its absolute success. The theory of the design is promising but a proper vibration analysis is needed to completely confirm the design. This investigation has covered a broad spectrum of aspects associated with suspension and its design and use. The principles, models and theories used within this investigation hold great potential for further development of this new idea for varying a vehicles ride height.

APPENDIX A

University of Southern Queensland
Faculty of Engineering and Surveying

ENG 4111/2 Research Project
PROJECT SPECIFICATION

FOR: **JOSHUA WALTON**

TOPIC: **Investigation of a variable ride height suspension for an automobile**

SUPERVISOR: Dr. Jayantha Ananda Epaarachchi

ENROLMENT: ENG 4111 – S1, 2007
ENG 4112 – S2, 2007

PROJECT AIM: This project aims to model and analyse a variable height suspension system for a passenger car. The project intends to undertake detailed static and dynamic analysis of a variable height suspension system, model the suspension system and propose a complete design for a particular automobile.

PROGRAMME: **Issue A, 20th March 2007**

1. Research the development of suspension technology including but not confined to traditional coil springs, air springs, shock absorbers and adjustable configurations.

2. Analyse and critically evaluate existing means of height adjustable suspension and how it succeeds or fails to meet the project aim.

3. Research legal and physical constraints on the design of a new suspension assembly.

4. Consider and document alternative methods for changing a car’s ride height.

5. Design a suspension assembly that can function in every role of existing suspensions as well as allowing height and handling variation tied together in the one adjustment.

6. Static and Dynamic Analysis of the proposed system.

7. Design and 3D modelling of the final design

8. Implementation and costing

As time permits:

9. Construct prototype assembly and conduct testing to evaluate its behaviour relative to the theoretical testing.

AGREED: ____________________(student)
(supervisor)

(dated) ____/_____/_______.

APPENDIX B

```
%----------------------------------------------------------------------%
% ----------------------freevibrationsolution.M------------------------- %
%----------------------------------------------------------------------%
%
% PROJECT: Investigation of a Variable Ride Height Suspension for an
% Automobile

% FILE NAME: freevibrationsolution.m
%
% Input Variables:  Nil
% Output Variables: eigen
%
% This program determines the eigenvalue solutions for a vibration analysis
% on a four DOF half-car model.  The program defines all the required data
% for the car and produces mass, damper and spring matrices for the
% equation of motion of the vehicle.  Once these matrices are found it
% solves for the eigenvalue solutions of a dynamic matrix made up of all
% three of the matrices needed for the vehicles equation of motion.  The
% eigenvalue solutions are displayed in the matlab window.
%
% Written by Joshua Walton - 0050026275
% Date of last revision: 20th October 2008
%
%---------------------------------------------------------------------%
%---------------------------------------------------------------------%

%Define the spring constant for the front spring connecting the sprung mass
%to the unsprung mass
    ksf = 21600; %N/m

%Define the spring constant for the rear spring connecting the sprung mass
%to the unsprung mass
    ksr = 21600; %N/m

%Define the spring constant for the front tyre as it comes in contact with
%the road
    %ktf = 100*ksf; %N/m
    ktf = 847200; %N/m

%Define the spring constant for the rear tyre as it comes in contact with
%the road
    %ktr = 100*ksr; %N/m
    ktr = 847200; %N/m

%Define the damping constant for the front shock absorber connecting the
%sprung and unsprung mass
    csf = 0.8; %N/(m/s)

%Define the damping constant for the rear shock absorber connecting the
%sprung and unsprung mass
    csr = 0.8; %N/(m/s)

%Define the damping constant for the front shock absorber connecting the
%tire to the road
    ctf = 0.8; %N/(m/s)

%Define the damping constant for the rear shock absorber connecting the
%tire to the road
    ctr = 0.8; %N/(m/s)

%Define the unsprung mass for the front and back of the car
    muf = 30;  %Front unsprung mass
    mur = 60;  %Rear unsprung mass

%Define the weight and dimensions of the car
    mfr = 421;
    mrr = 451;
    l = 2.7;
    lf = 1.4;
    lr = 1.3;
```

```
%Define the total sprung mass
    ms = mfr + mrr;

%Define a height for the car where the centre of gravity would be located
%at half this distance from the ground
    h = 1;

%Define the moment of inertia for the pitching rotation of the car
    Is=(1/12)*((2*mfr)+(2*mrr))*((l.^2)+(h.^2));

%Create the mass, damper and spring matrices
    M = [ms 0 0 0;0 Is 0 0;0 0 muf 0;0 0 0 mur];

    C = [(-csf-csr), ((lf*csf)-(lr*csr)), csf, csr;
        ((lf*csf)-(lr*csr)),((-(lf.^2)*csf)-((lr.^2)*csr)),(-lf*csf),(lr*csr);
        csf, (-lf*csf), (-ctf-csf), 0;
        csr, lr*csr, 0, (-csr-ctr)];

    K = [(-ksf-ksr), ((lf*ksf)-(lr*ksr)), ksf, ksr;
        ((lf*ksf)-(lr*ksr)),((-(lf.^2)*ksf)-((lr.^2)*ksr)),(-lf*ksf),(lr*ksr);
        ksf, (-lf*ksf), (-ktf-ksf), 0;
        ksr, lr*ksr, 0, (-ksr-ktr)];

%Create identity and zero matrices for use in creating the dynamic matrix
    I = [1,0,0,0;
        0,1,0,0;
        0,0,1,0;
        0,0,0,1];

    Zero = [0,0,0,0;
        0,0,0,0;
        0,0,0,0;
        0,0,0,0];

%Create the dynamic matrix
    DYN = [(inv(M)*C),(inv(M)*K);
        I, Zero];

%Determine the eigenvalues of the dynamic matrix
    eigen = eig(DYN)

%--------------------------------------------------------------------------%
%----------------------------END OF PROGRAM--------------------------------%
%--------------------------------------------------------------------------%
```

APPENDIX C

```
%-----------------------------------------------------------------------%
% -------------------------AnalysisDriver.M---------------------------- %
%-----------------------------------------------------------------------%
%
% PROJECT: Investigation of a Variable Ride Height Suspension for an
% Automobile

% FILE NAME: AnalysisDriver.m
%
% Input Variables:  Nil
% Output Variables: Nil
%
% Driving File for differential analysis of 4 degree of freedom system
% modelling the suspension on an automobile.  This program defines values
% and calls both a differential solver and a drawing program before
% dispaying the solution data in graph form as it closes.
%
% Written by Joshua Walton - 0050026275
% Date of last revision: 20th October 2008
%
%----------------------------------------------------------------------%
%----------------------------------------------------------------------%
%Define the wheelbase of the car
    l = 2.7;

%Define the speed of the car;
    Vkm = 60;  %speed in km/h
    V = (60*1000)/3600;

%Create the amplitude value for the road function
    global F0
    F0 = 0.05;

%Create the angular velocity for the road function
    lambda = 2;  %Wavelength of oscillation
    w = ((V/lambda)*2*pi);  %angular velocity of oscillation

%Create the delay value for the road function
    del = 2.7/V;

%----------------------------------------------------------------------%
%----------------------------------------------------------------------%
%Define the time matrix for the analysis
    tspan = [0:0.01:10];

%Determine the initial position of the rear wheel accounting for the delay
%in the road function
    zurinitial = F0*sin(w*del);
    zsinitial = zurinitial/2;
    phiinitial = tan(zurinitial/l);

%Define initial conditions matrix
    y0 = [zsinitial;0;phiinitial;0;0;0;zurinitial;0];

%----------------------------------------------------------------------%
%----------------------------------------------------------------------%
%Call ODE function to solve the differential function
    [t,y] = ode113('halfcarmodel',tspan,y0);

%----------------------------------------------------------------------%
%----------------------------------------------------------------------%
%Call function to run animation of the vibration simulation
    fin = FourDOFplot(t,y)

%----------------------------------------------------------------------%
%----------------------------------------------------------------------%
%Solve road functions for the purpose of creating graphs
    zof = F0*sin(w*t);
    zor = F0*sin(w*(t-del));
    zofdot = F0*w*cos(w*t);
    zordot = F0*w*cos(w*(t-del));
```

```
    zof2dot = -F0*w*w*sin(w*t);
    zor2dot = -F0*w*w*sin(w*(t-del));

%Convert the angle of the car into degrees for the purpose of creating
%graphs
    angle = (y(:,3)*180)/pi;
    angledot = (y(:,4)*180)/pi;

% Change the figure window to encompass the whole screen
    scrsz = get(0,'ScreenSize');
    set(gcf,'Position',scrsz,'color', [0.992157 0.917647 0.796078],...
        'Name','Position Results')

%Plot the results relating to the position of the model
    subplot (611);
    plot(t,zof);
    xlabel('time (s)');
    ylabel('(m)');
    title('Vertical Position of the Ground Beneath the Front Wheel',...
        'FontWeight','Bold');

    subplot (612);
    plot(t,zor);
    xlabel('time (s)');
    ylabel('(m)');
    title('Vertical Position of the Ground Beneath the Rear Wheel',...
        'FontWeight','Bold');

    subplot (613);
    plot(t,y(:,1));
    xlabel('time (s)');
    ylabel('(m)');
    title('Vertical Position of the Vehicles C.G. (Sprung Mass)',...
        'FontWeight','Bold');

    subplot (614);
    plot(t,angle);
    xlabel('time (s)');
    ylabel('(deg)');
    title('Angular Displacement of the Vehicle Body from Horizontal (Sprung
Mass)',...
        'FontWeight','Bold');

    subplot (615);
    plot(t,y(:,5));
    xlabel('time (s)');
    ylabel('(m)');
    title('Vertical Position of the Front Wheel (Unsprung Mass)',...
        'FontWeight','Bold');

    subplot (616);
    plot(t,y(:,7));
    xlabel('time (s)');
    ylabel('(m)');
    title('Vertical Position of the Rear Wheel (Unsprung Mass)',...
        'FontWeight','Bold');

% New figure window
% Plot the results relating to the velocity of the model.
    hfig = figure;
    set(gcf,'Position',scrsz,'color', [0.992157 0.917647 0.796078],...
        'Name','Velocity Results');

    subplot (611);
    plot(t,zofdot);
    xlabel('time (s)');
    ylabel('(m/s)');
    title('Vertical Velocity of the Ground Beneath the Front Wheel',...
        'FontWeight','Bold');

    subplot (612);
    plot(t,zordot);
    xlabel('time (s)');
    ylabel('(m/s)');
    title('Vertical Velocity of the Ground Beneath the Rear Wheel',...
        'FontWeight','Bold');
```

```
    subplot (613);
    plot(t,y(:,2));
    xlabel('time (s)');
    ylabel('(m/s)');
    title('Vertical Velocity of the Vehicles C.G. (Sprung Mass)',...
        'FontWeight','Bold');

    subplot (614);
    plot(t,angledot);
    xlabel('time (s)');
    ylabel('(deg/s)');
    title('Angular Velocity of the Vehicles Body (Sprung Mass)',...
        'FontWeight','Bold');

    subplot (615);
    plot(t,y(:,6));
    xlabel('time (s)');
    ylabel('(m/s)');
    title('Vertical Velocity of the Front Wheel (Unsprung Mass)',...
        'FontWeight','Bold');

    subplot (616);
    plot(t,y(:,8));
    xlabel('time (s)');
    ylabel('(m/s)');
    title('Vertical Velocity of the Rear Wheel (Unsprung Mass)',...
        'FontWeight','Bold');

%-----------------------------------------------------------------------%
%-----------------------------------------------------------------------%
%Repeat the process used in the half car model to determine the
%accelerations of the model

%-----------------------------------------------------------------------%
%-----------------------------------------------------------------------%
%Define the spring constant for the front spring connecting the sprung mass
%to the unsprung mass
    ksf = 21600; %N/m

%Define the spring constant for the rear spring connecting the sprung mass
%to the unsprung mass
    ksr = 21600; %N/m

%Define the spring constant for the front tyre as it comes in contact with
%the road
    ktf = 847233; %N/m

%Define the spring constant for the rear tyre as it comes in contact with
%the road
    ktr = 847233; %N/m

%Define the damping constant for the front shock absorber connecting the
%sprung and unsprung mass
    csf = 0.8; %N/(m/s)

%Define the damping constant for the rear shock absorber connecting the
%sprung and unsprung mass
    csr = 0.8; %N/(m/s)

%Define the damping constant for the front shock absorber connecting the
%tire to the road
    ctf = 80; %N/(m/s)

%Define the damping constant for the rear shock absorber connecting the
%tire to the road
    ctr = 80; %N/(m/s)

%Define the unsprung mass for the front and back of the car
    muf = 30;  %Front unsprung mass
    mur = 60;  %Rear unsprung mass

%Define the dimensions of the car
    mfr = 421;
    mrr = 451;
    l = 2.7;
```

```
    lf = 1.4;
    lr = 1.3;

%Define the total sprung mass
    ms = mfr + mrr;

%Define a height for the car where the centre of gravity would be located
%at half this distance from the ground
    h = 1;

%Define the moment of inertia for the pitching rotation of the car
    Is=(1/12)*((2*mfr)+(2*mrr))*((l.^2)+(h.^2));

%Calculate the accelerations for the model
    zs2dot = (-(((csf+csr).*y(:,2))/ms))-((((lr*csf)...
        -(lf*csf)).*y(:,4))/ms)+((csf.*y(:,6))/ms)+((csr.*y(:,8))/ms)...
        -(((ksf+ksr).*y(:,1))/ms)-((((lr*ksr)-(lf*ksf)).*y(:,3))/ms)...
        +((ksf.*y(:,5))/ms)+((ksr.*y(:,7))/ms);

    theta2dot = (-((((lr*csr)-(lf*csf)).*y(:,2))/Is))...
        -(((((lr.^2)*csr)+((lf.^2)*csf)).*y(:,4))/Is)...
        -((lf*csf.*y(:,6))/Is)+((lr*csr.*y(:,8))/Is)...
        -((((lr*ksr)-(lf*ksf)).*y(:,1))/Is)...
        -(((((lr.^2)*ksr)+((lf.^2)*ksf)).*y(:,3))/Is)...
        -((lf*ksf.*y(:,5))/Is)+((lr*ksr.*y(:,7))/Is);

    zuf2dot = ((ktf*F0*sin(w*-t))/muf)-((ctf*F0*w*cos(w*-t))/muf)...
        +((csf.*y(:,2))/muf)-((lf*csf.*y(:,4))/muf)...
        -(((csf+ctf).*y(:,6))/muf)+((ksf.*y(:,1))/muf)...
        -((lf*ksf.*y(:,3))/muf)-(((ksf+ktf).*y(:,5))/muf);

    zur2dot = ((ktr*F0*sin(w*(-t+del)))/mur)...
        -((ctr*F0*w*cos(w*(-t+del)))/mur)+((csr.*y(:,2))/mur)...
        +((lr*csr.*y(:,4))/mur)-(((csr+ctr).*y(:,8))/mur)...
        +((ksr.*y(:,1))/mur)+((lr*ksr.*y(:,3))/mur)...
        -(((ksr+ktr).*y(:,7))/mur);

%Convert angle into degrees for the purpose of graphing
    angle2dot = (theta2dot*180)/pi;

% New figure window
% Plot the results relating to the Acceleration of the model.
    hfig = figure;
    set(gcf,'Position',scrsz,'color', [0.992157 0.917647 0.796078],...
         'Name','Acceleration Results');

    subplot (611);
    plot(t,zof2dot);
    xlabel('time (s)');
    ylabel('(m/s/s)');
    title('Vertical Acceleration of the Ground Beneath the Front Wheel',...
        'FontWeight','Bold');

    subplot (612);
    plot(t,zor2dot);
    xlabel('time (s)');
    ylabel('(m/s/s)');
    title('Vertical Acceleration of the Ground Beneath the Rear Wheel',...
        'FontWeight','Bold');

    subplot (613);
    plot(t,zs2dot);
    xlabel('time (s)');
    ylabel('(m/s/s)');
    title('Vertical Acceleration of the Vehicles C.G. (Sprung Mass)',...
        'FontWeight','Bold');

    subplot (614);
    plot(t,angle2dot);
    xlabel('time (s)');
    ylabel('(deg/s/s)');
    title('Angular Acceleration of the Vehicles Body (Sprung Mass)',...
        'FontWeight','Bold');

    subplot (615);
    plot(t,zuf2dot);
```

```
    xlabel('time (s)');
    ylabel('(m/s/s)');
    title('Vertical Acceleration of the Front Wheel (Unsprung Mass)',...
        'FontWeight','Bold');

    subplot (616);
    plot(t,zur2dot);
    xlabel('time (s)');
    ylabel('(m/s/s)');
    title('Vertical Acceleration of the Rear Wheel (Unsprung Mass)',...
        'FontWeight','Bold');

%-------------------------------------------------------------------------%
%Clear everything
    clear;
    clc;

%-------------------------------------------------------------------------%
%-----------------------------END OF PROGRAM------------------------------%
%-------------------------------------------------------------------------%
```

APPENDIX D

```
%-----------------------------------------------------------------------%
% ---------------------------halfcarmodel.M------------------------------ %
%-----------------------------------------------------------------------%
%
% PROJECT: Investigation of a Variable Ride Height Suspension for an
% Automobile

% FILE NAME: halfcarmodel.m
%
% Input Variables:  t,y
% Output Variables: xdot
%
% This is a differential solver for a half-car dynamic analysis of
% vibration.  This function recieves a period of time and intitial
% conditions and solves for the vibration response of a car.
%
% Written by Joshua Walton - 0050026275
% Date of last revision: 20th October 2008
%
%----------------------------------------------------------------------%
%----------------------------------------------------------------------%
%Function to differentiate the car model%
    function xdot = halfcarmodel(t,y);

%----------------------------------------------------------------------%
%Define the spring constant for the front spring connecting the sprung mass
%to the unsprung mass
    ksf = 21600; %N/m

%Define the spring constant for the rear spring connecting the sprung mass
%to the unsprung mass
    ksr = 21600*1.1; %N/m

%Define the spring constant for the front tyre as it comes in contact with
%the road
    ktf = 847233; %N/m

%Define the spring constant for the rear tyre as it comes in contact with
%the road
    ktr = 847233; %N/m

%Define the damping constant for the front shock absorber connecting the
%sprung and unsprung mass
    csf = 0.8; %N/(m/s)

%Define the damping constant for the rear shock absorber connecting the
%sprung and unsprung mass
    csr = 0.8; %N/(m/s)

%Define the damping constant for the front shock absorber connecting the
%tire to the road
    ctf = 80; %N/(m/s)

%Define the damping constant for the rear shock absorber connecting the
%tire to the road
    ctr = 80; %N/(m/s)

%Define the unsprung mass for the front and back of the car
    muf = 30;  %Front unsprung mass
    mur = 60;  %Rear unsprung mass

%Define the dimensions of the car
    mfr = 421;
    mrr = 451;
    l = 2.7;
    lf = 1.4;
    lr = 1.3;

%Define the total sprung mass
    ms = mfr + mrr;
```

```
%Define a height for the car where the centre of gravity would be located
%at half this distance from the ground
    h = 1;

%Define the moment of inertia for the pitching rotation of the car
    Is=(1/12)*((2*mfr)+(2*mrr))*((l.^2)+(h.^2));

%Define the speed of the car;
    Vkm = 60;  %speed in km/h
    V = (60*1000)/3600; %speed in m/s

%Create the amplitude value for the road function
    F0 = 0.05;

%Create the frequency value for the road function
    lambda = 2;  %Wavelength of oscillation
    w = ((V/lambda)*2*pi);  %Angular velocity

%Create the delay value for the road function
    del = 2.7/V;

%-----------------------------------------------------------------------%

%Create the results matrix for the differential solver
    xdot = zeros(8,1);

%Define the functions driving the derivative
    xdot(1) = y(2);
    xdot(2) = (-(((csf+csr)*y(2))/ms))-((((lr*csf)-(lf*csf))*y(4))/ms)...
        +((csf*y(6))/ms)+((csr*y(8))/ms)-(((ksf+ksr)*y(1))/ms)...
        -((((lr*ksr)-(lf*ksf))*y(3))/ms)+((ksf*y(5))/ms)+((ksr*y(7))/ms);
    xdot(3) = y(4);
    xdot(4) = (-((((lr*csr)-(lf*csf))*y(2))/Is))...
        -(((((lr.^2)*csr)+((lf.^2)*csf))*y(4))/Is)-((lf*csf*y(6))/Is)...
        +((lr*csr*y(8))/Is)-((((lr*ksr)-(lf*ksf))*y(1))/Is)...
        -(((((lr.^2)*ksr)+((lf.^2)*ksf))*y(3))/Is)-((lf*ksf*y(5))/Is)...
        +((lr*ksr*y(7))/Is);
    xdot(5) = y(6);
    xdot(6) = ((ktf*F0*sin(w*-t))/muf)-((ctf*F0*w*cos(w*-t))/muf)...
        +((csf*y(2))/muf)-((lf*csf*y(4))/muf)-(((csf+ctf)*y(6))/muf)...
        +((ksf*y(1))/muf)-((lf*ksf*y(3))/muf)-(((ksf+ktf)*y(5))/muf);
    xdot(7) = y(8);
    xdot(8) = ((ktr*F0*sin(w*(-t+del)))/mur)...
        -((ctr*F0*w*cos(w*(-t+del)))/mur)+((csr*y(2))/mur)...
        +((lr*csr*y(4))/mur)-(((csr+ctr)*y(8))/mur)+((ksr*y(1))/mur)...
        +((lr*ksr*y(3))/mur)-(((ksr+ktr)*y(7))/mur);

%------------------------------------------------------------------------%
%-----------------------------END OF PROGRAM------------------------------%
%------------------------------------------------------------------------%
```

APPENDIX E

```
%----------------------------------------------------------------------%
% ---------------------------FourDOFplot.M------------------------------ %
%----------------------------------------------------------------------%
%
% PROJECT: Investigation of a Variable Ride Height Suspension for an
% Automobile

% FILE NAME: FourDOFplot.m
%
% Input Variables:  t,y
% Output Variables: fin
%
% This is a program to animate a cars vibration response to a given road
% input.  This program uses data from a half-car model and uses it to
% create a visualisation of the car as it travels over the road.  This is
% only a side on view of the car even though the plotting is done in 3D.
% The visualisation has been created so that if the model was changed to
% a full-car model then the program could be modified to display a 3D
% image of the vehicles response.
%
% Written by Joshua Walton - 0050026275
% Date of last revision: 20th October 2008
%
%---------------------------------------------------------------------%
%---------------------------------------------------------------------%
%Function called by the analysis driver
    function [fin] = FourDOFplot(t,y)

%---------------------------------------------------------------------%
%---------------------------------------------------------------------%
%Create the coordinate vectors for the side profile picture of the 4DOF
%vehicle simulation
%---------------------------------------------------------------------%
%Creating the Tyre Tread Cylinders
    set=[-1:0.1:1];
    set1=(sqrt(8-set.^2)/30)+315;
    r=[236, 315, set1(1,:), 315,236];
    [xt,yt,zt]=cylinder(r,100);
    zt=zt*240;

%Creating the Tyre Rim Cylinders
    set2=[212:-0.5:189];
    set3=[189:0.5:212];
    r2=[236,212,set2(1,:),0,0,0,set3(1,:),212,236];
    [xr,yr,zr]=cylinder(r2,100);
    zr=zr*240;

%Creating the tyre Hub Cylinders
    set4=78*ones(40);
    set5=[78:-0.5:50];
    set6=[50:0.5:78];
    r3=[0,set6(1,:),set4(1,:),set5(1,:),0];
    [xh,yh,zh]=cylinder(r3,100);
    zh=(zh*120)+30;

%Front Right Wheel Tread
    xt1=zt-120;
    yt1=yt-2;
    zt1=xt+315;

%Front Right Wheel Rim
    xr1=zr-120;
    yr1=yr-2;
    zr1=xr+315;

%Front Right Wheel Hub
    xh1=zh-120;
    yh1=yh-2;
    zh1=xh+315;
```

```
%Rear Right Wheel Tread
    xt3=xt1;
    yt3=yt1+2700;
    zt3=zt1;

%Rear Right Wheel Rim
    xr3=xr1;
    yr3=yr1+2700;
    zr3=zr1;

%Rear Right Wheel Hub
    xh3=xh1;
    yh3=yh1+2700;
    zh3=zh1;

%Create the point of the centre of gravity for the car
    l=2700;
    h=1000;
    cgx = 0;
    cgy = l/2;
    cgz = h/2;

%-----------------------------------------------------------------------%
%-----------------------------------------------------------------------%
%Create figure window the size of the screen ready for the simulation
    scrsz = get(0,'ScreenSize');
    figure('Position',scrsz,'MenuBar','none','color', [1 1 1]);

%-----------------------------------------------------------------------%
% Create the variable m for use with the exit control
    global m;
    m=0;

% Create increment to distinguish when the simulation has reached the end
    tinc = 1;

% Create final increment number to be used as a flag when the time
% increment reaches that value.
    incf = length(t);

% Exit button for terminating the animation
    uicontrol('String','Exit','Units','Normalized',...
        'Position',[0.05 0.05 0.1 0.06],'FontWeight','bold',...
        'Tag','pushbutton1','Callback','global m,m=1;');

%Determine the points on the profile of the car body relative to the
%position of the centre of gravity and pitch of the car
    archy = [-395:50:395];
    archang = acos(archy/395);
    archz = (395*sin(archang))+285;
    bodyy = [-385,archy,385,2315,(archy+2700),3085,3450,3915,3915,...
        3865,3900,3295,2565,1100,400,-750,-906,-946,-946,-700,-385];
    bodyz = [220,archz,220,220,archz,220,270,400,630,650,1020,1060,...
        1448,1400,1000,750,540,500,280,230,220];
    bodyhyp = sqrt(((bodyy-cgy).^2)+((bodyz-cgz).^2));
    plength = length(bodyhyp);
    bodyx = zeros(plength,1);

%Check the quadrant of the body point and calculate the angle accordingly
    poscheck = 1;

    while poscheck <= plength
        if (bodyy(1,poscheck)-cgy) >= 0
            if (bodyz(1,poscheck)-cgz) >= 0
                bodyang(1,poscheck) = atan((bodyz(1,poscheck)-cgz)./...
                    (bodyy(1,poscheck)-cgy));
            end
        end

        if (bodyy(1,poscheck)-cgy) <= 0
            if (bodyz(1,poscheck)-cgz) >= 0
                bodyang(1,poscheck) = (atan((bodyz(1,poscheck)-cgz)./...
                    (bodyy(1,poscheck)-cgy)))+pi;
            end
        end
```

```
        if (bodyy(1,poscheck)-cgy) <= 0
            if (bodyz(1,poscheck)-cgz) <= 0
                bodyang(1,poscheck) = (atan((bodyz(1,poscheck)-cgz)./...
                    (bodyy(1,poscheck)-cgy)))+pi;
            end
        end

        if (bodyy(1,poscheck)-cgy) >= 0
            if (bodyz(1,poscheck)-cgz) <= 0
                bodyang(1,poscheck) = (atan((bodyz(1,poscheck)-cgz)./...
                    (bodyy(1,poscheck)-cgy)))+(2*pi);
            end
        end
        poscheck = poscheck + 1;
    end

%-----------------------------------------------------------------------%
%-----------------------------------------------------------------------%
%Create variables to be used in plotting an image of the road

%Define the speed of the car;
    Vkm = 60;  %speed in km/h
    V = (60*1000)/3600; %speed in m/s

%Create the amplitude value for the road function
    F0 = 0.05;

%Create the frequency value for the road function
    lambda = 2;  %Wavelength of oscillation
    w = V/lambda;  %Frequency of oscillation

%Create the delay value for the road function
    del = 2.7/V;

%-----------------------------------------------------------------------%
%-----------------------------------------------------------------------%
%Begin Simulation Loop until interrupted
    while m==0;

%Find the current body profile using the angular offset supplied from the
%differential solution
        zsy = (bodyhyp.*cos(bodyang+y(tinc,3)))+cgy;
        zsz = (bodyhyp.*sin(bodyang+y(tinc,3)))+cgz+(y(tinc,1));

%Find the current positions of the wheels
        zuf1 = zt1 + (y(tinc,5)*1000);
        zuf2 = zr1 + (y(tinc,5)*1000);
        zuf3 = zh1 + (y(tinc,5)*1000);

        zur1 = zt3 + (y(tinc,7)*1000);
        zur2 = zr3 + (y(tinc,7)*1000);
        zur3 = zh3 + (y(tinc,7)*1000);

%Determine the road function at this instant in time
        tspan = [0:0.01:10];
        roady = [-1200:5:4000];
        roadz = (F0*1000)*sin((roady-((tspan(tinc))*V*1000))/...
            ((lambda*1000)/(2*pi)));
        roadlength = length(roady);
        roadx = zeros(roadlength,1);

%Plot Wheel Treads
        hold off;
        surf(xt1,yt1,zuf1,'FaceColor',[0.1,0.1,0.1],'EdgeColor','none'),...
            camlight right,camlight right,campos([900,1500,800]),...
            camtarget([-1000,1500,800]),camva('auto'),xlim([-1200 900]),...
            ylim([-1200 4000]),zlim([-400 1600]);
        axis equal;
        showaxes('hide');
        %visible off;
        hold on;
        surf(xt3,yt3,zur1,'FaceColor',[0.1,0.1,0.1],'EdgeColor','none');

%Plot Wheel Rims
        surf(xr1,yr1,zuf2,'FaceColor',[0.8,0.8,0.8],'EdgeColor','none');
        surf(xr3,yr3,zur2,'FaceColor',[0.8,0.8,0.8],'EdgeColor','none');
```

```
%Plot Wheel Hubs
        surf(xh1,yh1,zuf3,'FaceColor',[0.8,0.8,0.8],'EdgeColor','none');
        surf(xh3,yh3,zur3,'FaceColor',[0.8,0.8,0.8],'EdgeColor','none');

%Plot the profile of the body of the car
        patch(bodyx,zsy,zsz,'g');

%Plot the profile of the road at this instant
        plot3(roadx,roady,roadz);

%Display the time
        text(0,500,2200,{t(tinc)},'FontSize',50);
        text(0,1300,2200,'Seconds','FontSize',30);

%Increment time marker
        if tinc < incf
        tinc = tinc + 1;
        elseif tinc == incf
        tinc = 1;
        end

%Create pause to slow down the animation and allow it to be viewed
        pause(0.001)

%End the while loop of the simulation when the exit button has been pushed
    end

%Define finish marker to be returned to the driving file
    fin = 1;

%Close the animation window
    close

%-------------------------------------------------------------------------%
%-----------------------------END OF PROGRAM------------------------------%
%-------------------------------------------------------------------------%
```

APPENDIX F

Consequential Effects

A car’s normal functioning is very easily taken for granted. It is hard to imagine that every time I or anyone else enters a car they are literally placing their lives in an engineer’s hand. As I embark on this design process I am fully aware of the consequences of creating something that is crucial to a car’s and its occupant’s safety.

It may be safe to say that in most cases when a car experiences a failure of some sort it is possible to control the incident and prevent injury to the occupants. However when a design is gambling with peoples lives I want to make sure it’s perfectly safe. I recall the firestone incident in America a few years ago whereby tyres on Ford Explorers would burst while travelling at high speeds. This was proven to have come about by using poorly manufactured tyres and the result was numerous deaths. The suspension is such an integral part of a cars basic function that accepting any chance of consequential effects is unacceptable.

The new design must be proven to be free from any chance of mechanical failure when exposed to greater than average stresses. In this case a safety factor will be used on top of the maximum expected normal loading. In order to achieve this fail safe state it may require me to over engineer the strut to a certain degree. While this may be viewed as wasteful in regards to both materials and power usage it is worth it for ensuring a fail free product. Should a mechanical failure occur with the strut so close near the tyre, the final result could be catastrophic? I can only begin to imagine such things occurring as a punctured tyre and steering failure which is reason enough to over-engineer.

A car’s stability is derived from its suspension which means that the controllability of the car into which the strut will be fitted will be directly affected. In designing and testing this new strut I need to be confident that all the aspects of the struts reaction to the road will not adversely affect the cars stability. Consequences that may come about by upsetting the car’s stability may be an inability to maintain control of the car at high speeds or around corners. With traffic moving in the opposite direction there is no room for mistakes in this area of the design.

Ultimately there is potential for loss of life if this new design does not function properly. The two main areas for concern are mechanical failure and loss of control however within these two broad umbrellas are limitless aspects of the design that will contribute to either undermining or approving the design for safe use. During the design process every aspect of the strut will be assessed as to the risk of either failing or causing failure and will be engineered until proven to pose no threat. I realise I have an ethical responsibility to ensure that my desire to achieve success does not override my commitment to quality and safety. If at any point I become aware that the design is not suitably meeting the safety standards of modern day vehicles I recognise that I have the responsibility to report on this and then seek to rectify the problem.

In the same ethical mind frame I realise that care has to be taken to respect the designs and patents of existing manufacturers. While I will be gaining a thorough understanding of existing suspension designs to prevent me from covering the same ground of others before me, I will be careful to respect their intellectual property.

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