DESIGNING AND SIMULATE AN ACTIVE PID CONTROLLER FOR A VEHICLE SUSPENSION SYSTEM VIA CONTROLLABLE DAMPING DEVICES

by

Mohd Ridhuan Bin Darus

Dissertation submitted in partial fulfilment of

the requirements for the

Bachelor of Engineering (Hons)

(Mechanical Engineering)

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
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Approved:

Dr. Vu Trieu Minh
Project Supervisor

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

Nov 2009

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

Mohd Ridhuan Bin Darus

ABSTRACT

This report was prepared mainly for the purpose of giving a brief idea on the understanding of the chosen project. It is also basically discusses the preliminary research done on the topic, which is **Designing and simulate an active PID controller for a vehicle suspension system via controllable damping devices.** The objective of this project is to simply design a vehicle suspension system by using Simulink in Matlab and performing a real-time simulation depicting the necessary information in regard to the objective of this project. By saying that the old-fashioned conventional passive suspension system cannot be able to provide a safety driving and comfort for the driver, the active suspension system will do. Implying a closed-loop controller for the vehicle suspension system will assist by reacting to an error signal and supply an output for correcting elements. A controllable damping device will serve as a mean that allows the implication of an active suspension system is been done. Main problem in these days is to find the best configuration that the driver favors in a real-time situation. Therefore, after I have ran through a series of simulations and testing it is shown that it is easier to execute the Matlab programme in the 2-dof and 4-dof which consist of roll and pitch rather than the 7-dof system. But after all, it is not an excuse for not completing the 7-dof vehicle suspension system and carry out the simulation with less disturbances.

ACKNOWLEDGEMENTS

The author wishes to take the opportunity to express his utmost gratitude to the individual that have taken the time and effort to assist the author in completing the project. Without the cooperation of these individuals, no doubt the author would have faced some minor complications throughout the course.

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CHAPTER 1: INTRODUCTION

1.1 BACKGROUND OF STUDY

The roads commonly used by motor vehicles are uneven. This unevenness causes vertical movements of the vehicle and the passengers during the driving process. The vehicle is connected to the road by the tire. Small unevenness in comparison to the tire contact patch size can be compensated by the tire elasticity, whereas larger unevenness entails a vertical acceleration or deflection of the wheels. In order not to transfer these accelerations into the vehicle structure, a length compensating element has to be placed between the wheel and the vehicle structure. Steel springs are the technologically simplest elements with variable length. Due to this fact it is also the most common length compensating element, whose force is a function of the length variation. It is usually used in the suspensions of motor vehicles. Different parts connected with springs generate oscillating systems. So there has to be added an energy absorbing element, the damper. Most suspension systems are passive or reactive. For example, a tire hitting a bump or dropping into a hole may not stay in contact with the road surface. The suspension system then reacts by compressing or extending the spring. These actions affect handling and ride quality, and send shock and vibration to the vehicle body. Therefore, the need for an active suspension system is vital to be taken into consideration.

1.2 PROBLEM STATEMENT

Every cars manufacturer has been emphasizing the point of maintaining a comfort ride and maximizing the safety while driving as their major aspect when producing a car. The main concern in achieving those prospects relies in the suspension system which they cannot simply put it aside without taking serious study on it. According to the present data which being shared by the supervisor noting that a normal conventional passive suspensions only consist of the spring and damping properties which are time-invariant. Passive elements can only store energy for some portion of a suspension cycle (springs) or dissipate energy (shock absorbers). No external energy is directly supplied to this type of suspension. This consequence clearly gives a picture where such vehicle cannot set the parameters of the suspension in regards to the desired output that provide comfort and safety driving to the driver.

1.3 OBJECTIVES AND SCOPE OF STUDY

The main objective of this research is:

- ✓ To complete the 7-dof suspension system and carry out respective simulation in regards to the stated problem

Currently, author in the phase where a full scale model of a car suspension system is to be implemented and further designing a 7-dof vehicle suspension system. And upon completion of the 7-dof vehicle suspension system, author will carry out the simulation and testing to further complete the objective.

CHAPTER 2: LITERATURE REVIEW

2.1 INTRODUCTION

Suspension springing and damping operates chiefly on the vertical oscillations of the vehicle. Ride comfort (vibrational loads on occupants and cargo) and driving safety (distribution of forces against the road surface as wheel-load factors fluctuate) are determined by the suspension. Several spring-damper systems will serve to illustrate the synergetic operation of vehicle components. In this case, an active PID controller is necessary in order to design an automotive suspension system which is a closed-loop control system. PID controller will play the role where it will serve as a mean of a control unit which can react to an error signal and supply an output for correcting elements; the PID controller itself which composed of the proportional mode-derivative mode-integral mode.

2.2 VEHICLE SUSPENSION SYSTEM

A suspension spring serves two purposes. First, it acts as a buffer between the suspension and frame to absorb vertical wheel and suspension movement without passing it on to the frame. Second, each spring transfers part of the vehicle weight to the suspension components it rests on, which transfers it to the wheels. Springs give way to absorb the vertical force of the moving wheel during jounce, then release that force during rebound as they return to their original shape and position.

The ideal spring creates little or no friction because friction interferes with spring movement. However, the less friction within a spring, the longer it continues to oscillate after it compresses and extends because nothing interferes with its movement. On a vehicle suspension, a shock absorber provides the friction needed to control and quickly stop spring oscillation. In Europe, shock absorbers are known as "dampers", which is actually a more accurate name, since the springs actually absorb road shocks while shock absorbers damp the spring action. Automotive shock absorbers use hydraulic friction, rather than mechanical, or surface-to-surface, friction to control spring oscillation.

Like mechanical friction, hydraulic friction generates heat by resisting movement. However, unlike mechanical friction, hydraulic friction is created without surface-to-surface contact between the moving parts. Therefore, a hydraulic shock absorber has a much longer service life than a mechanical device designed to perform the same task. The movement a shock absorber resist is suspension movement, since one end of the shock mounts to the frame and the other end attaches to a suspension member. The heat created by the internal hydraulic resistance of the shock absorber dissipates into the air surrounding the shock.

Listed below are a number of model of cars which has been implying active suspension/adaptive suspension system since the early of 19's dating until this present day

- 1987 Mitsubishi Galant "Dynamic ECS", world's first production semi-active electronically controlled suspension system
- 1989 Citroën XM (Hydractive, semi-active)
- 1990 Infiniti Q45 "Full-Active Suspension (FAS)", world's first production fully active suspension system
- 1990 Toyota Supra (Toyota Electronically Managed Suspension, TEMS)
- 1991 Mitsubishi GTO "Electronic Controlled Suspension"
- 1991 Toyota Soarer 'Active'
- 1992 Toyota Celica (Japan only)
- 1992 Citroën Xantia VSX (Hydractive 2, semi-active)
- 1993 Cadillac, several models with road sensing suspension.
- 1994 Citroën Xantia Activa (Hydractive 2 and active roll control)
- 1996 Jaguar XK8 'CATS' (optional)
- 1997 Jaguar XJ 'CATS' (standard on XJR model)
- 1999 Mercedes-Benz CL-Class, Active Body Control

- 1999+ Lexus LX470
- 2001 Citroën C5 (Hydractive 3, semi-active)
- 2002+ Jaguar S-Type 'CATS' (S-Type R model)
- 2002+ Mazda6 wagon 4wd
- 2002 BMW 7 Series
- 2002 Maserati Coupé
- 2002 Cadillac Seville STS, first MagneRide
- 2003 Mercedes-Benz S-Class
- 2003 Chevrolet Corvette, some Cadillacs and other GM vehicles with MagneRide
- 2004 Opel Astra 'IDS+' (optional)
- 2004 - 2007 Volvo S60R "4-C Active Chassis"
- 2004 - 2007 Volvo V70R "4-C Active Chassis"
- 2005 Citroën C6 (Hydractive 3+, semi-active)
- 2007 Lexus GS, Active Stabilizer Suspension System
- 2007 Maserati GranTurismo
- 2008 + Audi TT Magnetic Rid

Active or adaptive suspension is an automotive technology that controls the vertical movement of the wheels via an onboard system rather than the movement being determined entirely by the surface on which the car is driving. The system therefore virtually eliminates body roll and pitch variation in many driving situations including cornering, accelerating and braking. This technology allows car manufacturers to achieve a higher degree of both ride quality and car handling by keeping the tires perpendicular to the road in corners, allowing for much higher levels of grip and control. An onboard computer detects body movement from sensors located throughout the vehicle, and, using data calculated by opportunecontrol techniques, controls the action of the suspension.

Active suspensions can be generally divided into two main classes: pure active suspensions, and semi-active suspensions.

2.2.1 ACTIVE SUSPENSION

Active suspensions, the first to be introduced, use separate actuators which can exert an independent force on the suspension to improve the riding characteristics. The drawbacks of this design (at least today) are high cost, added complication/mass of the apparatus needed for its operation, and the need for rather frequent maintenance and repairs on some implementations. Maintenance can also be problematic, since only a factory-authorized dealer will have the tools and mechanics who know how to work on the system, and some issues can be difficult to diagnose reliably.

Citroen's Active Wheel incorporates an in-wheel electrical suspension motor that controls torque distribution, traction, turning maneuvers, pitch, roll and suspension damping for that wheel, in addition to an in-wheel electric traction motor.

Hydraulic actuated

Hydraulically actuated suspensions are controlled with the use of hydraulic servomechanisms. The hydraulic pressure to the servos is supplied by a high pressure radial piston hydraulic pump. Sensors continually monitor body movement and vehicle ride level, constantly supplying the computer with new data.

As the computer receives and processes data, it operates the hydraulic servos, mounted beside each wheel. Almost instantly, the servo regulated suspension generates counter forces to body lean, dive, and squat during various driving maneuvers.

In practice, the system has always incorporated the desirable self-levelling suspension and height adjustable suspension features, with the latter now tied to vehicle speed for improved aerodynamic performance, as the vehicle lowers itself at high speed.

Colin Chapman - the inventor and automotive engineer who founded Lotus Cars and the Lotus Formula One racing team - developed the original concept of computer management of hydraulic suspension in the 1980s, as a means to improve cornering in racing cars. Lotus developed a version of its 1985 Excel model with electro-hydraulic active suspension, but this was never offered to the public.

Computer Active Technology Suspension (CATS) co-ordinates the best possible balance between ride and handling by analysing road conditions and making up to 3,000 adjustments every second to the suspension settings via electronically controlled dampers.

Electromagnetic recuperative

This type of active suspension uses linear electromagnetic motors attached to each wheel independently allowing for extremely fast response and allowing for regeneration of power used through utilizing the motors as generators. This comes close to surmounting the issues with hydraulic systems with their slow response times and high power consumption. It has only recently come to light as a proof of concept model from the Bose company, the founder of which has been working on exotic suspensions for many years while he worked as an MIT professor.

2.2.2 SEMI-ACTIVE SUSPENSION

Semi-active systems can only change the viscous damping coefficient of the shock absorber, and do not add energy to the suspension system. Though limited in their intervention (for example, the control force can never have different direction than that of the current speed of the suspension), semi-active suspensions are less expensive to design and consume far less energy. In recent times, research in semi-active suspensions has continued to advance with respect to their capabilities, narrowing the gap between semi-active and fully active suspension systems.

Solenoid/valve actuated

This type is the most economic and basic type of semi-active suspensions. They consist of a solenoid valve which alters the flow of the hydraulic medium inside the shock absorber, therefore changing the dampening characteristics of the suspension setup. The solenoids are wired to the controlling computer, which sends them commands depending on the control algorithm (usually the so called "Sky-Hook" technique).

Magneto rheological damper

Another fairly recently-developed method incorporates magneto rheological dampers with a brand name MagneRide. It was initially developed by Delphi Corporation for GM and was standard, as many other new technologies, for Cadillac Seville STS (from model 2002), and on some other GM models from 2003. This was an upgrade for semi-active systems ("automatic road-sensing suspensions") used in upscale GM vehicles for decades, and it allows, together with faster modern computers, changing the stiffness of all wheel suspensions independently on every road inch on highway speed. These dampers are finding increased usage in the USA and already leases to some foreign brands, mostly in more expensive vehicles. In this system, being in development for 25 years, the damper fluid contains metallic particles. Through the onboard computer, the dampers' compliance characteristics are controlled by an electromagnet. Essentially, increasing the current flow into the damper raises the compression/rebound rates, while a decrease softens the effect of the dampers. Information from wheel sensors (about suspension extension), steering, acceleration sensors and some others is used to calculate the optimized stiffness. Very fast reaction of the total system allows, for instance, make softer passing by a single wheel above a bump or a rock on the road.

2.3 INTRODUCTION TO VEHICLE SUSPENSION SYSTEM

As one of the most important systems in a vehicle the suspension is a major focus of automotive engineers. Suspension is the term given to the system of shock absorbers and springs as well as linkages, which connect a vehicle to its wheels. Its main functions are to isolate vehicle body from road induced vibration, to maintain contact between tyre and road, to Control body pitch and roll, to limit wheel movement, and to support range of loads.

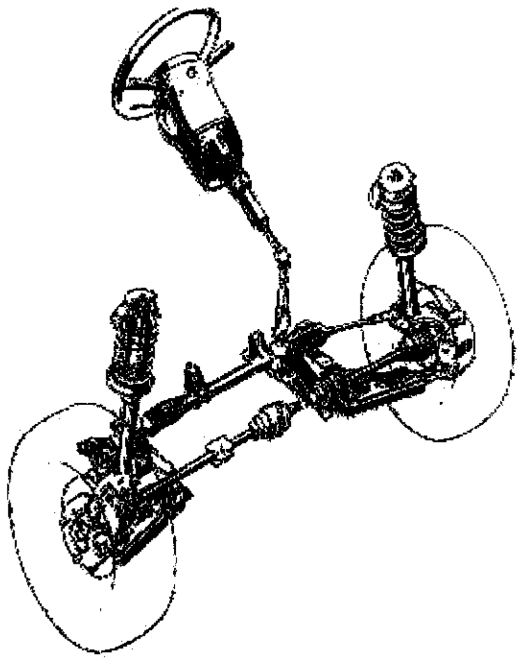


Figure 1: McPherson suspension

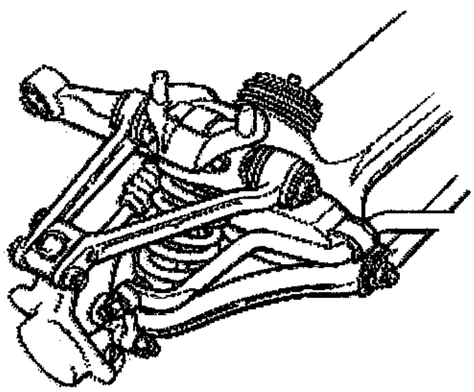


Figure 2: Double-wishbone suspension

The first one is called McPherson suspension. The shock absorber (also known as damper) and the coil spring are attached to an A-arm at the bottom to the wheel and attached at the top to the vehicle body. The second one is a double-wishbone suspension. It has two parallel wishbone-shaped arms to locate the wheel. Each wishbone (or arm) has two mounting positions to the chassis and one at the wheel hub. The shock absorber and coil spring mount to the wishbones to control vertical movement.

Hydraulic shock absorbers are most often used in a vehicle suspension. It can dissipate the kinetic energy in the system and convert it into heat.

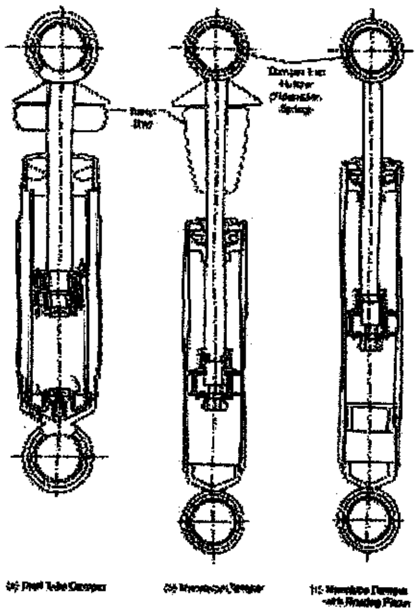


Figure 3 : Shock absorbers

Picture above shows three types of hydraulic shock absorbers used in automotive applications. Whenever the hydraulic oil flow through the damper valve in the shock absorber, damping force is generated. The more restrictive the damper valve is, the more damping force can be generated.

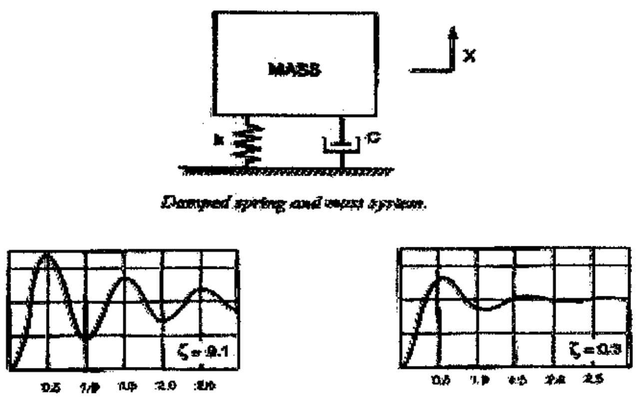


Figure 4 : Damper spring and damper system

Damped spring &damper system demonstrates the effects of a shock absorber on the body movement. It can reduce the oscillation of the suspension movement by dissipating the energy. It is customary to have less damping for compression motion than that of extension motion so that less force is transmitted to the vehicle when it encounters bump-type disturbance. By comparison, more damping is provided for rebound motion in order to quickly dissipate energy stored in the suspension system.

Vehicle Models

When we design or analyse a suspension system, we need a physical model.

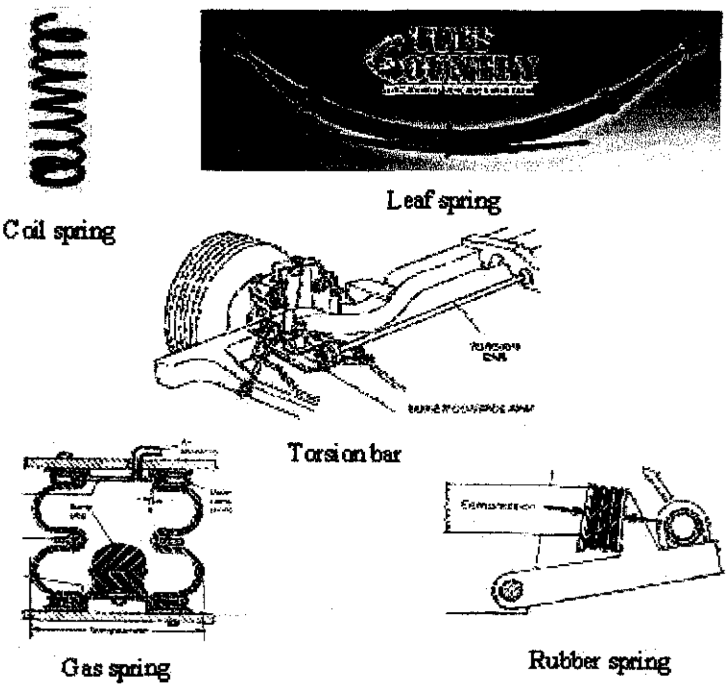


Figure 5 : Springs

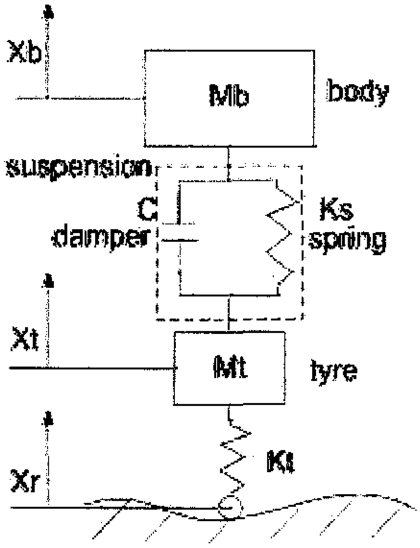


Figure 6 : Quarter-car model

Quarter car model shows a simplified linear quarter car model. Each corner of the vehicle can be simplified as a body mass, a suspension (which is composed of a shock absorber / damper and a spring) and a tyre mass. It is assumed there is a spring in the tyre to model the compressibility of the tyre. The movement of the vehicle body is as going to be mentioned in the next chapter.

If we extend the quarter car model to the full vehicle, we can get a full vehicle model as shown below.

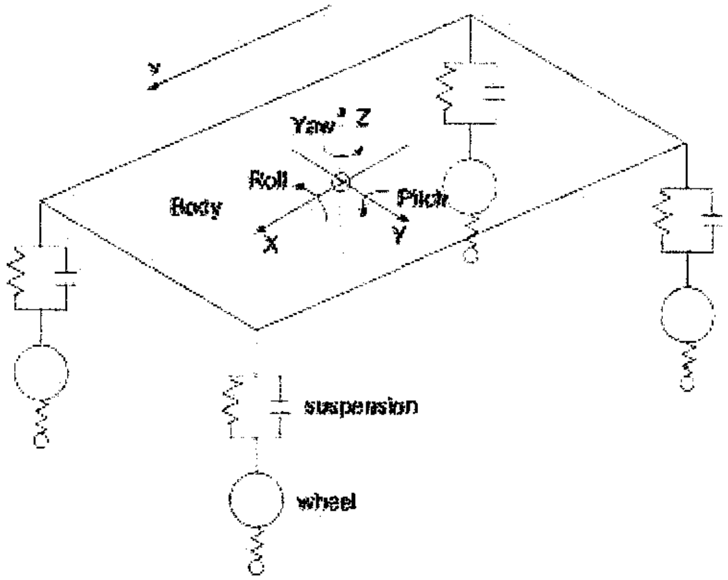


Figure 7 : Full vehicle model

Besides the vertical movement at each corner, the vehicle body also has three rotational movements, the roll, pitch and yaw.

<u>How to evaluate a suspension performance</u>

Experienced drivers used to carry out subjective assessments to judge a suspension performance. They drive the car over all kinds of roads and judge the suspension by personal feelings, e.g. noise, vibration and harshness. However, people's feelings are tricky and sometimes for the same car drivers may have different subjective judgements. Therefore, we need some objective standards to evaluate a suspension performance.

In vehicle suspension designs, there are 4 main criteria to objectively evaluate a suspension performance. The first is the body acceleration, which is used to assess ride comfort. Previous research has shown that the vehicle body acceleration is highly related to a passenger's feeling about ride comfort. The second is the dynamic tyre deflection, which is connected with handling performance. When a tyre deflection is too big, it may lose contact with the road. In this case, the driver can lose control of the vehicle. The third is the suspension deflection which means the suspension should have enough working space. When the suspension deflection is too big, the wheel may hit the vehicle body and make the driving very harsh. The last one is the body attitude. We want to keep the vehicle body level all the time during the driving; therefore pitch, roll and warp angle are very important.

Since the body acceleration is related to the feeling of a human being, how will it affect a driver's judgement? Modern vehicle tends to have a natural frequency of about 1.0-2.0Hz. The reason is that this frequency is very close to the way an adult is walking based on a pace of 30in and speed of 2.5-4mph. Previous medical researches have proven that most uncomfortable frequencies lie in 20-200Hz. For example, if people are exposed to 4-8Hz vertical vibrations, they will soon get fatigued. Human head and neck are sensitive to 18-20Hz vibrations and visceral region is sensitive to 5-7Hz vibrations. Besides the high frequency vibrations, some low frequency vibrations can also make people feel uncomfortable. For example, sea sickness is related to the vibrations below 0.75Hz. Also, lateral and pitch movements can make people feel uncomfortable.

Trade-offs in suspension design

In the past years automotive engineers are trying every method to improve the suspension design. However, due to some inherent trade-offs in suspension itself, it is very difficult to design a perfect suspension. The best known is the compromise between ride and handling. For example, to have a comfortable ride, soft spring and damper are helpful; however, the soft spring and damper may result in excessive tyre movement and make the handing worse. On the other hand, to stiff the spring and damper for a good handling may make the suspension feel very harsh. Accordingly, most conventional passive suspensions may only satisfy the essential requirements and will compromise on some of the less important considerations. For example, luxury limousines tend to use soft suspensions to offer good ride comfort, while sports cars usually have stiff suspensions to achieve superior handling performance.

2.3.1 ACTIVE SUSPENSIONS

With the development of modern computers, electronics, hydraulics and control technologies, a new suspension system; the active suspension, provides a possible solution to the conflict requirements between ride and handling. An active suspension refers to a suspension system which uses a micro-computer and sensors in a feedback loop to improve the suspension performance.

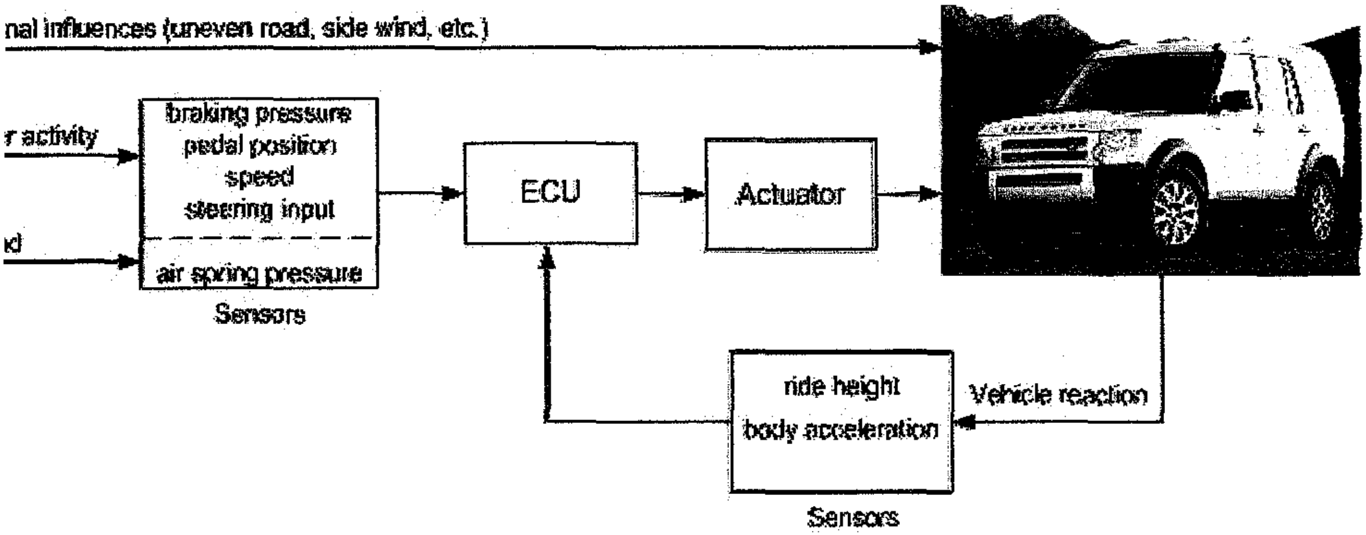


Figure 8 : Simple diagram of an active suspension

Arrangement of an active suspension shows a typical arrangement of the active suspension. In general, it is composed of:

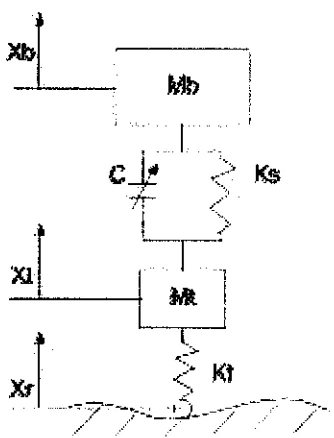
- Sensors – various sensors are installed around the vehicle to monitor the vehicle conditions and driver activities.
- Electronic control unit (ECU) – all the sensor signals are fed to a microcomputer, also known as ECU. With the aid of a programmed map memory, calculations are made as to what adjustment should be made to the suspension.
- Actuators – the instructions from ECU are converted into electrical signals and directed to various actuators to control the suspension. Hydraulic actuators are most often used for their compact volume and light weight.

Categories of active suspensions

Depending on various hardware employed in active suspensions, they can be divided into four categories.

Semi-active suspensions

The term semi-active suspension is often used to refer to a controlled damper under closed-loop control, which means the control is realized by varying the damper's damping rate



Semi-active suspension

A semi-active suspension is only capable of dissipating energy. According to different damper configurations, semi-active dampers can be classified into the following categories.

Dampers with controllable orifice

The damping force in a shock absorber is generated when the oil flow through the hydraulic orifices in the damper valve of the shock absorber. The smaller the orifice is, the larger damping force can be generated. Therefore, we can control the opening of the orifice to adjust the shocker's damping force. Currently ZF Sachs (a German tier 1 company) offers a line of semi-active shock absorbers under the name of CDC (continuously damping control) as shown below.

Figure 9 : Damper with controllable orifice

Dampers with controllable fluid

If the hydraulic orifice is fixed, we can vary the oil viscosity to control the damping force. The bigger the oil viscosity is, the larger damping force can be generated. ER (Electrorheological) or MR (Magneto-rheological) fluid can be used for this purpose. There are polarizable particles of a few microns in the oil. When electrical of magnetic field is applied to the oil, the particles will be polarised and distributed in a sequential order as shown in.

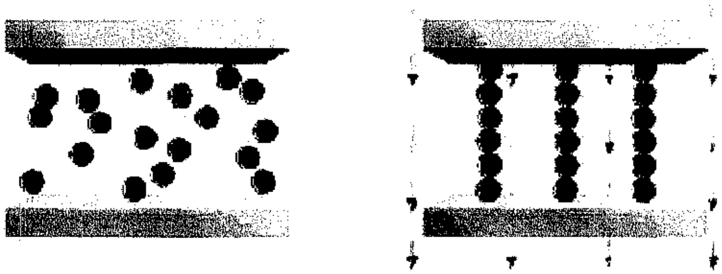


Figure 10 : Particles in MR damper

Work principle of electro-heological & magnetorheological dampers: Particles in an MR/ER fluid left without & right with applied magnetic/electrical field.

As a result, the oil viscosity changes, depending on the strength of the electrical/magnetic filed. Semi-active suspensions have been successfully used in some vehicle models, e.g. Audi A8, Lancia Thesis and the new Opel / Vauxhall Astra.

Fully active suspensions

Different from semi-active suspensions, a fully active suspension does not change the damper characteristics, but add a force generator in parallel with the passive damper and spring as shown in

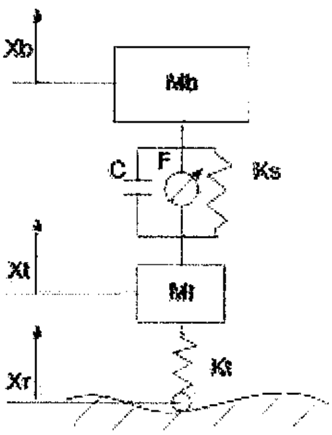


Figure 11 : Fully active suspension

Therefore, the suspension can not only dissipate energy, but also inject energy into the system. That is why we call it fully active suspension. Normally the power of the force generator is supplied by the engine; therefore, compared with semi-active suspensions, active suspensions have higher cost and power consumptions. But as a return, it has better perform-ance than semi-active ones. Depending on the response speed of the actuator, there are fast active and slow active suspensions. Slow active suspensions have low cost and power consumption, but the performance is not as good as fast active ones.

The applications of fully active suspension can be found on Toyota Soarer, Nissan Q45A and some Mercedes-Benz models, for example, the SL500.

2.4 THE ACTIVE SUSPENSION INVESTIGATED AT UNIVERSITY OF BATH

Introduction

Cooperated with Jaguar, Ford and TOKICO (USA) Inc, a prototype hydro-pneumatic active suspension has been investigated at the University of Bath. From individual components to overall system, extensive computer simulations and experimental measurements were carried out. It is hoped that this work could lead to further understandings and improved designs of a computer controlled active suspension. For confidential considerations, no detailed information about the system is enclosed with this talk.

The active suspension system

The active suspension investigated in this research shows the layout of the active suspension investigated in this research. It is a fully slow active suspension system. The power is supplied by a gear pump driven by the engine. The oil from the pump goes through a fail safe valve unit first. This unit has two main functions. The first is to shut off the system in case of emergency. In this circumstance the system will behave as a passive suspension. The second is to adjust the supply pressure. When the active suspension need not work, the fail safe valve unit will reduce the supply pressure level. It increases supply pressure only when it is necessary. In this way, the system power consumption is reduced. Then the oil goes to a flow control valve at each corner, which controls the flow to and from the gas strut (composed of a coil spring and a gas shock absorber) to adjust the suspension movement. The flow control valve is controlled by an ECU, which makes judgement upon receiving various sensor signals around the vehicle.

The prototype active suspension was implemented on a Jaguar S-Type saloon and extensive experiments carried out.

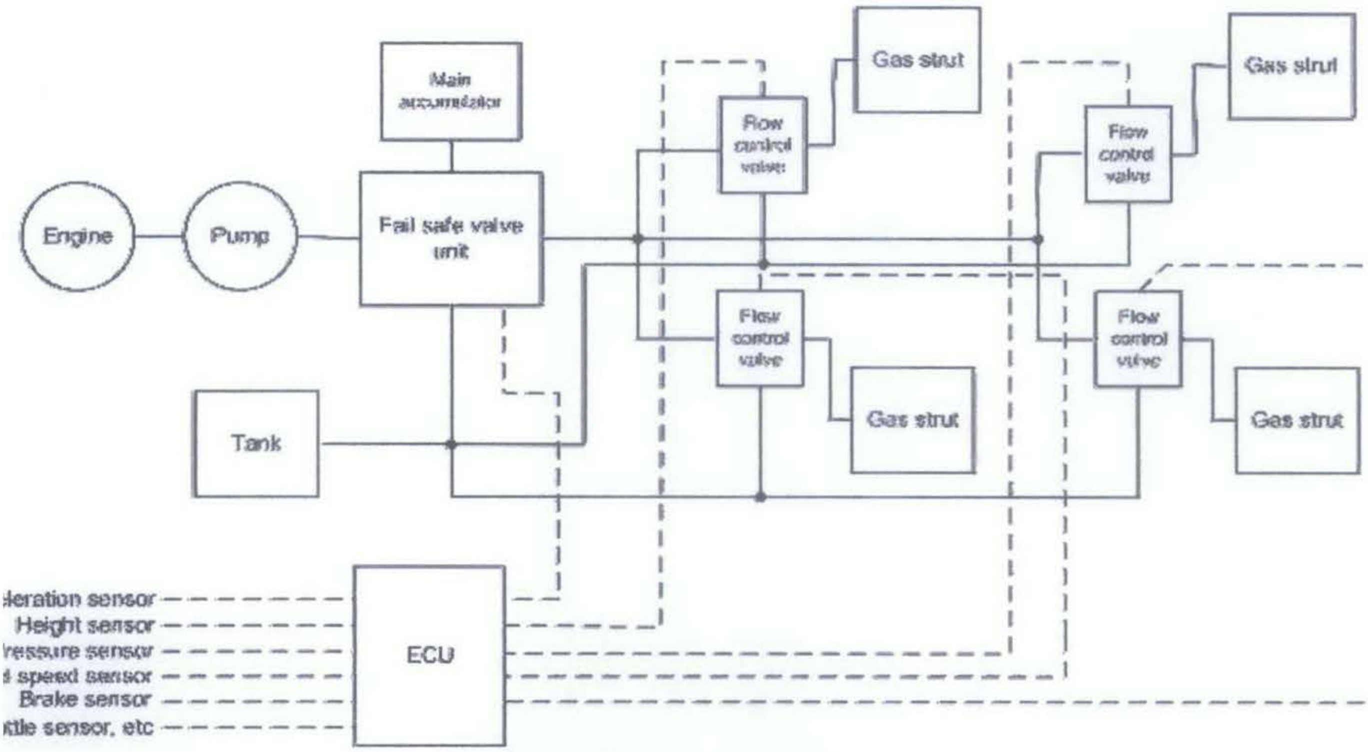
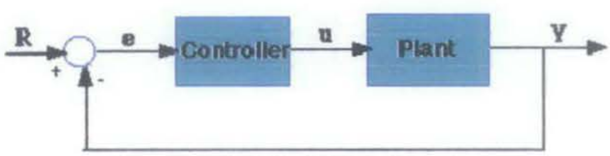


Figure 12 : Architecture of a system

2.5 PID CONTROLLER

PID Controller (closed-loop controller) is a combination of all three modes of control (proportional, integral and derivative) which enables a controller to be produced and has no offset error and reduces the tendency for oscillations. Such a controller is known as a three-mode controller or the well-known PID controller.



$$K_p + \frac{K_I}{s} + K_D s = \frac{K_D s^2 + K_p s + K_I}{s}$$

Figure 13 : PID controller

Kp = proportional gain

KI = integral gain

Kd = derivative gain

The variable (e) represents the tracking error, the difference between the desired input value (R) and the actual output (Y). This error signal (e) will be sent to the PID controller, and the controller computes both the derivative and the integral of this error signal. The signal (u) just past the controller is now equal to the proportional gain (Kp) times the magnitude of the error plus the integral gain (Ki) times the integral of the error plus the derivative gain (Kd) times the derivative of the error.

$$u = K_P e + K_I \int e\,dt + K_D \frac{de}{dt}$$

This signal (u) will be sent to the plant, and the new output (Y) will be obtained. This new output (Y) will be sent back to the sensor again to find the new error signal (e). The controller takes this new error signal and computes its derivative and it's integral again. This process goes on and on.

2.6 SKYHOOK CONTROL OF AN SDOF SYSTEM

One method to eliminate the tradeoff between resonance control and high frequency isolation is to reconsider the configuration of the suspension system. For instance, consider moving the damper from between the suspended mass and the base to the position shown in Fig. 2.9. The damper is now connected to an inertial reference in the sky (i.e., a ceiling that remains vertically fixed relative to a ground reference). Notice that this is a purely fictional configuration, since for this to actually happen, the damper must be attached to a reference in the sky that remains fixed in the vertical direction, but is able to translate in the horizontal direction. Ignoring this problem at the moment, we will focus on the performance of this configuration.

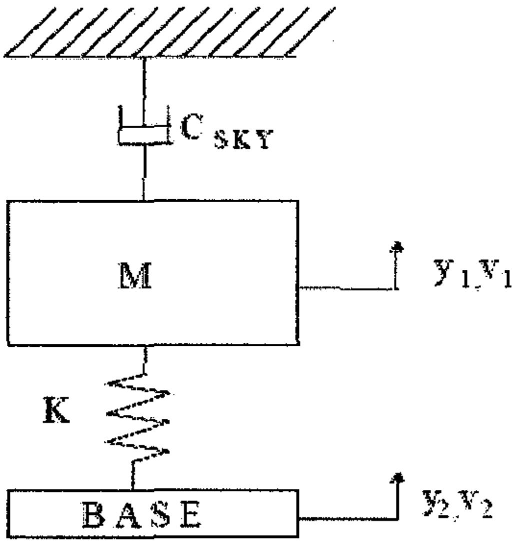


Figure 14 : Skyhook

The transmissibility of this configuration can be derived to be

$$\frac{y_1}{y_2} = \frac{1}{1-(\frac{\omega}{\omega_n})^2 + j2\zeta_p(\frac{\omega}{\omega_n})}$$

Another approach to achieving skyhook damping is to use semiactive dampers. Semiactive dampers allow for the damping coefficient, and therefore the damping force, to be varied between high and low levels of damping. Early semiactive dampers were mechanically adjustable by opening or closing a bypass valve. The only power required for the damper is the relatively small power to actuate the valve. For this research, we are using a magnetorheological damper which varies the damping by electrically changing the magnetic field applied to the magnetorheological fluid.

2.7 MAGNETORHEOLOGICAL DAMPER

The purpose of this section is to introduce the theoretical and practical applications of a magnetorheological (MR) fluid for a controllable MR damper. First, the concept of the MR fluid will be introduced. Next, the practical realization of an MR damper will be discussed. Finally, the performance of the MR damper used for this research will be investigated.

Magnetorheological Fluids

Magnetorheological fluids are materials that exhibit a change in rheological properties (elasticity, plasticity, or viscosity) with the application of a magnetic field. The MR effects are often greatest when the applied magnetic field is normal to the flow of the MR fluid. Another class of fluids that exhibit a rheological change is electrorheological (ER) fluids. As the name suggests, ER fluids exhibit rheological changes when an electric field is applied to the fluid. There are, however, many drawbacks to ER fluids, including relatively small rheological changes and extreme property changes with temperature. Although power requirements are approximately the same, MR fluids only require small voltages and currents, while ER fluids require very large voltages and very small currents. For these reasons, MR fluids have recently become a widely studied 'smart' fluid. Besides the rheological changes that MR fluids experience while under the influence of a magnetic field, there are often other effects such as thermal, electrical, and acoustic property changes. However, in the area of vibration control, the MR effect is most interesting since it is possible to apply the effect to a hydraulic damper. The MR fluid essentially allows one to control the damping force of the damper by replacing mechanical valves commonly used in adjustable dampers. This offers the potential for a superior damper with little concern about reliability, since if the MR damper ceases to be controllable, it simply reverts to a passive damper.

Construction of an MR Damper

Magnetorheological (MR) fluids are manufactured by suspending ferromagnetic particles in a carrier fluid. The ferromagnetic particles are often carbonyl particles, since they are relatively inexpensive. Other particles, such as iron-cobalt or iron-nickel alloys, have been used to achieve higher yield stresses from the fluid. Fluids containing these alloys are impractical for most applications due to the high cost of the cobalt or nickel alloys. A wide range of carrier fluids such as silicone oil, kerosene, and synthetic oil can be used for MR fluids. The carrier fluid must be chosen carefully to accommodate the high temperatures to which the fluid can be subjected. The carrier fluid must be compatible with the specific application without suffering irreversible and unwanted property changes. The MR fluid must also contain additives to prevent the sedimentation of and promote the dispersion of the ferromagnetic particles. A top-level functional representation of the MR damper is shown in Fig. 15. The fluid that is transferred from above the piston to below (and vice-versa) must pass through the MR valve. The MR valve is a fixed-size orifice with the ability to apply a magnetic field, using an electromagnet, to the orifice volume. This results in an apparent change in viscosity of the MR fluid, causing a pressure differential for the flow of fluid which is directly proportional to the force required to move the damper rod.

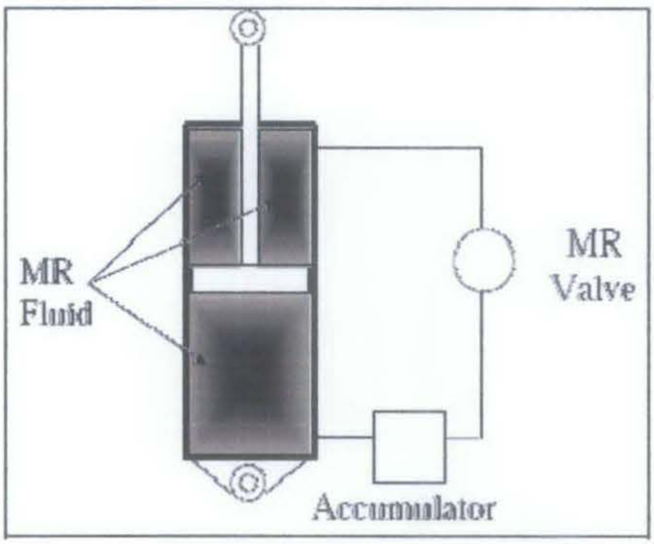


Figure 15 : Functional representation

The accumulator is a pressurized volume of gas that is physically separated from the MR fluid by a floating piston or bladder. The accumulator serves two purposes. The first is to provide a volume for the MR fluid to occupy when the shaft is inserted into the damper cylinder. The second is to provide a pressure offset so that the low pressure side of the MR valve is not reduced enough to cause cavitation of the MR fluid. An elegant and compact design of the MR damper developed by Lord Corporation and used for this research is shown in Fig.16. All of the external components have been incorporated internally. This provides a compact design that is very similar in size and shape to existing passive dampers. The only external parts are the two electrical leads for the electromagnet, which are connected to the controller.

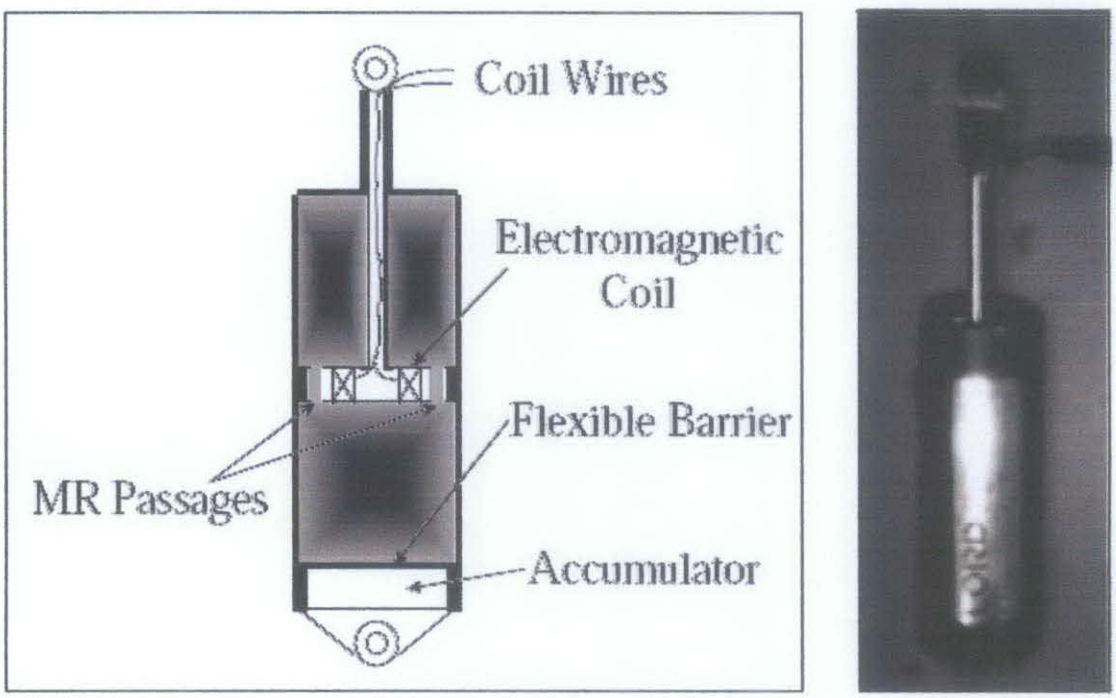


Figure 16 : LORD MR compact damper visual and actual

Performance of the MR Damper

For typical passive dampers, the damper performance is often evaluated based on the force vs. velocity characteristics. For an ideal viscous damper, the force vs. velocity performance is shown in Fig. 17. The slope of the force vs. velocity line is known as the damper coefficient, C. Frequently, the force vs. velocity line is bilinear and asymmetric, with a different value of C for jounce (compression) and rebound (extension), as shown in Fig. 18. In the case of a vehicle suspension, the damping curve is shaped (or tuned) by a ride engineer for each particular application. Therefore, the operational envelope of a passive damper is confined to a pre-designed force-velocity characteristic.

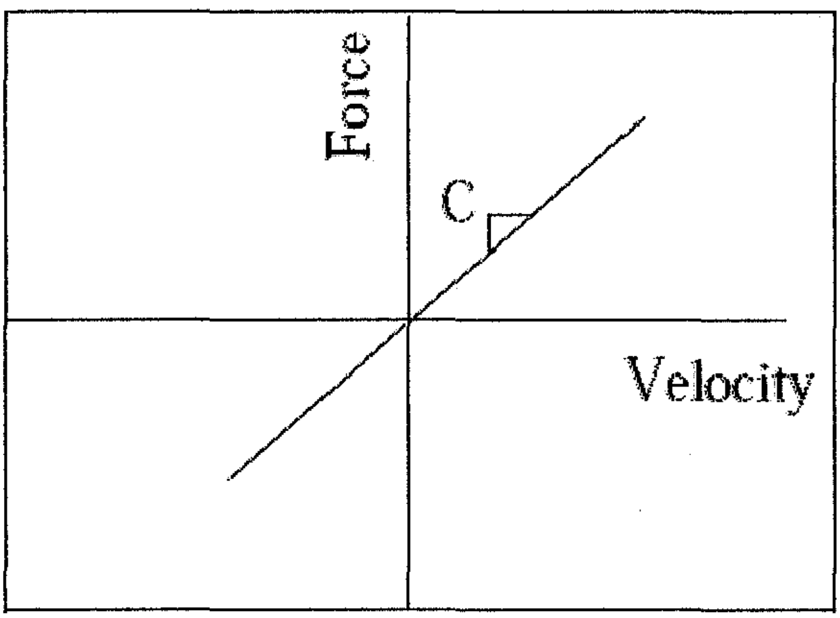


Figure 17 : Linear damper characteristics

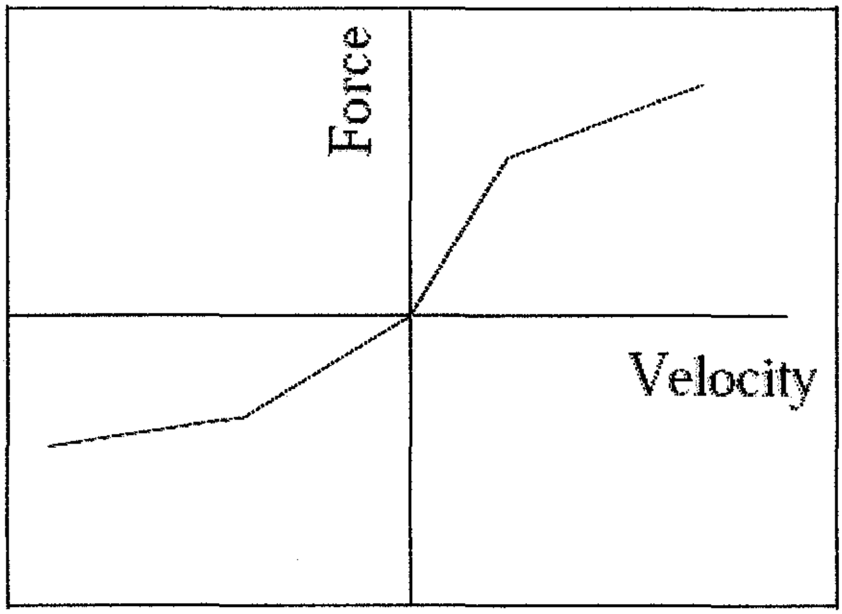


Figure 18 : Bilinear, Asymmetric Damping Characteristics

In the case of MR dampers, the ideal force vs. velocity characteristics are as shown in Fig. 19. This results in a force vs. velocity envelope that can be described as an area rather than a line in the force-velocity plane. Effectively, the controller can be programmed to emulate any damper force-velocity characteristic or control policy within the envelope.

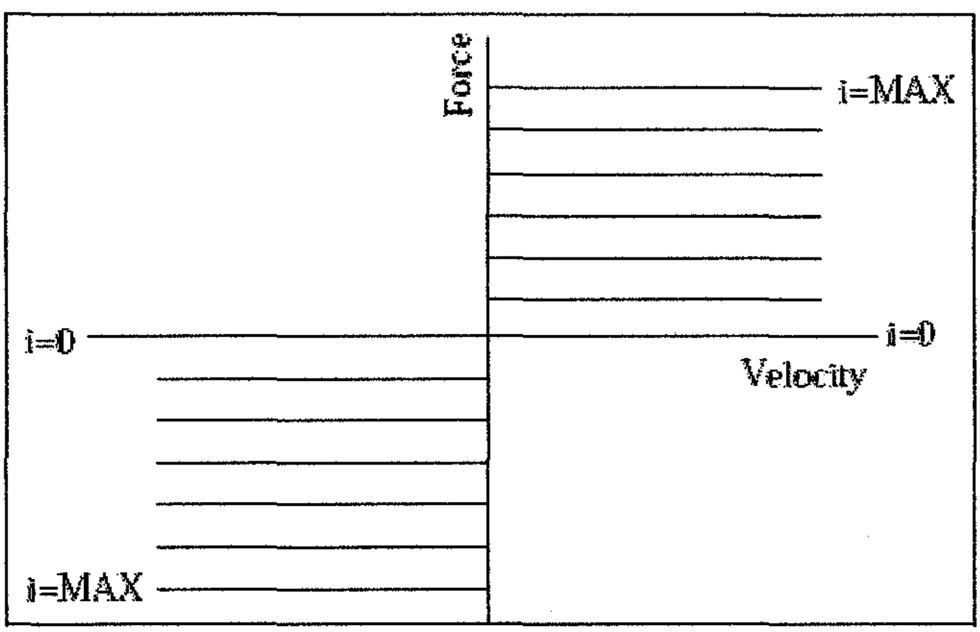


Figure 19 : Ideal MR damper performance

We can model the ideal MR damper according to

F i MRDAMPER =a

where a is a constant and *i* is the damper current. Figure 2.11 shows the nonlinear force-velocity characteristics for the MR damper used for this research. The model does not capture the fine details of the actual MR damper, but it captures the gross behavior of the MR damper. Some of the effects missing from the model include the magnetic field saturation, hysteresis, and the force due to the pressurized accumulator. As will be shown in later chapters, this approximation is sufficient for designing MR dampers for most applications, including the seat suspensions.

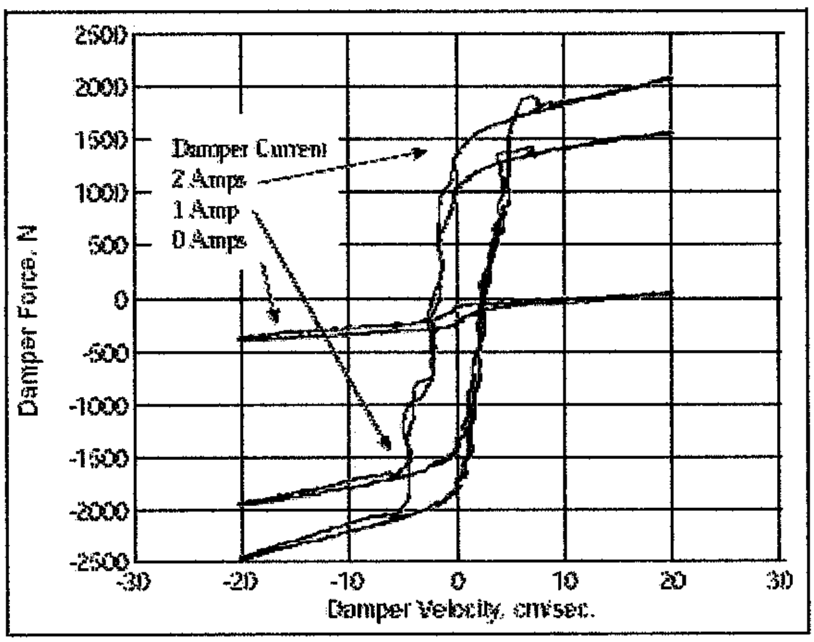
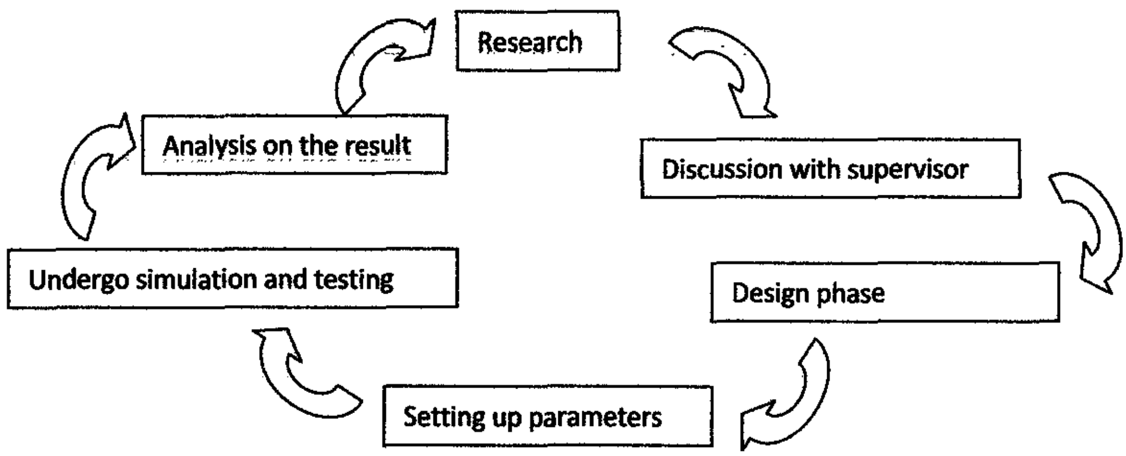


Figure 20 : MR damper performance envelope

CHAPTER 3: METHODOLOGY

3.1 RESEARCH METHODOLOGY

In order to make sure the project is in well being and both supervisor and I can keep track with the current progress of the job, a simple basic methodology might be useful:



3.2 PROJECT IDENTIFICATION

3.2.1 7-DOFs VEHICLE SUSPENSION SYSTEM

Before the author can start over with the 7-DOFs vehicle suspension system and at the same time covering the full scale model of a vehicle, the author first had to build up the simulink for the 2-DOFs and 4-DOFs of a vehicle suspension system. Basically both 2-dof and 4-dof vehicle suspension system will be the main foundation for realizing the 7-dof vehicle suspension system.

3.2.1.1 (A) ¼ MODEL, 2-DOFs SUSPENSION SYSTEM

At the very early stage of developing the 7-DOFs vehicle suspension system, author first takes the ¼ model, 2-DOFs suspension system to give the project a kick start. Consequently the vehicle suspension system will later proceed with the 4-dof and 7-dof.

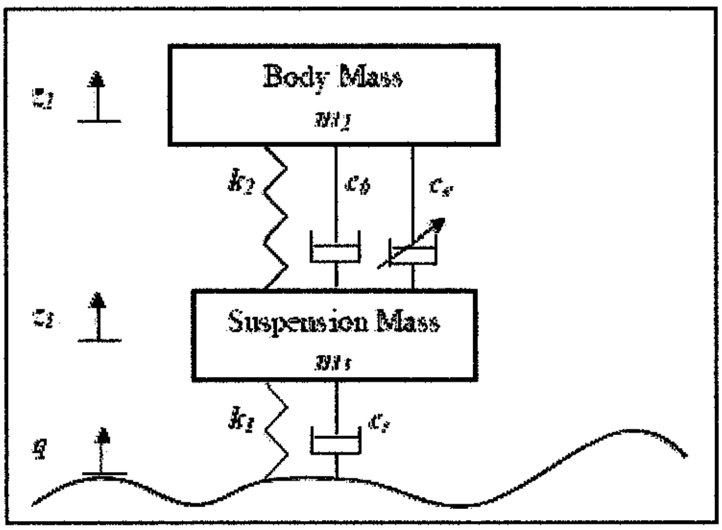


Figure 21 : 1/4 model of the 2-DOF's suspension system

From the depicted diagram, author further formulated the dynamics formula from the respective values.

The dynamics formula:

$$m_1\ddot{z_1} + (c_o + c_e)(\dot{z_1} - \dot{z_2}) + k_2(z_1 - z_2) + k_1(z_1 - q) + c_t(\dot{z_1} - \dot{q}) - m_1 g - F_r = 0$$

$$m_2\ddot{z_2} + (c_o + c_e)(\dot{z_2} - \dot{z_1}) + k_2(z_2 - z_1) - m_2 g + F_r = 0$$

Where, F_r is constant friction force

To let $c_e(\dot{z_1} - \dot{z_2}) = f_d$

And to ignore $c_t(\dot{z_1} - \dot{q}) = F_t$

Therefore, the following model was introduced

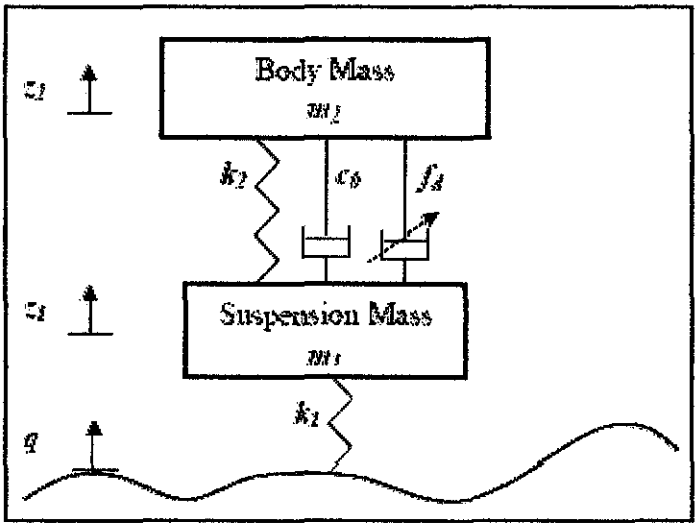


Figure 22 : 1/4 model without C

And the newly formulated dynamics formula will be as follows

$$m_1\ddot{z}_1 = -c_0(\dot{z}_1 - \dot{z}_2) - k_2(z_1 - z_2) - k_1(z_1 - q) + f_d - F_r + m_1 g$$

$$m_2\ddot{z}_2 = -c_0(\dot{z}_2 - \dot{z}_1) - k_2(z_2 - z_1) - f_d + F_r + m_2 g = 0$$

Into the matrix format:

$$[M]\{\ddot{Z}\} + [C_o]\{\dot{Z}\} + [K]\{Z\} + F_d = \{Q\}$$

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix}\begin{pmatrix} \ddot{z}_1 \\ \ddot{z}_2 \end{pmatrix} + \begin{bmatrix} c_0 & -c_0 \\ -c_0 & c_0 \end{bmatrix}\begin{pmatrix} \dot{z}_1 \\ \dot{z}_2 \end{pmatrix} + \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}\begin{pmatrix} z_1 \\ z_2 \end{pmatrix} + \begin{pmatrix} -f_d + F_r - m_1 g \\ f_d - F_r + m_2 g \end{pmatrix} = \begin{pmatrix} k_1 q \\ 0 \end{pmatrix}$$

Where:

$$M = \begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \qquad C_o = \begin{bmatrix} c_o & -c_o \\ -c_o & c_o \end{bmatrix} \qquad K = \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}$$

$$F_d = \begin{pmatrix} -f_d + F_r - m_1 g \\ f_d - F_r + m_2 g \end{pmatrix} \quad Q = \begin{pmatrix} k_1 q \\ 0 \end{pmatrix}$$

F_r is a constant friction

3.2.1.2 (B) ½ MODEL, 4-DOFs (PITCH) SUSPENSION SYSTEM

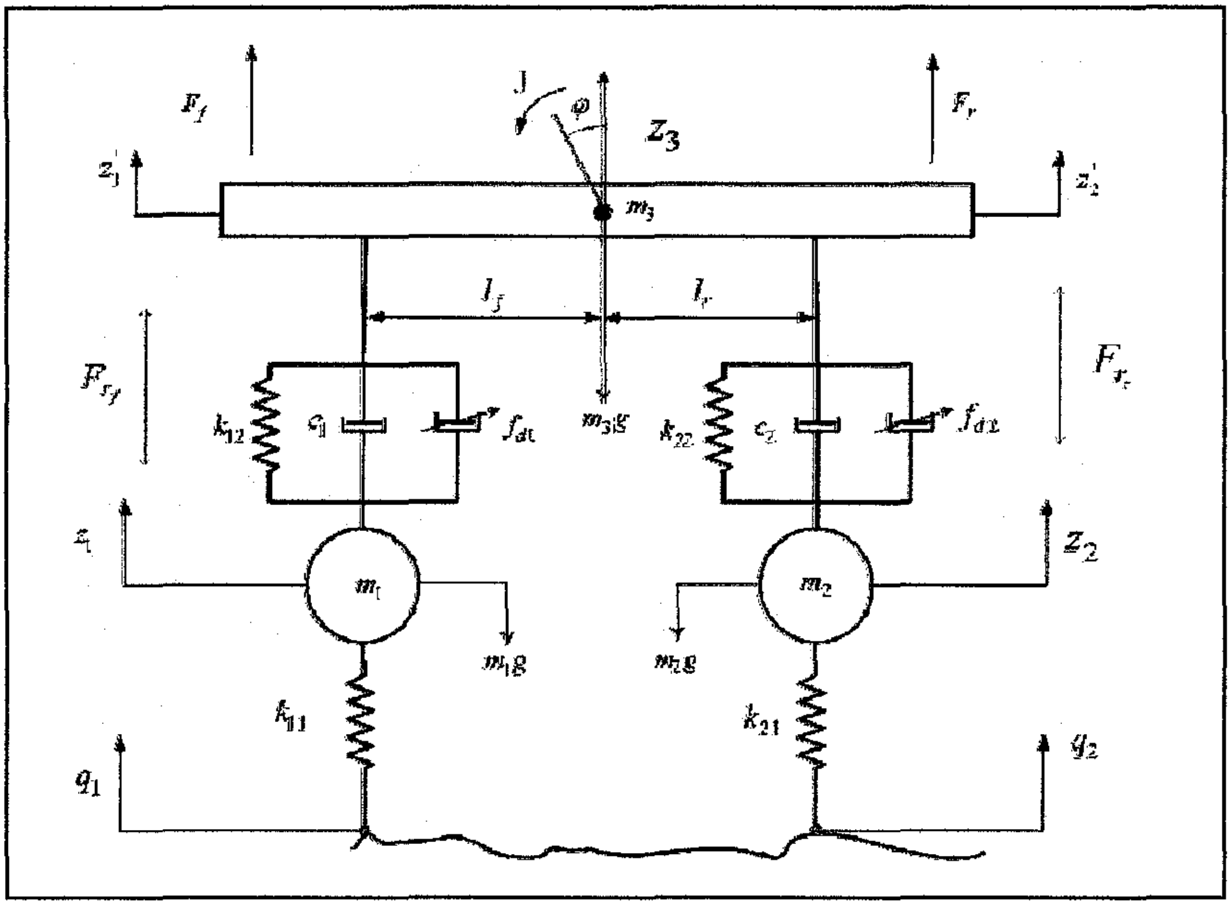


Figure 23 : 1/2 model of 4-DOF's suspension system (pitch)

$$m_1\ddot{z}_1 = k_{11}(z_1 - q_1) + c_1(\dot{z}'_1 - \dot{z}_1) + k_{12}(z'_1 - z_1) + f_{d1} + F_{r_f} + m_1 g$$

$$m_2\ddot{z}_2 = k_{21}(z_2 - q_2) + c_2(\dot{z}'_2 - \dot{z}_2) + k_{22}(z'_2 - z_2) + f_{d2} + F_{r_r} + m_2 g$$

$$m_3\ddot{z}_3 = k_{12}(z_1 - z'_1) + c_1\left(\dot{z}_1 - \dot{z}'_1\right) + c_2\left(\dot{z}_2 - \dot{z}'_2\right) + k_{22}(z'_2 - z_2) - f_{d1} - f_{d2} - F_{r_f} - F_{r_r} + m_3 g$$

$$J\ddot{\varphi} = -\left[k_{12}(z_1 - z'_1) + c_1\left(\dot{z}_1 - \dot{z}'_1\right)\right] l_f + \left[c_2\left(\dot{z}_2 - \dot{z}'_2\right) + k_{22}(z'_2 - z_2)\right] l_r - l_f f_{d1} + l_f F_{r_f} + l_r f_{d2} - l_r F_{r_r}$$

Where:

$$z'_1 = z_3 - \varphi l_f$$

$$z'_2 = z_3 + \varphi l_r$$

3.2.1.3 (C) ½ MODEL, 4-DOFs (ROLL) SUSPENSION SYSTEM

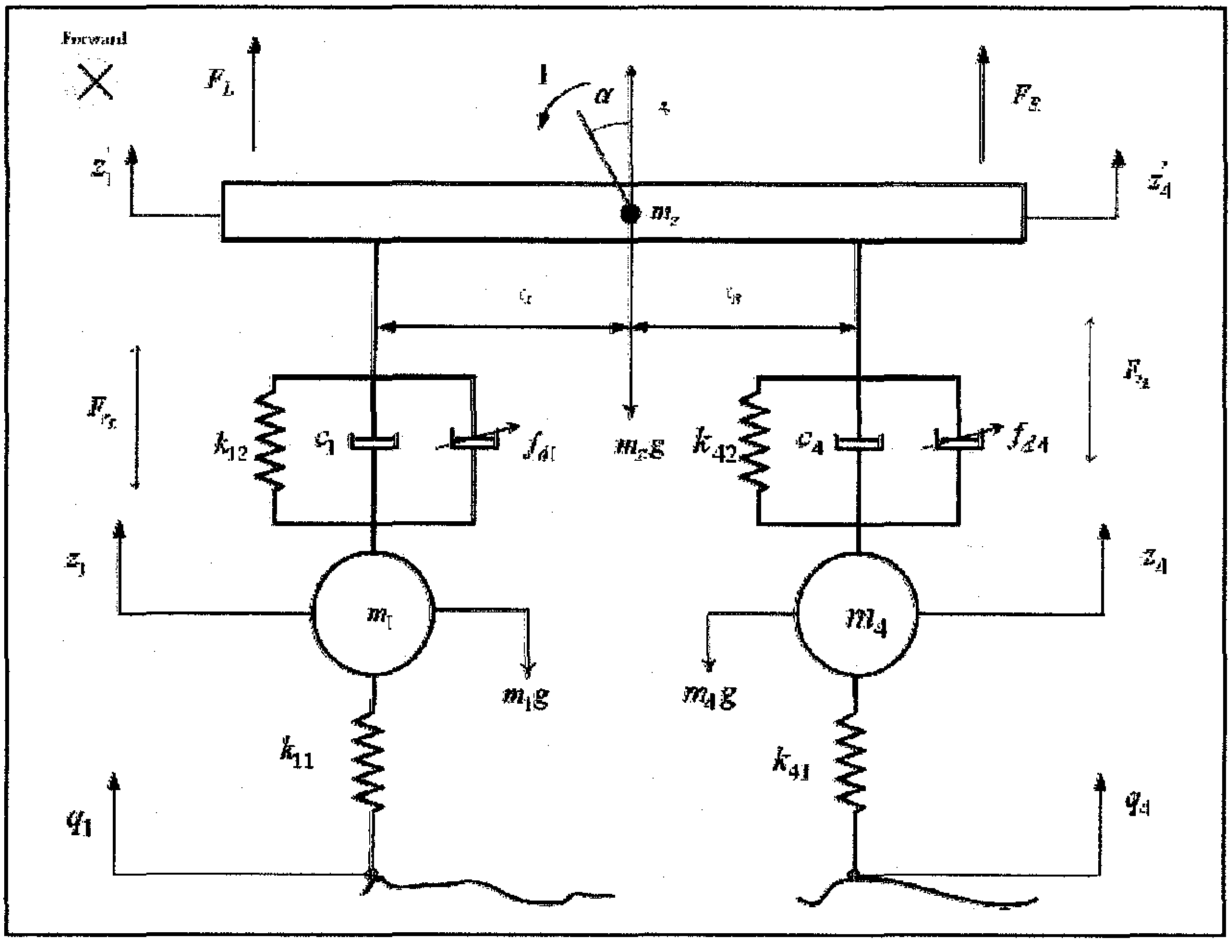


Figure 24 : 1/2 model (roll)

$$m_1\ddot{z}_1 = k_{11}(z_1 - q_1) + c_1(\dot{\acute{z}}_1 - \dot{z}_1) + k_{12}(\acute{z}_1 - z_1) + f_{d1} + F_{r_L} + m_1 g$$

$$m_4\ddot{z}_4 = k_{41}(z_4 - q_4) + c_4(\dot{\acute{z}}_4 - \dot{z}_4) + k_{42}(\acute{z}_4 - z_4) + f_{d4} + F_{r_R} + m_4 g$$

$$m_z\ddot{z}_z = k_{12}(z_1 - \acute{z}_1) + c_1\left(\dot{z}_1 - \dot{\acute{z}}_1\right) + c_4\left(\dot{z}_4 - \dot{\acute{z}}_4\right) + k_{42}(\acute{z}_4 - z_4) - f_{d1} - f_{d4} - F_{r_L} - F_{r_R} + m_z g$$

$$I\ddot{\alpha} = -\left[k_{12}(z_1 - \acute{z}_1) + c_1\left(\dot{z}_1 - \dot{\acute{z}}_1\right)\right] l_L + \left[c_4\left(\dot{z}_4 - \dot{\acute{z}}_4\right) + k_{42}(\acute{z}_4 - z_4)\right] l_R - l_L f_{d1} + l_L F_{rL} + l_r f_{d4} - l_R F_{r_R}$$

Where:

$$\acute{z}_1 = z_z - \alpha l_L$$

$$\acute{z}_4 = z_z + \alpha l_R$$

3.2.1.3 (C) FULL MODEL, 7-DOFs SUSPENSION SYSTEM

In the final stage of realizing the 7-DOFs vehicle suspension system, an integration of the 2-Dof and 4-Dof are necessary to have a clear view of the overall performance. The 7-Dof vehicle suspension system can be easily represented by a diagram depicting the architecture of full scale model of a vehicle as follows.

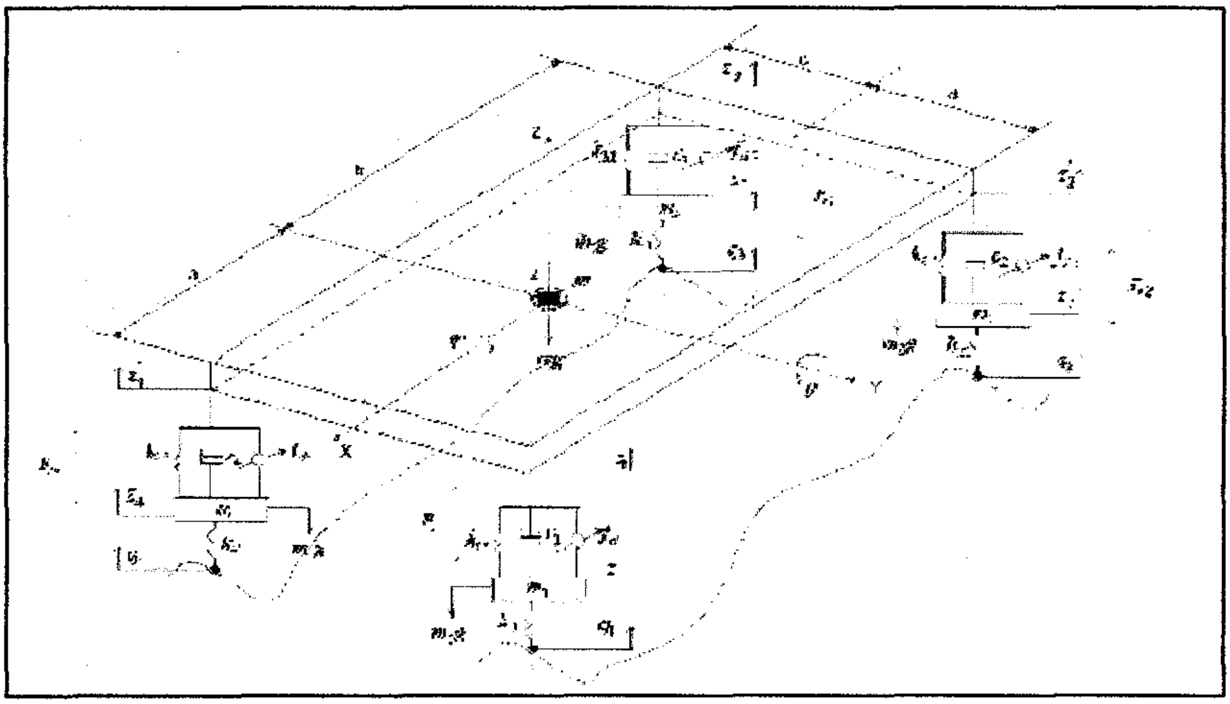


Figure 25 : Full model 7-DOF's suspension system

The basic idea of the full scale model will be consists of integrations of different DOFs which supposedly making up the whole picture of the vehicle. As can be observed from the above diagram, the familiar 2-DOFs composition and as well as for the 4-DOFs arrangement were merged altogether in order to produce the final respective design. This time around, the overall dynamic formula is to be taking into account to represent the full scale model of the 7-DOFs vehicle suspension system. In accordance to the 2-DOFs and not to mention the 4-DOFs which implies both of the roll and pitch will all be taken into consideration.

Therefore, the dynamic formula will be:

$$m_1\ddot{z}_1 = k_{11}(z_1 - q_1) + c_1(\dot{z}_1' - \dot{z}_1) + k_{12}(z_1' - z_1) + f_{d1} + F_{r1} + m_1 g$$
$$m_2\ddot{z}_2 = k_{21}(z_2 - q_2) + c_2(\dot{z}_2' - \dot{z}_2) + k_{22}(z_2' - z_2) + f_{d2} + F_{r2} + m_2 g$$
$$m_3\ddot{z}_3 = k_{31}(z_3 - q_3) + c_3(\dot{z}_3' - \dot{z}_3) + k_{32}(z_3' - z_3) + f_{d3} + F_{r3} + m_3 g$$
$$m_4\ddot{z}_4 = k_{41}(z_4 - q_4) + c_4(\dot{z}_4' - \dot{z}_4) + k_{42}(z_4' - z_4) + f_{d4} + F_{r_R} + m_4 g$$
$$m_z\ddot{z}_z = k_{12}(z_1 - z_1') + k_{22}(z_2 - z_2') + k_{32}(z_3 - z_3') + k_{42}(z_4 - z_4') + c_1(\dot{z}_1 - \dot{z}_1') + c_2(\dot{z}_2 - \dot{z}_2') + c_3(\dot{z}_3 - \dot{z}_3') + c_4\left(\dot{z}_4 - \dot{z}_4'\right) - f_{d1} - f_{d2} - f_{d3} - f_{d4} - F_{r_L} - F_{r1} - F_{r2} - F_{r3} - F_{r4} + mg$$
$$J_x\ddot{\varphi} = -\left[k_{32}(z_3 - z_3') + c_3(\dot{z}_3 - \dot{z}_3') + k_{42}(z_4 - z_4') + c_4\left(\dot{z}_4 - \dot{z}_4'\right)\right]c + \left[k_{12}(z_1 - z_1') + c_1(\dot{z}_1 - \dot{z}_1') + k_{22}(z_2 - z_2') + c_2(\dot{z}_2 - \dot{z}_2')\right]d - (f_{d3} + f_{d4})c - (F_{r3} + F_{r4})c + (f_{d1} + f_{d2})d - (F_{r1} + F_{r2})d$$
$$J_y\ddot{\theta} = -\left[k_{12}(z_1 - z_1') + c_1(\dot{z}_1 - \dot{z}_1') + k_{42}(z_4 - z_4') + c_4\left(\dot{z}_4 - \dot{z}_4'\right)\right]a + \left[k_{22}(z_2 - z_2') + c_2(\dot{z}_2 - \dot{z}_2') + k_{32}(z_3 - z_3') + c_3(\dot{z}_3 - \dot{z}_3')\right]b - (f_{d1} + f_{d4})a + (F_{r1} + F_{r4})a + (f_{d2} + f_{d3})b - (F_{r2} + F_{r3})b$$

Where:

$$z_1' = z - (a\theta + d\varphi)$$
$$z_2' = z + (b\theta - d\varphi)$$
$$z_3' = z + (b\theta + c\varphi)$$
$$z_4' = z + (a\theta - c\varphi)$$

F_{r1} is a constant friction for 1-front left half suspension

F_{r2} is a constant friction for 2-rear left half suspension

F_{r3} is a constant friction for 3-rear right half suspension

F_{r4} is a constant friction for 4-front right half suspension

3.3 TOOL

Utilized so far up to this phase:

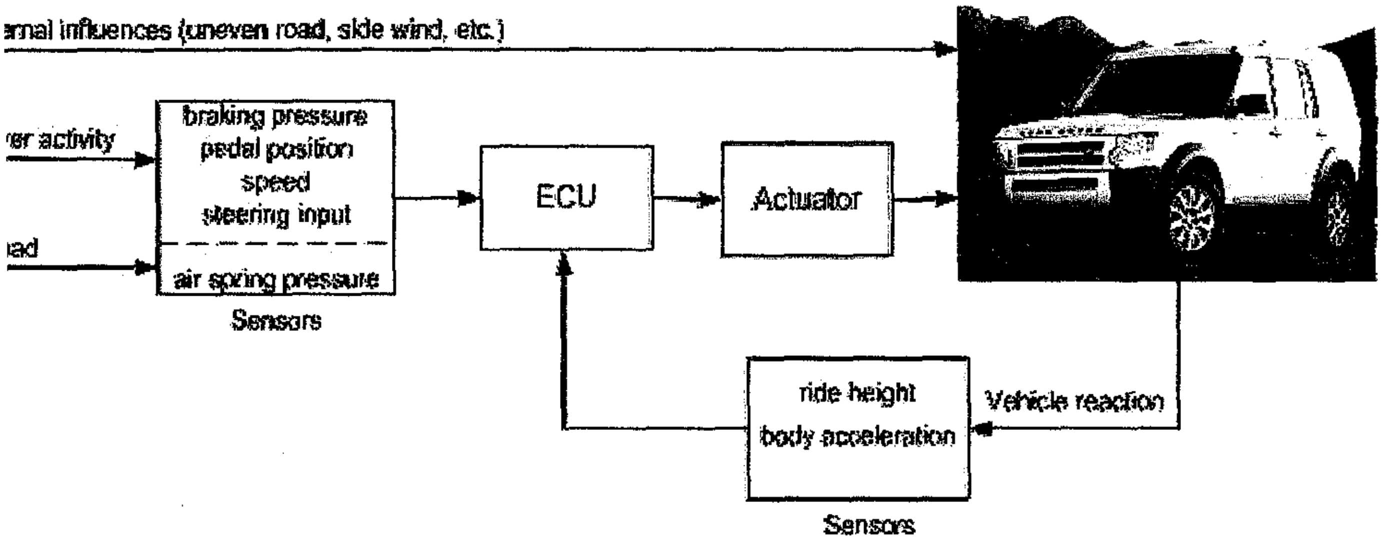
1. MATLAB language of technical computing, The Mathworks Inc
2. Microsoft Office

CHAPTER 4: RESULTS AND DISCUSSIONS

4.0 RESULTS

In accordance to the timeline that I and Dr Vu have agreed to, I have managed to run through a series of activities which had been planned from the early stage of this project. Listed down are the results and followed by the discussions which reflect the findings that I have encountered all through the process of designing a suspension system with a PID controller for a full model of a car.

The Schematic of The Active Suspension System



The basic schematic diagram of an active suspension system as being agreed

Sensor: Displacement of the body of the car will be detected by a suspension deflection sensor

ECU : Programmed matlab and simulink

Actuator: Magnetorheological damper

4.1 7-DOFs VEHICLE SUSPENSION SYSTEM

For the results author will represent the simulation in Simulink MatLAB and this implies in sequencing order where it will started from the 2-DOFs, 4-DOFs and finally the 7-DOFs vehicle suspension system. In such cases, the PID controller for each of the DOFs will be accompanied in order to meet the objective of the project.

In accordance with the derived formula of dynamic equations for the 7-DOFs vehicle suspension system, author started to design the system and the simulation testing will followed up upon the completion. The whole idea of building up the system is by working on the Simulink by referring to the full model of the vehicle and at the same time to make sure the dynamic formula derived for the 7-DOFs is mathematically tally with the model.

For the whole process of developing the system to the final stage of the completed 7-DOFs vehicle suspension system, author have to do it in such way that it is constructed according to the planned step-by-step from the very beginning of the process until the final architecture of the system.

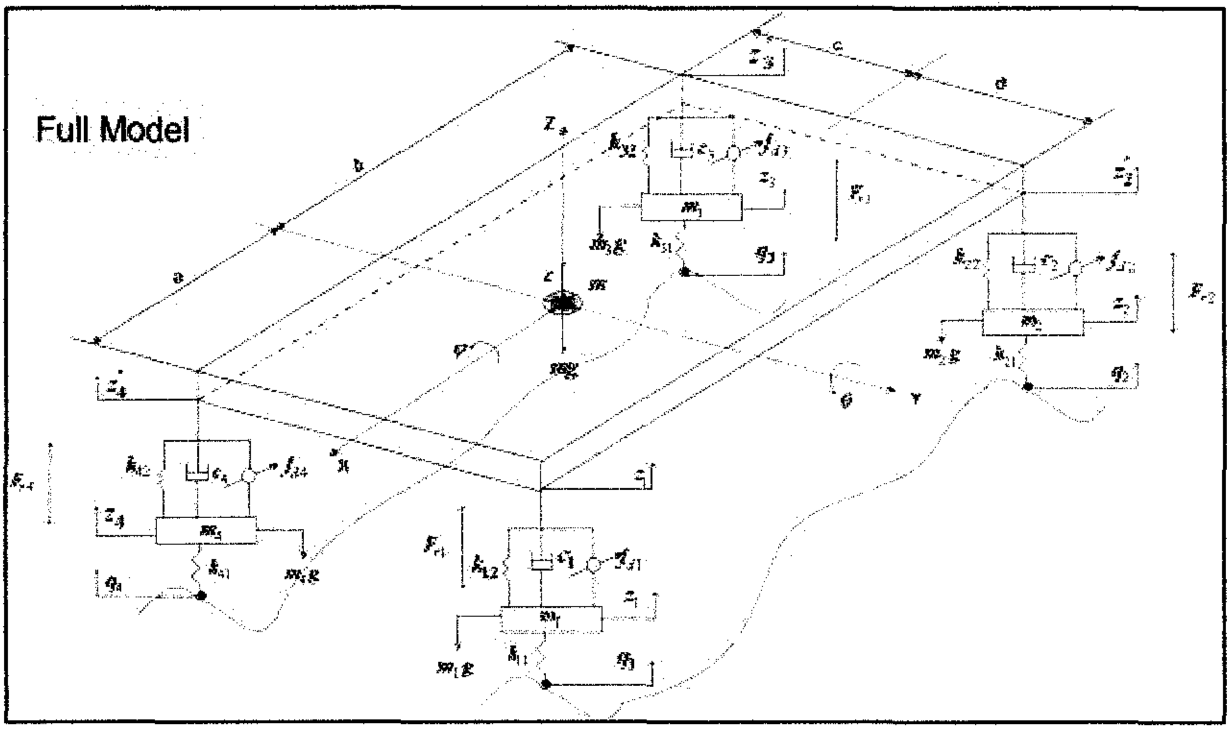


Figure 26 : Full model of 7-DOFs suspension system

To further simplify, author start over by following the underlined dynamic equation.

$$m_1\ddot{z}_1 = k_{11}(z_1 - q_1) + c_1(\dot{\acute{z}}_1 - \dot{z}_1) + k_{12}(\acute{z}_1 - z_1) + f_{d1} + F_{r1} + m_1 g \text{ -------- } (1)$$
$$m_2\ddot{z}_2 = k_{21}(z_2 - q_2) + c_2(\dot{\acute{z}}_2 - \dot{z}_2) + k_{22}(\acute{z}_2 - z_2) + f_{d2} + F_{r2} + m_2 g \text{ -------- } (2)$$
$$m_3\ddot{z}_3 = k_{31}(z_3 - q_3) + c_3(\dot{\acute{z}}_3 - \dot{z}_3) + k_{32}(\acute{z}_3 - z_3) + f_{d3} + F_{r3} + m_3 g \text{ ------- } (3)$$
$$m_4\ddot{z}_4 = k_{41}(z_4 - q_4) + c_4(\dot{\acute{z}}_4 - \dot{z}_4) + k_{42}(\acute{z}_4 - z_4) + f_{d4} + F_{r_R} + m_4 g \text{ ------- } (4)$$

Author will first started the Simulink from the 1st equation and follow up with the next equations.

<u>First Equation</u>

$$m_1\ddot{z}_1 = k_{11}(z_1 - q_1) + c_1(\dot{\acute{z}}_1 - \dot{z}_1) + k_{12}(\acute{z}_1 - z_1) + f_{d1} + F_{r1} + m_1 g$$

Simulink

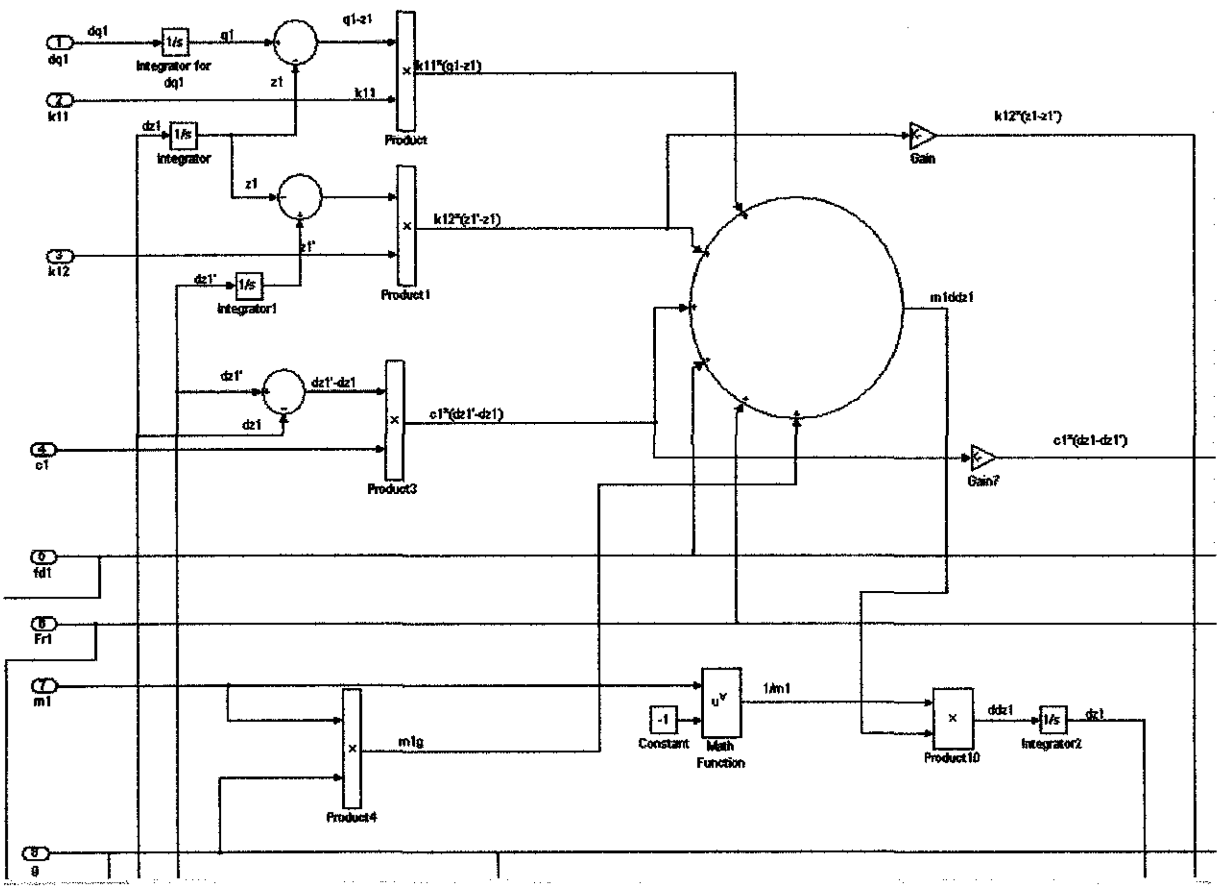


Figure 27 : Simulink of the front-left-car dynamic equations

Second Equation

$$m_2\ddot{z}_2 = k_{21}(z_2 - q_2) + c_2(\dot{\acute{z}}_2 - \dot{z}_2) + k_{22}(\acute{z}_2 - z_2) + f_{d2} + F_{r2} + m_2 g$$

Simulink

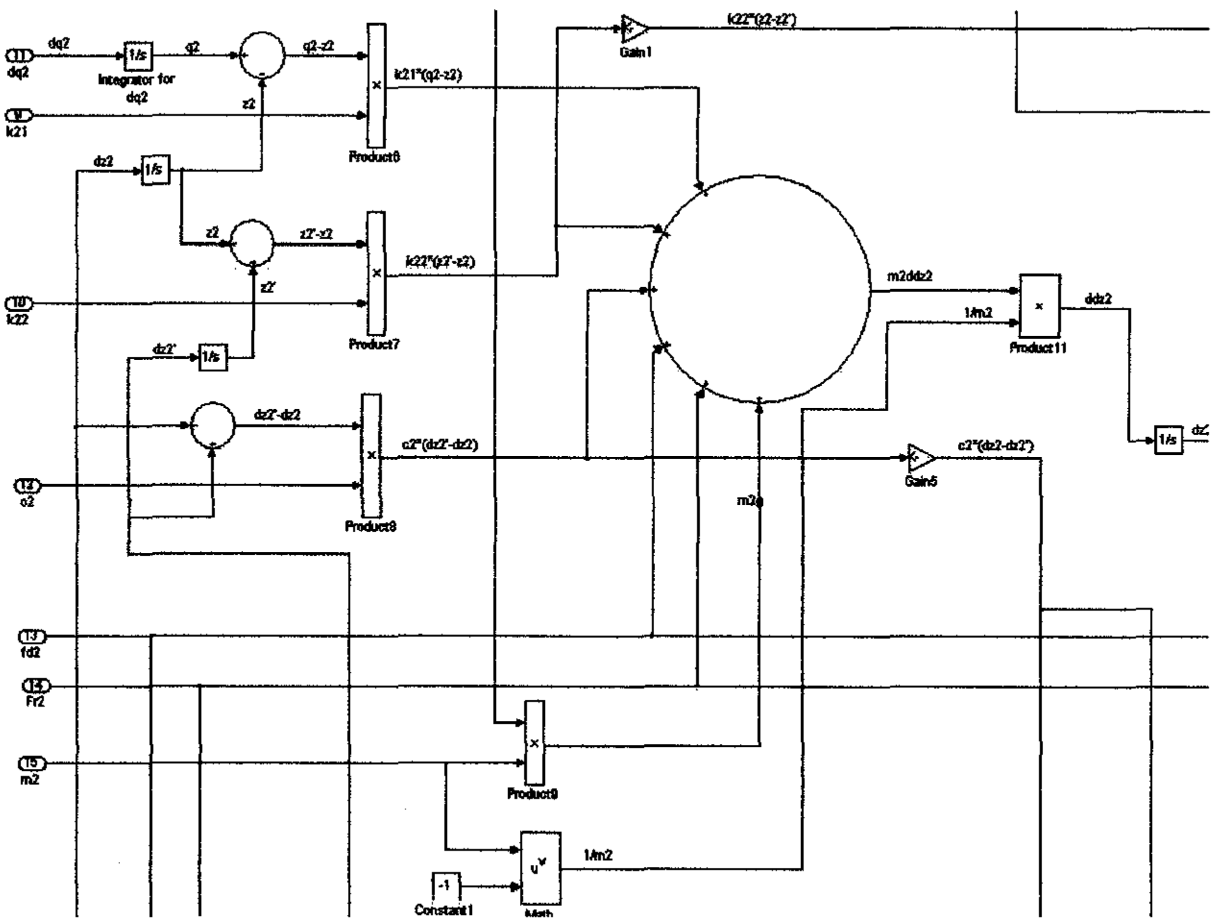


Figure 28 : Simulink for the rear-left-car dynamic equation

Third Equation

$$m_3\ddot{z}_3 = k_{31}(z_3 - q_3) + c_3(\dot{z}_3' - \dot{z}_3) + k_{32}(z_3' - z_3) + f_{d3} + F_{r3} + m_3 g$$

Simulink

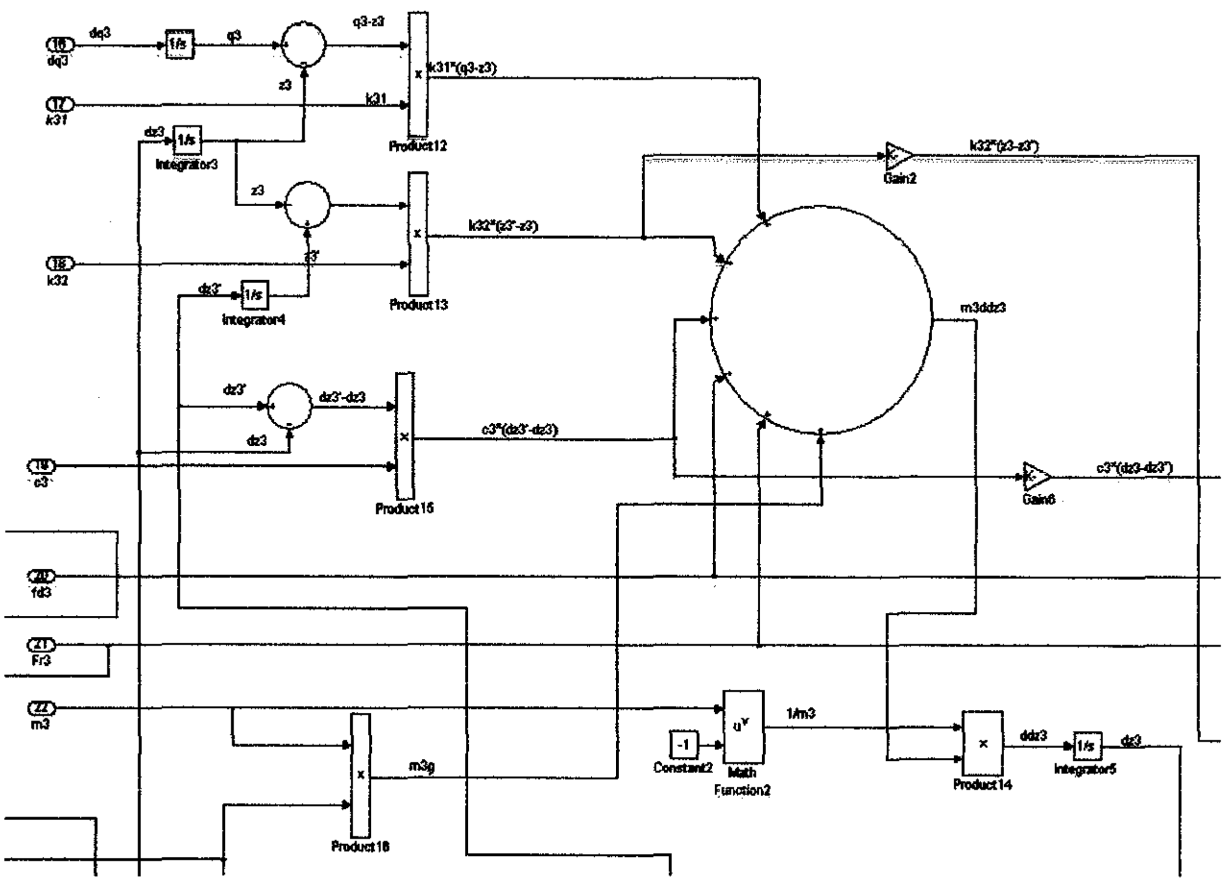


Figure 29 : Simulink for the rear-right-car dynamic equation

Fourth Equation

$$m_4\ddot{z}_4 = k_{41}(z_4 - q_4) + c_4(\dot{\acute{z}}_4 - \dot{z}_4) + k_{42}(\acute{z}_4 - z_4) + f_{d4} + F_{r_R} + m_4 g$$

Simulink

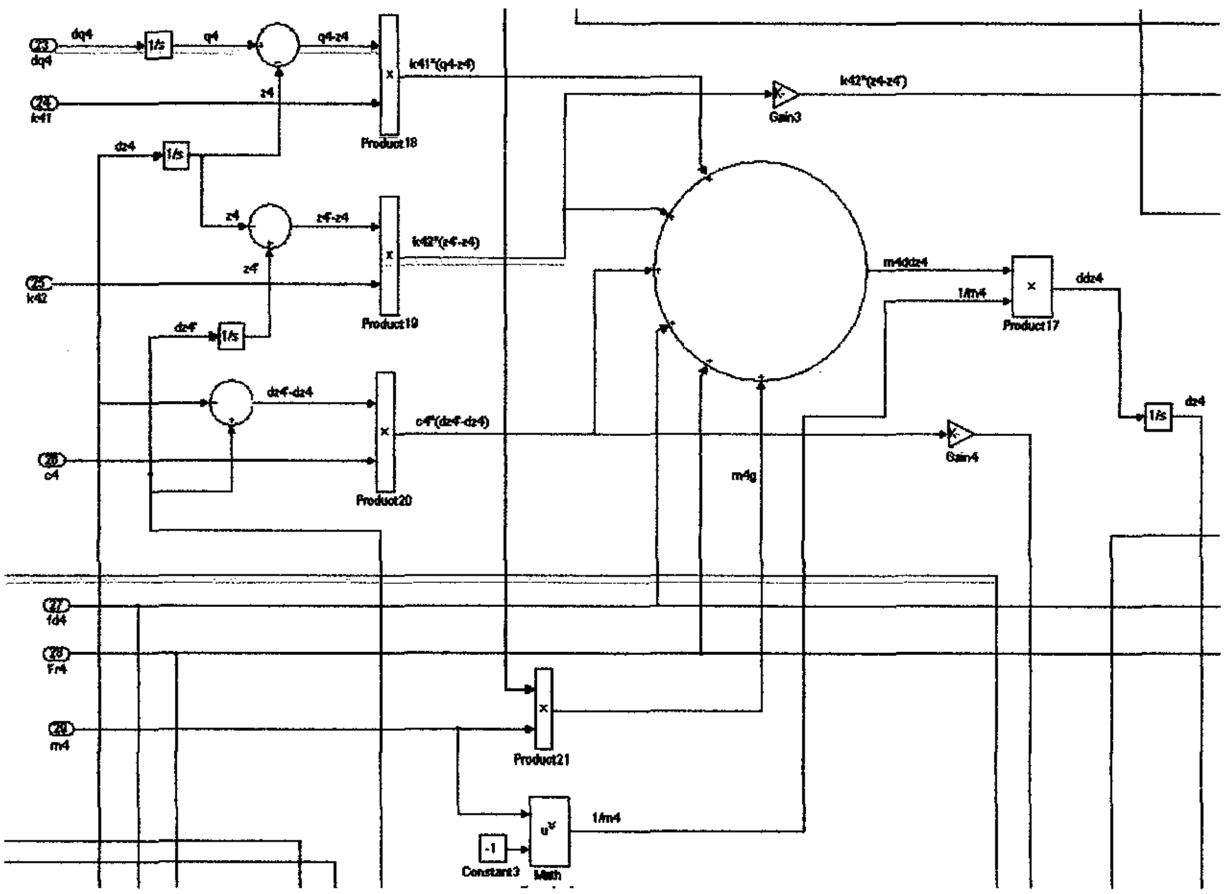


Figure 30 : Simulink of the front-right-car dynamic equation

Next author proceed with the rest of the equations which are the fifth equation, sixth equation and finally the seventh equation that will make up the whole picture of the 7-DOFs vehicle suspension system. The fifth equation is considered as the mass of the vehicle and it is representing overall mass that will be taken into account to further carry on the project.

<u>Fifth Equation</u>

$$m_z\ddot{z}_z = k_{12}(z_1 - \acute{z}_1) + k_{22}(z_2 - \acute{z}_2) + k_{32}(z_3 - \acute{z}_3) + k_{42}(z_4 - \acute{z}_4) + c_1(\dot{z}_1 - \dot{z}_1)$$
$$+ c_2(\dot{z}_2 - \dot{z}_2) + c_3(\dot{z}_3 - \dot{z}_3) + c_4\left(\dot{z}_4 - \dot{z}_4\right) - f_{d1} - f_{d2} - f_{d3} - f_{d4} - F_{r_L} - F_{r1}$$
$$- F_{r2} - F_{r3} - F_{r4} + mg$$

<u>Sixth Equation</u>

$$J_x\ddot{\varphi} = -\left[k_{32}(z_3 - \acute{z}_3) + c_3(\dot{z}_3 - \dot{z}_3) + k_{42}(z_4 - \acute{z}_4) + c_4\left(\dot{z}_4 - \dot{z}_4\right)\right]c$$
$$+ \left[k_{12}(z_1 - \acute{z}_1) + c_1(\dot{z}_1 - \dot{z}_1) + k_{22}(z_2 - \acute{z}_2) + c_2(\dot{z}_2 - \dot{z}_2)\right]d - (f_{d3} + f_{d4})c$$
$$- (F_{r3} + F_{r4})c + (f_{d1} + f_{d2})d - (F_{r1} + F_{r2})d$$

<u>Seventh Equation</u>

$$J_y\ddot{\theta} = -\left[k_{12}(z_1 - \acute{z}_1) + c_1(\dot{z}_1 - \dot{z}_1) + k_{42}(z_4 - \acute{z}_4) + c_4\left(\dot{z}_4 - \dot{z}_4\right)\right]a$$
$$+ \left[k_{22}(z_2 - \acute{z}_2) + c_2(\dot{z}_2 - \dot{z}_2) + k_{32}(z_3 - \acute{z}_3) + c_3(\dot{z}_3 - \dot{z}_3)\right]b - (f_{d1} + f_{d4})a$$
$$+ (F_{r1} + F_{r4})a + (f_{d2} + f_{d3})b - (F_{r2} + F_{r3})b$$

The sixth and seven equations will be representing the 4-DOFs which consist of both roll and pitch. Individually both of the architecture will have to integrate along with the rest of other individual equations and a complete system of a full scale vehicle suspension system can be simulated as being planned.

<u>Fifth Equation</u>

$$m_z\ddot{z}_z = k_{12}(z_1 - \acute{z}_1) + k_{22}(z_2 - \acute{z}_2) + k_{32}(z_3 - \acute{z}_3) + k_{42}(z_4 - \acute{z}_4) + c_1(\dot{z}_1 - \dot{z}_1)$$
$$+ c_2(\dot{z}_2 - \dot{z}_2) + c_3(\dot{z}_3 - \dot{z}_3) + c_4\left(\dot{z}_4 - \dot{z}_4\right) - f_{d1} - f_{d2} - f_{d3} - f_{d4} - F_{r_L} - F_{r1}$$
$$- F_{r2} - F_{r3} - F_{r4} + mg$$

Simulink

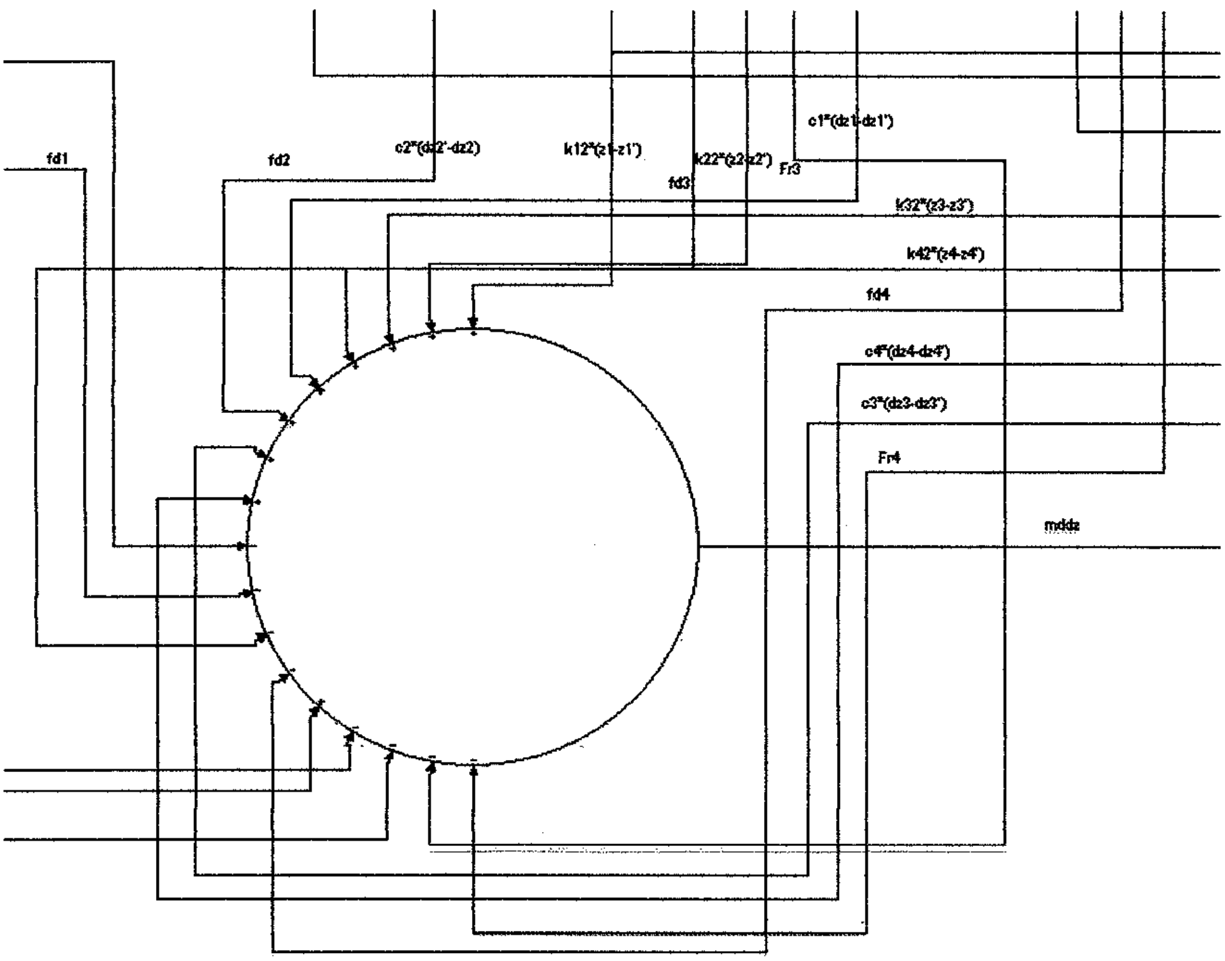


Figure 31 : Simulink of the vehicle overall mass

Sixth Equation

$$J_x\ddot{\varphi} = -\left[k_{32}(z_3 - \acute{z}_3) + c_3(\dot{z}_3 - \dot{\acute{z}}_3) + k_{42}(z_4 - \acute{z}_4) + c_4\left(\dot{z}_4 - \dot{\acute{z}}_4\right)\right]c + \left[k_{12}(z_1 - \acute{z}_1) + c_1(\dot{z}_1 - \dot{\acute{z}}_1) + k_{22}(z_2 - \acute{z}_2) + c_2(\dot{z}_2 - \dot{\acute{z}}_2)\right]d - (f_{d3} + f_{d4})c - (F_{r3} + F_{r4})c + (f_{d1} + f_{d2})d - (F_{r1} + F_{r2})d$$

(a) $\left[k_{32}(z_3 - \acute{z}_3) + c_3(\dot{z}_3 - \dot{\acute{z}}_3) + k_{42}(z_4 - \acute{z}_4) + c_4\left(\dot{z}_4 - \dot{\acute{z}}_4\right)\right] c$

(b) $\left[k_{12}(z_1 - \acute{z}_1) + c_1(\dot{z}_1 - \dot{\acute{z}}_1) + k_{22}(z_2 - \acute{z}_2) + c_2(\dot{z}_2 - \dot{\acute{z}}_2)\right] d$

(c) $(f_{d3} + f_{d4})c$

(d) $(F_{r3} + F_{r4})c$

Simulink

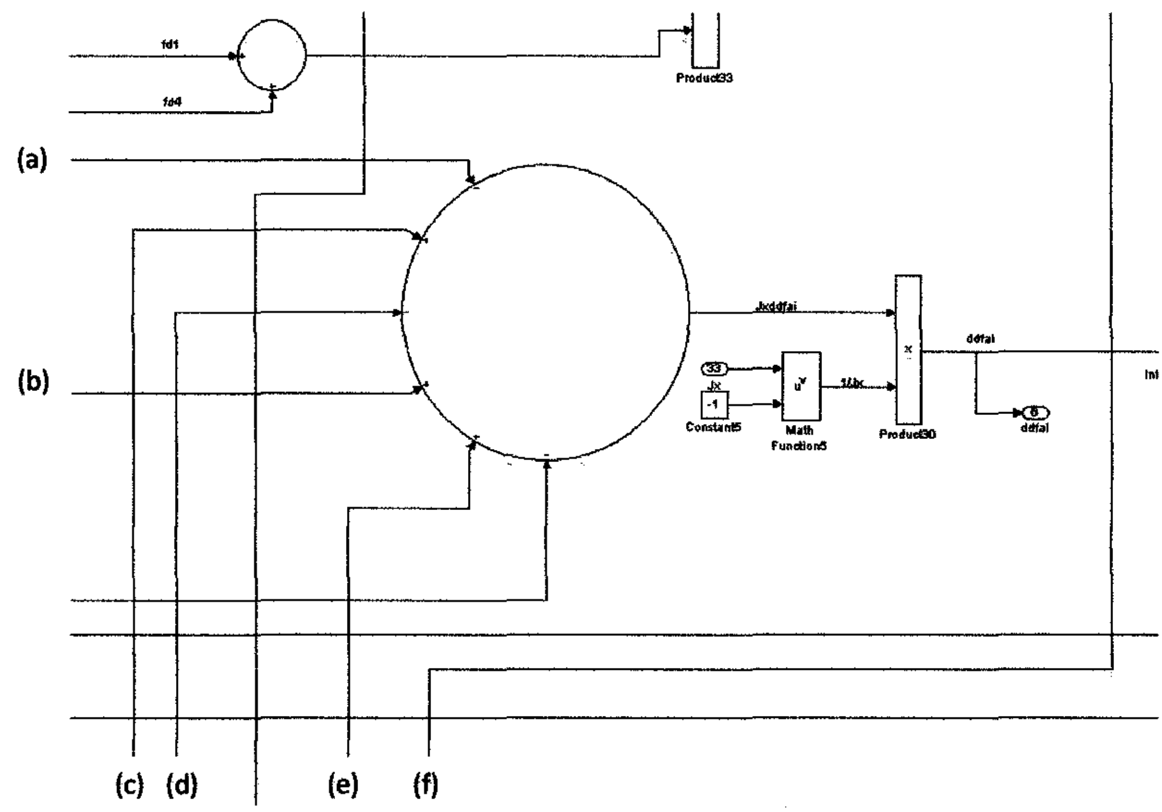


Figure 32 : Simulink of the 4-dof pitch vehicle suspension system

Simulink for Seventh Equation

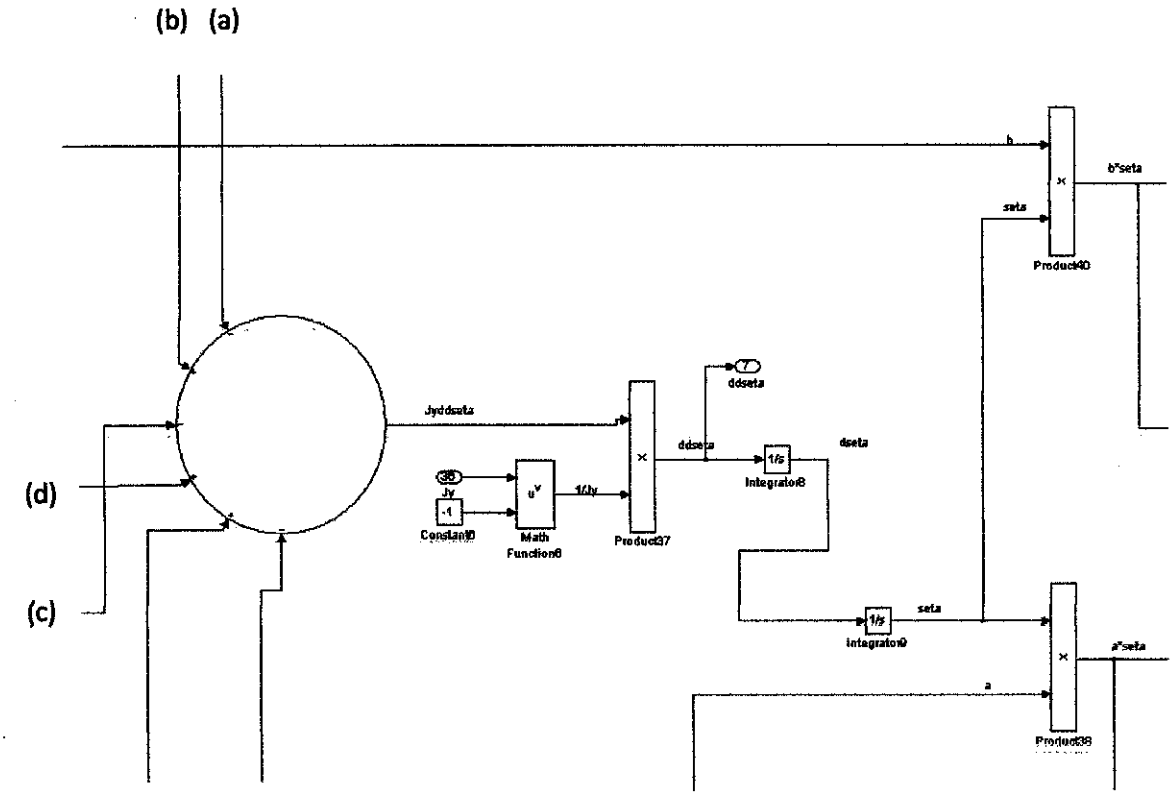


Figure 33 : Simulink of the 4-dof roll suspension system

CHAPTER 5: CONCLUSION

Therefore, to mention again that the roads are commonly used by motor vehicles are uneven. And this unevenness causes vertical movements of the vehicle and the passengers during the driving process. To overcome this, an approach to improve the overall performance of automotive vehicle is taken by implying active elements for the suspension system which alter only the vertical force reactions of the suspensions. In other words, by designing and simulate an active PID controller for a vehicle suspension system via controllable damping devices. The current progress of this project hopefully will help a lot and contributing towards the completion of the project. After following accordingly the flow of the project, the 7-DOFs vehicle suspension system can finally be tested with an active PID controller.

GANTT CHART

No	Detail/Week	1	2	3	4	5	6	7	8	9		10	11	12	13	14
1	Project work continues										Mid-Semester Break					
2	Submissions of progress report 1															
3	Project work continues															
4	Submissions of progress report 2															
5	Seminar (compulsory)															
6	Project work continues															
7	Poster exhibition															
8	Submission of Dissertation Final Draft															
9	Oral presentation											During study week				
10	Submission of dissertation (hardbound)											7 days after oral presentation				

References

Bolton, W. (2003). *Mechatronics, electronic control systems in mechanical and electrical engineering, third edition.* England, UK: Pearson Prentice Hall.

Chart, C., Kersha, J.F. & Ed, D. (2007). *Classroom manual: Automotive steering, suspension, and wheel alignment: third edition.* Harlow, UK: Pearson Prentice-Hall.

Gillespie , T.D. (1992). *Fundamentals of vehicle dynamics.* Washington, USA: SAE International .

Minh, V.T. (2008). *Advance vehicle dynamics.* Kuala Lumpur, MY: Petronas University of Technology Press.

Prof. Messner, B. (1997). PID tutorial. Retrieved February 10, 2009, from Control tutorials for matlab Web site:http://www.engin.umich.edu/group/ctm/PID/PID.html

site: opus.**bath**.ac.uk/1882/