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***An investigation into the accuracy of
mathematical models against experimental
verification***

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'This thesis is the result of the author's original research. It has been composed by the author and has not been previously submitted for examination which has lead to the award of a degree.'

Signed:

Date:

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1 Abstract

This paper presents the results of experiments performed on a VFR400 race motorcycle. The experiments were conducted using accelerometers on the sprung and unsprung masses. Models made in Simulink for a quarter car, half car and half car with pitch were developed. Unlike other models, the damping of the tyres was included in the model to observe how the oscillation decayed. The experimental results were compared to the models with close relationships being found.

The motorcycle was ridden over a specially designed step so that the road input or system excitation could be determined with the accelerometers recording the accelerations of critical areas of the motorcycle

Although issues were encountered during experimentation and the full data was not useable, the first oscillation allowed comparison for accelerations, velocities and displacements. As well as these parameters, the experimental data was used to determine the damping for the sprung and unsprung masses.

It was found that even simplified in plane dynamics models can give an insight into the behaviour of the suspension characteristics. The ability to determine the damping allows the potential for active suspension systems to be considered for further research

2 Introduction

Much research has gone into motorcycle dynamics in recent years due to the availability of technology to improve safety of motorcycles. No longer is the dynamics of a motorcycle just about how fast a lap time can be achieved, although in some instances this is still the goal. In the academic community, complex multibody codes and models have been produced which can isolate the vibration modes of a motorcycle and allow changes to be quickly analysed. Little research has been done to verify the output of even relatively simple models such as the four degree of freedom half car model.

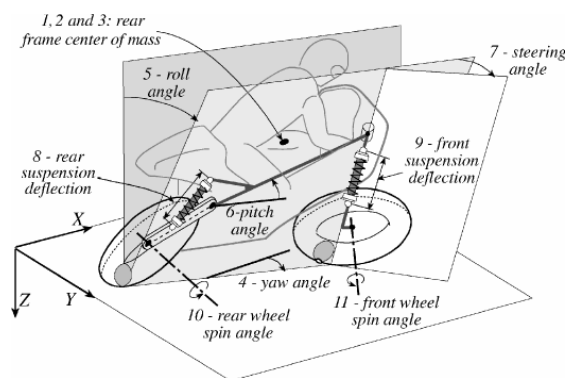


Figure 1 - Cossalter, V (2009), eleven degree of freedom model

In this paper, research into combining both mathematical modelling and experimental testing will be conducted. Testing of a race ready VFR400 will be conducted using accelerometers to attain data about the motion of the body and wheels of the motorcycle. This will then be compared to various mathematical models created to determine firstly, if mathematical models are accurate enough to be used to partially replace testing of motorcycles. Secondly, which of the three models created (quarter car, half car and half car with pitch) is the most suitable in terms of accuracy, speed, ease of implementation and simplicity.

The above aims will be analysed by means of comparison of the accelerations, velocities

and displacements. A novel method of determining the damping settings on the motorcycle for use in the model is implemented to increase the accuracy of the models. This was performed to attempt to get the experimental results and model output as similar as possible.

3 Research

3.1 Degrees of Freedom (DOF)

Degrees of freedom (DOF) are an important aspect of defining physical, dynamical systems. Degrees of freedom can be described as “the number of independent parameters that are needed to specify the configuration of a system”. In the context of motorcycles, there are 6 spatial degrees of freedom with another 5 that are generally considered. These can be seen in the eleven degree of freedom model shown in figure 1 (Cossalter and Lot, 2002).

As can be seen in figure 1, 6 of the spatial DOF are those that are experienced in everyday life. These are linear motion in the X,Y and Z axes and the rotation about each axis.

However, in motorcycle dynamics, as the entire motorcycle is divided into smaller, interrelated systems and mass bodies, the number of DOF increases.

This paper will focus on a model that can move in the X and Z spatial DOF with pitch included. This leaves a four DOF model that includes the front and rear unsprung masses, the sprung mass motion, and includes the pitch DOF. Two of the degrees of freedom for rotational motion (Yaw and roll) have been neglected, as well as the steer angle. These have been neglected due to the increased complexity it introduces with out of plane motion.

3.2 Mass – Spring Dampers

When modelling a dynamical system, it can be useful to remove the complexity of certain aspects. For example, removing the rake angle and linkage ratio of a motorcycle and considering both as vertical mass spring damper systems. This allows faster computation without large errors. If more accuracy is required, then data obtained from measurements or models can be inputted to the vertical model as a correction.

It is possible to convert the rake angle to a vertical system by using the following equation;

Equation 1– Resolving Rake angle to vertical

$$K' = K * \frac{1}{(\cos(\phi))^2}$$

Where;

K' = Vertical Spring Rate

K = Spring Rate of Fitted Springs

ϕ = Rake Angle

This can also be modified to consider the rear suspension as a vertical system;

Equation 2 - Resolving Linkage Ratio to Vertical

$$K' = K * \frac{1}{(LR^2)}$$

Where 'LR' signifies the linkage ratio of the rear suspension system. Sharp (2004, p. 20) explains that rear suspension "uses a single spring/damper unit with a mechanical linkage connection to the swinging arm." page number

However, although this is the case for most motorcycles, and the linkage ratio can change depending on rear shock absorber extension

and angle, for complete accuracy, convoluted methods are required to consider all aspects of the rear motorcycle geometry. For this reason, the simplification of a single average value is used, with error analysis conducted to verify the simplification as justifiable.

Once these two simplifications have been applied, the motorcycle can be considered as a single degree of freedom model as shown in figure 2.

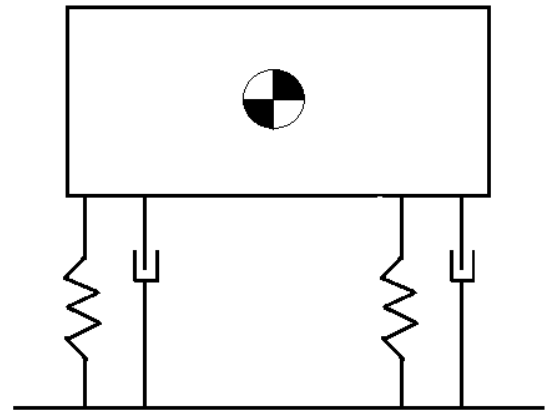


Figure 2- 1 DOF Mass Spring Damper System

This is the simplest model that takes into account the entire motorcycle. However, it can be modified to consider the pitch characteristics, making it a 2 DOF system. This is due to the option of the front and rear spring and damper values being different, causing an unbalanced moment around the Centre of Gravity (COG).

Another option for a 2 DOF system is to consider the sprung mass as two masses. This can be achieved through inserting another mass below the sprung mass and considering the wheel as just a spring, or "equivalent to a parallel combination of spring and a damper" (Bhanu, 2009, p. 3) This can also be considered true for the sprung suspension system.

It is now possible to include the wheel mass and its response to road inputs. This modification can then be applied in two ways resulting in a model with up to 4 degrees of freedom. The options for the models will be

discussed in finer detail from section 3.7 onwards.

3.3 Motorcycle Dynamics

Motorcycle dynamics can usually be classified into two forms, in and out of plane. In-plane rectilinear motion will be the focus of this paper, with a brief discussion on out-of-plane dynamics.

When considering only the in-plane dynamics, it is mainly the motorcycle suspension that is being analysed. In most cases, this can be reduced to the mass spring damper simplification. However, there are other factors to consider other than the spring rates and damping coefficients. These include the magnitude of the sprung and unsprung masses, rake, trail, triple clamp offset, linkage ratio, swingarm length, rear shock absorber angle, position of COG, wheelbase and wheel radii. The rake and linkage ratio for example, has an important effect on the forces experienced by the shock absorbers, be it front or rear. As was shown in equations 1 and 2, the values of the rake and trail have an important effect on the vertical spring rates.

It is possible to calculate how these factors and other previously mentioned specifications, have an impact on the suspension position. Most of the specifications such as rake and trail, are a single average value. However, under dynamic conditions, they themselves become dynamic and change with suspension position etc. These calculations can be implemented into the main calculations. However, due to the small amount of accuracy it would give the experiment, it is still small compared to other errors incurred such as the accuracy of the sensors. For this reason, the calculations for dynamic rake and trail have been neglected.

Free oscillating systems use the assumptions that the input is modelled as an initial

condition, which instantaneously stops at the start of measurement. However, for most conditions experienced by a motorcycle, the input will be a time dependent force. When considering motorcycle dynamics, it is important to examine any external inputs into the system, for example the road, as 'an external excitation to the system' (Gandhi, Adarsh and Ramachandran, 2017, p.4) This changes the system from being a free oscillatory body to a fully dynamic system, with a force inputted which changes with respect to time.

3.4 Vibration Modes

When vibration problems are analysed, it is useful to consider them as simple harmonic oscillators. This is due to the simplicity of the equations, and the ability to determine frequencies relatively easily. When considering complex dynamical systems such as a motorcycle, even in rectilinear motion, it can become much more difficult to attain these frequencies. This problem is more pronounced when attempting to find the frequencies of single or groups of components. This is due to them have more than one factor influencing their behaviour in terms of the interaction from the rest of the system.

(R.S.Sharp, 1971, p. 2) investigated this problem, modelling the motorcycle as "two rigid frames, joined at the steering axis with freedom". This relatively simple model allowed calculation of the wobble and weave oscillatory modes, as well as the capsize motion. Research conducted by Cossalter et al., (2009) shows that a larger number of oscillatory modes have been found by using more recent, advanced multibody models. Shown in figure 3 is a plot of 12 different

oscillatory modes, found by Cossalter et al., (2009).

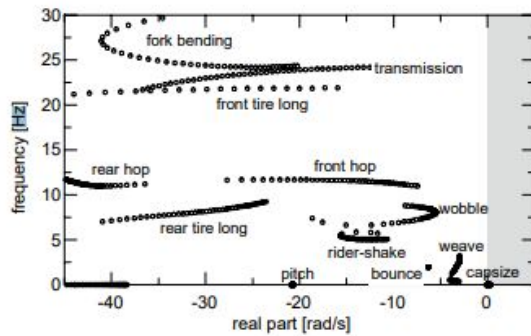


Figure 3- Root Locus Plot Of Motorcycle Vibration Modes

Vibration Mode	Frequency
Fork bending	25 – 20 Hz
Transmission	22- 25 Hz
Front tire longitudinal	20 – 22 Hz
Rear hop	10 – 15 Hz
Front hop	10 – 15 Hz
Rear tire longitudinal	7 Hz
Rider-shake	5 Hz
Wobble	7 Hz
Bounce	2 Hz
Weave	0 – 3.5 Hz
Capsize	Non – vibratory mode

Table 1 – Table of Motorcycle Vibration Modes

3.5 Quarter Car 2 DOF

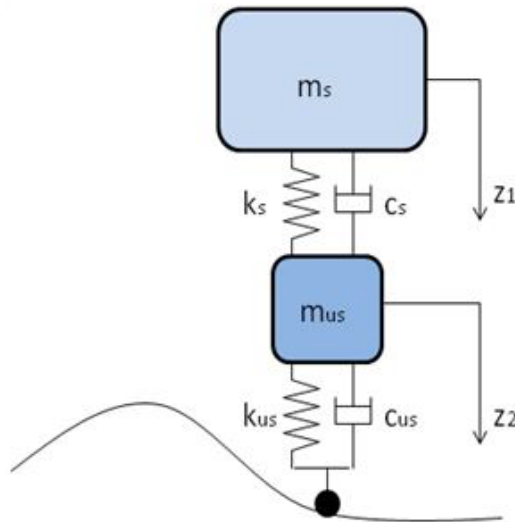


Figure 4- Quarter Car Model

More recent models have tended to increase the number of bodies included to find further oscillatory modes. However, as Limebeer and Sharp, (2006) suggest with the increased number of bodies comes increased complexity. Research conducted by Rajput and Bayliss, (2007) found that although their model was a simplified model compared to other more advanced multibody models, their results for the main modes of oscillation agreed with the more complex models. The complexity of the models and the oscillatory modes found were reviewed by Lake, Thomas and Williams, (2012) with their findings seen below.

When considering a multi degree of freedom (DOF) mass spring damper system, the simplest model used in automotive research is the Quarter-Car model. When discussing four-wheel vehicles, the quarter car represents one wheel and a fourth of the body mass. For a motorcycle, the “sprung” mass represents half of the motorcycle mass, minus unsprung masses such as wheels, tyres, brake calipers, brake discs etc. (Bhanu, 2009) explains that ‘this allows quick and simple analysis of the mathematical relationships.’

Although analysis can be conducted easily using the quarter car model, some important aspects are missed. As shown in section 3.3, a single DOF model can be used to model pitch. This occurs due to the difference in net force generated by the front and rear shock absorber systems. This includes the tyres and their influences on the pitch moment.

Due to its simplicity and ease of conceptualisation, the quarter car model is used as a benchmark to observe the difference in increasing model complexity on acceleration, velocity and displacement values. Increased complexity models include the 3 DOF half car model (excluding the effects of pitch) and the 4 DOF model which accounts for pitch. Separating out the pitch

between the two models allows a comparison of its effects on the system.

As well as studying the difference in the effects of pitch through using two models, it is useful to compare the differences in outcome with a more complex model. This is because with added complexity comes larger computation times with larger room for error.

3.6 Half Car 3 DOF

The half car model is an extension of the quarter car model which provides more detailed information regarding motorcycle dynamics. This is because it encompasses the entire suspension system, including all critical masses such as the unsprung masses, as well as the influences from the tyres.

In the models being developed, tyre damping is accounted for to give an insight into the effect this has on the results. In contrast, tyres are usually modelled as a simple spring as shown in figure 6.

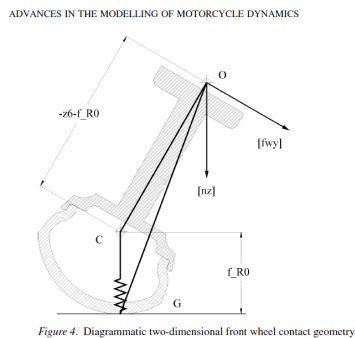


Figure 6- Diagram of Tyre Modelled as Simple Spring

The half car model, shown in figure 5, separates out the front and rear unsprung masses, allowing their influence on the sprung mass to be accounted for. In comparison to the 1 DOF model shown in figure 2, the 3 DOF model allows the motion of the sprung mass to be modelled. This is important as on a race motorcycle, the sprung mass is up to 10 times heavier than one of the unsprung masses. This

is because the sprung mass has a large influence on road holding, due to the weight transfer when including pitch.

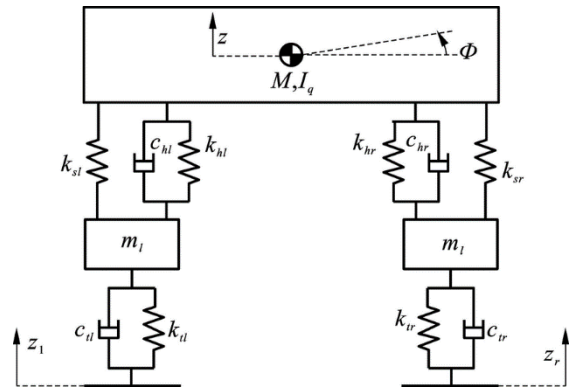


Figure 5 - Diagram of Half Car Model

Although the 3 DOF half car model does not give as accurate a representation of a motorcycle's true behaviour when excluding pitch. It is investigated in this paper with the use of basic calculations to a medium degree of accuracy. This is to determine if the 3 DOF model can provide results with less complexity of construction and faster run times.

3.7 Half Car with Pitch 4 DOF

To model the motorcycle as accurately as possible, the pitch of the motorcycle must be included, as it is an important aspect of the motorcycle's behaviour. However, adding the complexity of the pitch could slow down calculation times and possibly requires more time to build the model.

Above are the differential equations for modelling the motorcycle as a half car system. As can be seen, a fourth equation is now presented describing the torque that is created by the pitch moment. Pitch is heavily influenced by the spring and damping constants of the sprung shock absorbers.

Therefore, changing the balance of these constants will change the relative pitch experienced by the sprung mass. Conversely, if the constants K and C are modified proportionately, the pitch moment will be equal to the previous setup. However, changing the pitch characteristics is both influential and influenced by other factors.

Changing the sprung shock absorbers characteristics, even in proportion, will in reality result in different pitch characteristics that are not accounted for in the model. This is due to the influence that the changes will have on the geometry of the motorcycle. For example, a change in pitch will impact the dynamic rake angle of the motorcycle, which will in turn, affect the resistive force to the pitch moment. This is an important aspect of the modelling process and that could be part of further research that is conducted to improve the accuracy and applicability of the models shown here.

4 Methodology

4.1. Experimental setup

After the modifications to the experimental design and methods were completed the final setup was devised. This consisted of multiple accelerometers, potentiometers, hall effect sensors, switches, memory cards, power supplies, data transmission cables and wiring, as well as the final code used to control all the hardware. In this section, the placement, use and reason for inclusion of the various components will be explained, both diagrammatically and analytically.

Due to time constraints, real damping data from the motorcycles suspension was unable to be obtained. As a result, from this, a similar motorcycle in terms of mass and power was used. This was the same situation for the tyres, however the spring rate of the tyres

was measured directly. The damping will be verified by performing analysis on the data collected after verifying the various models against each other.

4.1.1 Hardware

As shown in section 3.5, a motorcycle can be modelled as a series of masses, connected with springs and dampers, and influenced by an input such as the shape of the road. Figure 4 shows the Quarter Car Model which can be used to explain the front or rear suspension independently on a motorcycle. Originally designed for cars, it uses one fourth of the cars sprung mass, and a half of a motorcycles. The main downfall of the quarter car model when analysing a full motorcycle suspension system is that the front and rear suspension must be analysed separately. This can be considered beneficial when keeping simplicity in mind, but it does not always attain the accuracy required.

To analyse the motorcycle using both the quarter car and half car methods, accelerometers were placed on the front and rear sprung masses. This allowed the flexibility required in testing the different models using the same experimental design.

Figure 5 shows the Half Car Model which is useful to model the entire motorcycles dynamic suspension response. This model has been developed to increase the accuracy over the quarter car model so is worthy of being tested. This can use the same accelerometers as those fitted for testing the quarter car model.

Originally it was planned to fit five accelerometers to the various masses on the motorcycle. This is explained in table 2

Accelerometer Number	Physical Placement
1	Front Unsprung Mass (Bottom Stanchion)
2	Front Sprung Mass (Top of Triple Clamp)
3	Centre of Gravity (COG)
4	Rear Sprung Mass (Under Tail Unit in Line with Rear Spindle)
5	Rear Unsprung Mass (Rear Swingarm Above Wheel Spindle)

Table 2– Placement of Accelerometers

These placements that are noted in the above table can be seen in figure 7 where it shows how these correlate to the positions on the Half Car Model.

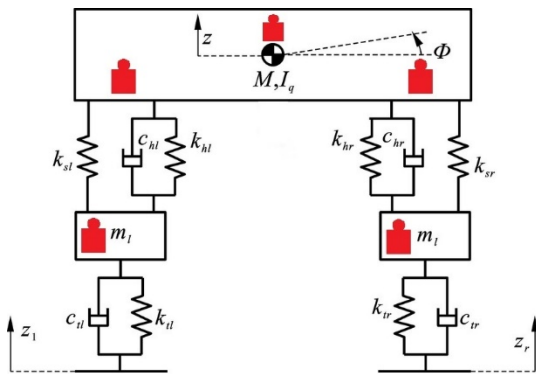


Figure 7- Placement on Masses of Half Car Model

After trial testing, it was found that with five accelerometers the sample rate was too low for accurate results. The accelerometer at the COG was removed to give a higher sample rate, and therefore more accurate results. With the compromise that the COG position and hence the sprung mass and pitch motion would then have to be calculated from the remaining four accelerometers.

As a verification tool, a front potentiometer was fitted to compare the outputs of both the front accelerometers and the potentiometer. This can be achieved as the first integral of acceleration is velocity, and the second integral of the sprung acceleration, minus the unsprung acceleration should give the fork travel. It was originally planned to fit a

potentiometer to the rear shock, but this proved more difficult than first expected, and therefore, was omitted to remove the risk of damage to the sensors. This leaves only the front potentiometer to give a confirmation of the accelerometers output.

To get a measure of the forward velocity of the motorcycle, and to gain another verification of the output of the accelerometers, a hall effect sensor was fitted to the front wheel. This was to measure the number of rotations and hence, determine the forward velocity.

4.1.2 Hardware Control

To control all the hardware, receive the data and convert it into a useful output, a control unit of some description is required. Out of the two options of a Datron 2D datalogger and an Arduino custom made datalogger, it was decided to use an Arduino due to the benefits it offered.

The Arduino in question was an Arduino DUE, which features an 84MHz microcontroller, 12 Analog inputs and an on board analog to digital convertor (ADC). To record the data, a memory card was fitted through a 'shield' which was installed onboard the Arduino and can be seen in Figure 8. The Arduino requires the controlling code to be uploaded with every change via a USB cable. Once uploaded, it can be disconnected from the computer if there is an external power source fitted, for the experiments two lithium ion batteries were used. A switch was also fitted to use to signal to the Arduino that the test was finished and to save the data to the memory card.



Figure 8 -- Arduino Uno Shield

The setup proposed above was fitted to the motorcycle via physical wiring, routed throughout the motorcycle to the required sensors.

4.2 Road Input

As a suspension system input, a specially designed step was constructed to give a known input to the unsprung masses for every millimetre of forward motion. The step specifications are below in table 3

Component	Dimension
Max Height (Amplitude)	50 mm (0.05 m)
Length	260 mm (0.26 m)
Shape (trigonometric function)	$1 - (\cos(x))^2$
Period of Function	$0:\pi$
Rate of Change of Height	$\frac{dy}{dx} = \sin 2x$

Table 3- Dimensions and Data for Step

Below is a graph (Figure 9) produced in MATLAB of the shape of the curve that the step is based on (not to scale) and a graph (Figure 10) of the rate of change of the height of the step plotted against linear distance.

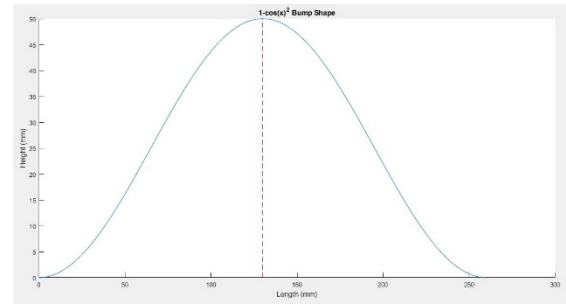


Figure 9 - $1 - (\cos(x))^2$ Step Displacement

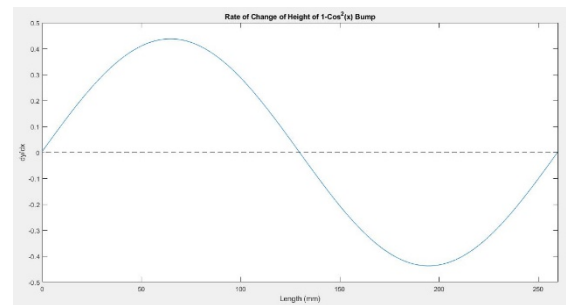


Figure 10- $1 - (\cos(x))^2$ Step Velocity

4.3 Sensor Calibration

From the accelerometer datasheet, it is possible to calibrate the accelerometers using the specifications provided. However, this is average test data collected from all accelerometers produced, and most importantly, measured across all axes of the accelerometer. Using different methods calibration can be achieved to different degrees of accuracy. The first and simplest method is to use a known acceleration (usually gravity) to compare the output of the actual accelerometer being used to the theoretical value (9.81 m/s^2). By placing the accelerometer flat, a reading on the axis being measured can be achieved. The reading obtained can then be used as the accelerometers reference point for $1g$ of acceleration. The issue with this method is that due to manufacturing imperfections each axis will have different sensitivities.

Taking this into consideration, the following method is performed on each accelerometer.

Through placing the accelerometer so that the axis is parallel to the earth's gravitational acceleration, the data point is recorded then the accelerometer is rotated 180 degrees. This will read negative 1g acceleration, giving twice the spread of data to average across the 2g range. This was repeated for each of the three axes, recording the reading on the other two axes perpendicular to the gravitational field. Using linear interpolation, the zero g point can be found and is used to increase the accuracy of the calibration. Once all readings are obtained, calculations to find the exact value of the increase in data reading per 1g are used. Also calculated, are the -1g values and the scale of the acceleration change. This is performed using the following formulae (equation 3 to 6)

Equation 3- Equation to determine change per 1 g

$$1g = +ve\ 1g\ reading - 0g\ reading$$

Equation 4- Equation to determine change per -1 g

$$-1g = 0g\ reading - -ve\ 1g\ reading$$

Equation 5 - Determining Sensitivity/ Scale of Accelerometer

$$sensitivity = \frac{+ve\ 1g + -ve\ 1g}{2}$$

The value of the acceleration 'a' for the data value obtained 'x', can be performed in one calculation (equation 6) using the above data.

Equation 6 -Equation to Determine Acceleration from bit data

$$a\ (in\ g) = \frac{x - 0g\ reading}{scale}$$

To get the value in m/s², the 1g reading is multiplied by 9.81.

4.4 Testing Format

After trial tests to verify the experimental methods validity, the final set of tests were conducted. The tests were conducted with multiple "runs" in each test while changing a parameter each time. These were then repeated to reduce the chance of anomalies and outliers skewing the data, and so results could be averaged to improve credibility. Overall 8 tests were conducted, with 4 runs in each, to observe the reproducibility. Table 4 below shows what parameters changed in each test.

Test No.	Test Name	Observed Parameters
1	HighFast	High Damping, Fast Velocity
2	HighSlow	High Damping, Slow Velocity
3	MidFast	Mid Damping, Fast Velocity
4	MidSlow	Mid Damping, Slow Velocity
5	LowFast	Low Damping, Fast Velocity
6	LowSlow	Low Damping, Slow Velocity
7	LowPFast	Low Tyre Pressure, Fast Velocity
8	LowPSlow	Low Tyre Pressure, Slow Velocity

Table 4- Names of Experiments and Settings

The designations of "High", "Mid" and "Low" in terms of damping refer to the settings of the adjusters on the front suspension. "Fast" and "Slow" refer to the overall forward velocity of the entire motorcycle. Explanations of the designations are in Table 4 below;

High Damping	All adjusters turned to hardest setting (Fully Clockwise)
Mid Damping	All adjusters turned to halfway setting
Low Damping	All adjusters turned to the softest position (Fully Counter clockwise)
Fast Velocity	Bike is pushed at maximum attainable velocity
Slow Velocity	Bike is pushed at slow/ walking speed
Low Tyre Pressure	Tyre pressure dropped to 25 PSI (Standard 32 PSI)

Table 5 - Explanation of Settings Descriptions

In each of the tests, only one parameter was changed whilst the others were kept constant. However, an issue of attaining the same forward velocity every time emerged. This can be circumvented as the models that were used are dependent on forward velocity, and its effect on the data output can be observed. This means that changes in forward velocity are not critical if it is accurately measured, due to the model being able to adapt to different velocities allowing comparisons between datasets.

4.5 Data Acquisition Process

The step that was designed as a road input is placed a consistent 15m from the start position of the motorcycle and not moved throughout the experiments to ensure the angle of the step to the motorcycles velocity was constant. The data acquisition system was prepared following a check of the motorcycle, including motorcycle trim, tyre pressures, suspension and geometry settings. Data acquisition preparation required ensuring that power sources were at full capacity. A clean, formatted micro SD memory card was inserted into the SD port on the Arduino shield.

After physical system checks, the calibration code was uploaded to check all sensors were

working as required, zeroed, that they were flat and outputting values which were at an acceptable sample rate. The governing test code was then uploaded, a trial run conducted to check sensors are set up correctly and that the data is saved to the SD card.

After all preliminary checks, the rider took the required position. The calibration code was then run a second time to check that sensors were calibrated and positioned for the rider's weight, as when the weight changes the dynamic geometry of the motorcycle also changes. The accelerometers were then fixed in position and not moved. The slow speed camera was inspected and tested, after which the tests were then conducted, with four runs of each test.

The same rider was used for every test, and the accelerometers were checked for calibration errors for each test to ensure that no measurement drift occurred.

4.6 Microprocessor choice

When starting this project, the original plan was to use a Datron 2D system. However, due to the lack of availability of accelerometers, and the price of purchasing more, it was decided to use an Arduino system. The Arduino system has the ability of root level programming, which offers more customisation and flexibility compared to the 2D system. This requires the user to write the code to run the hardware and to perform calculations. Compared to the possible restrictions of the 2D system, as the software is already designed by the manufacturer, the Arduino is a superior solution.

The original Arduino that was chosen was the Arduino UNO. This was due to the compact size, but this was soon realised to be a shortcoming. The lack of analog inputs meant that the UNO could not be used, so an

Arduino MEGA (PICTURE) was chosen due to the larger number of inputs available.



Figure 11 - Arduino UNO

After connecting the hardware and running the complete system, it was found that the MEGA was not powerful enough with the 16MHz clock speed to reach the required sample rate. This meant that the maximum and minimum data points of the acceleration data were next to each other (when viewed as indexed points as in figure 12 at the sample rate of 86Hz). It was therefore decided to use a more powerful Arduino, being the Arduino DUE. This model of Arduino has the same physical format as the MEGA, but with an 84MHz clock speed and larger SRAM and flash memory, allowing faster computation. After minor modifications to the code, the test was repeated with the same hardware, with a sample rate of 187Hz reached, allowing more accurate data.

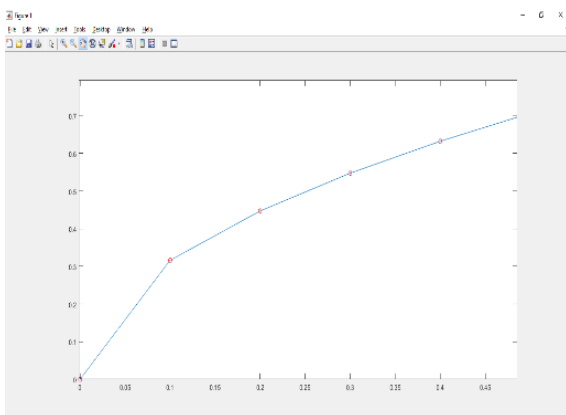


Figure 12 - Indexed Points

One of the drawbacks of the Arduino DUE is the lower number of analog inputs compared to the MEGA.

As can be seen the DUE has 12 analog inputs, compared to the 16 inputs of the MEGA. As each of the accelerometers has 3 outputs for each axis, only four accelerometers were able to be used. The COG accelerometer was removed, as although it would allow collection of useful data, it was not as important as the two sprung and unsprung accelerometers. Due to a front potentiometer being required, one of the accelerometers had to have an axis removed from the measurements. It was decided that this would be the "X" axis (left-right) of the rear unsprung accelerometer, as it has no use for the current measurements, which are all for "in-plane dynamics".

4.7 Accelerometer Choice

The original choice for an accelerometer was based on the availability, price and accuracy. For this reason, InvenSense MPU-6050 chip was selected. The MPU-6050 is a digital, 6 DOF, combined accelerometer and gyroscope.

The main feature that was first found to be desirable was that this chip has an onboard "Digital Motion Processor" (DMP) which conditions the output signal to remove noise. This allows it to give a more accurate result which does not need post-measurement processing and filtering. However, although this is a desirable feature, it is difficult to implement, with the length of the required code increasing from 95 lines for raw accelerometer output to 275 lines to use the DMP. This is only for a single accelerometer, and when trying to implement five, becomes quite complex and detracts from the original benefits. For this reason, after testing, it was decided to use a simpler system by using an analog accelerometer (Analog Devices ADXL335) to reduce the length and

complexity of the code required for the necessary data. For 5 accelerometers to be used, the length of the code, including calibration data (discussed below) is only 190 lines long. This simple piece of information gives an indication of some of the benefits of using analog accelerometers.

Although the MPU-6050 uses only one extra board connection, compared to the three of the ADXL335, the benefits of the simplicity of the code outweighs the lower number of board connections.

4.8 Mathematical Model Design

To test the difference between what can be measured on a physical, working motorcycle and what can be modelled, multiple models were devised and attempted.

4.9 Simulink Model

The first model was built in Simulink which uses the quarter car mass spring damper system as its basis for the calculations. Originally the model was built in MATLAB, using the same differential equations, but was found to be too time consuming to complete using this program. The issue arose when trying to input the “step” into the model. Using just the differential equations derived originally, the system of equations is for a non – forced oscillation, with the mass being given an initial condition, and left to freely oscillate. It was attempted to input the step into the model, but this proved more difficult than expected, so Simulink was used in its place due to the speed and simplicity Simulink offered.

5 Results

To include all the necessary parameters required for calculation, a specification/ data sheet was produced with all the necessary data to create the model. The model is based on the forward velocity of the motorcycle, dimensions of the step, various masses (sprung, unsprung and COG) of the motorcycle, spring rates of the front and rear suspension (obtained from measurement) and the damping data.

The models developed are all based on the differential equation derived from $F = ma$, $F = CV$ and $F = Kx$.

Equation 7- Characteristic Equation

$$m \cdot \frac{\partial^2 y}{\partial x^2} + C \cdot \frac{\partial y}{\partial x} + Kx = 0$$

The results from the experiments conducted was outputted in the form of a table of numbers in a .TXT file format. This was then inputted into Microsoft Excel where basic calculations and conversions were applied. This was done to be able to render the data into a useable format in terms of base units. The output of the Arduino is in the information type ‘bits’ due to the use of an analog to digital convertor (ADC).

The data from the sensors was imported into MATLAB where it could be manipulated into graphical form, which can be seen below. The acceleration data for the four locations can also be directly graphed. Whereas velocity and displacement data required the original data to be integrated. To get spring and damper force, the velocity and displacement data was multiplied by the corresponding constant.

Due to noise in the system, some of the data had to be filtered to allow operations to be applied. This was implemented using a Savitsky-Golay filter.

Below are the graphs for the experiments conducted under the conditions of mid-range damping, 32 psi front tyre pressure, 30 psi rear tyre pressure and slow forward velocity (equal to 2.89 m/s).

The graphs below include the accelerometer data for the front and rear sprung and unsprung masses, the hall sensor (used as a wheel speed sensor) and a front suspension potentiometer to verify the accelerometer output. The accelerometers all measure in the X, Y and Z directions, two of which are redundant for testing the current hypotheses, due to the tests being 'in-plane'. Table 6 shows which axes are used for measuring the vertical plane for each accelerometer, and which axis is displayed for each accelerometer below

<u>Accelerometer</u>	<u>Axis Analysed</u>
1 (Front Unsprung)	Y
2 (Front Sprung)	Z
4 (Rear Sprung)	Z
5 (Rear Unsprung)	Z

Table 6 - Axes Used For Measurements

The graphs below have been modified from the original measurements by removing the data from before the event, e.g. before contact with the step.

5.1 Simulink Model Data

Using Simulink multiple models were created and then analysed. These included the quarter car model, half car model, and half car model with pitch included 2,3 and 4 DOF respectively. Graphs of the sprung and unsprung masses have been created for all three models, including the pitch of the sprung mass in the third model.

Benefits of the half car model include the ability to model the effects of different spring rates and damping values, front to rear. This is especially important when including pitch as the different sprung spring rates and damping values can influence the pitch acceleration, velocity and angle.

To easily visualise the start, peak and end of the step for the front and rear, some of the graphs have dashed lines to signify these key points. The table below shows what the colours of the dashed lines indicate, as these have been added to show the critical points of the step.

<u>Colour</u>	<u>Meaning</u>
Magenta	Start of Step (Min Amplitude)
Red	Mid-Point of Step (Max Amplitude)
Green	End of Step (Min Amplitude)

Table 7 - Colour codes for Step Locations

Below with the graphs are also the differential equations that they have been created from.

5.2 Quarter Car Model Graphs

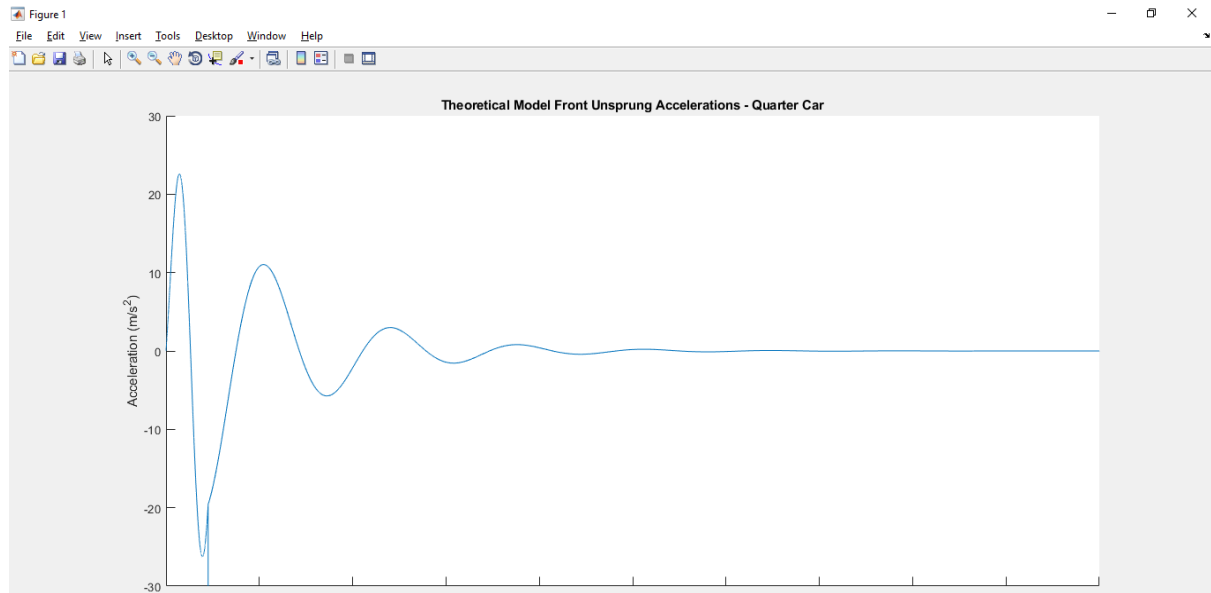


Figure 13

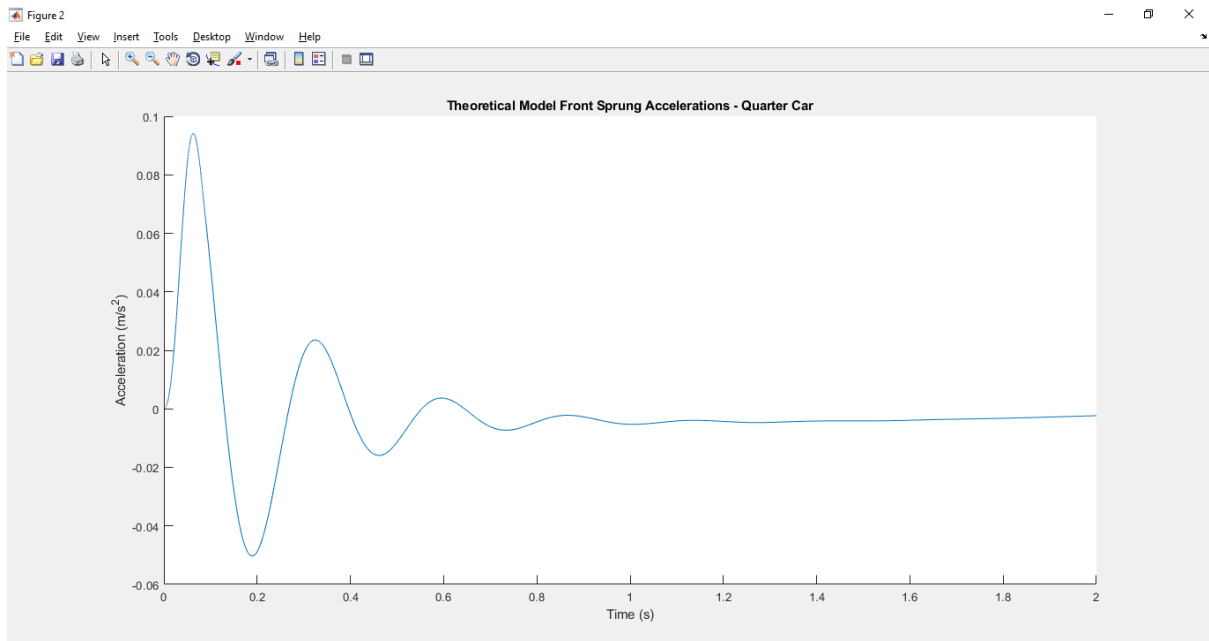


Figure 14

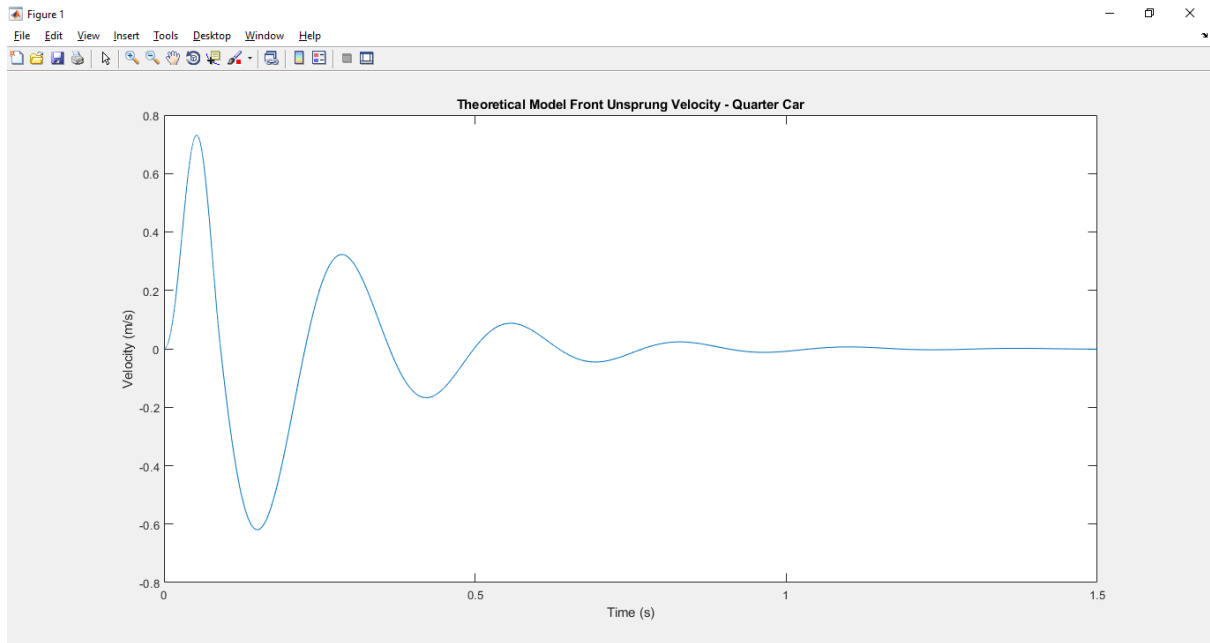


Figure 15

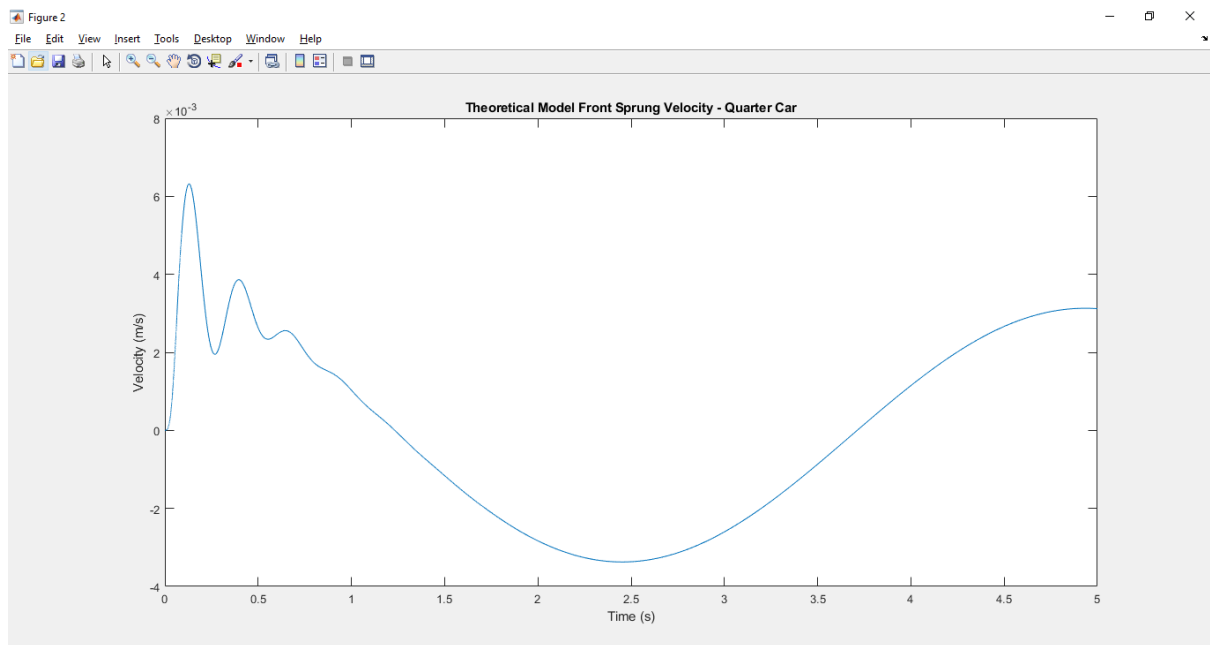


Figure 16

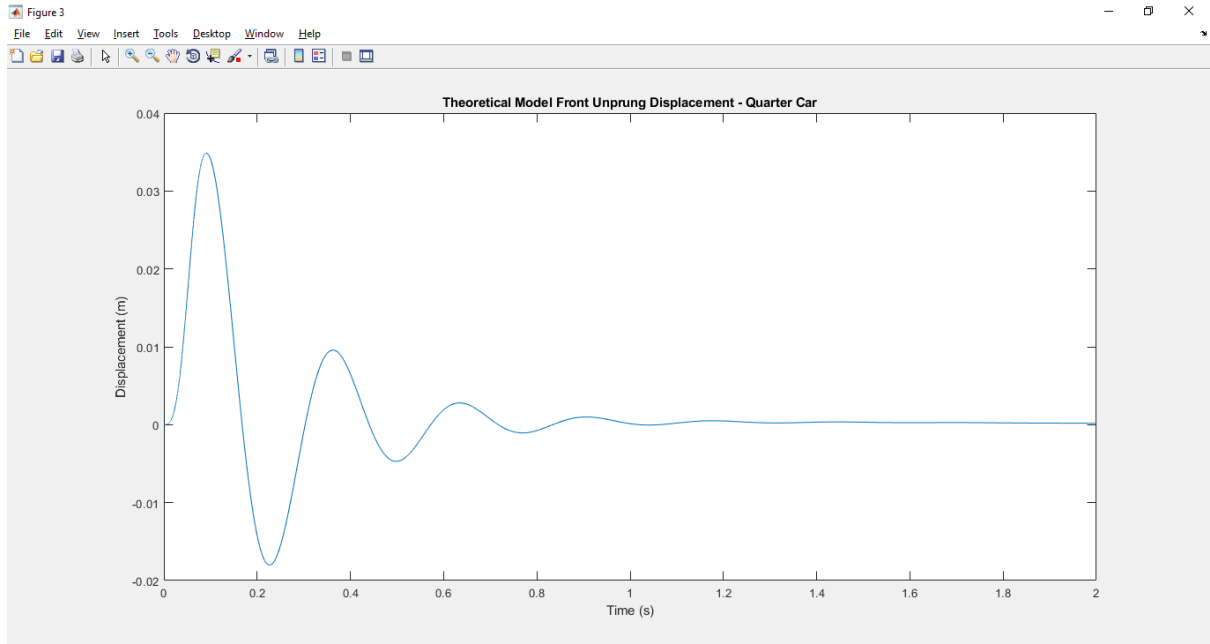


Figure 17

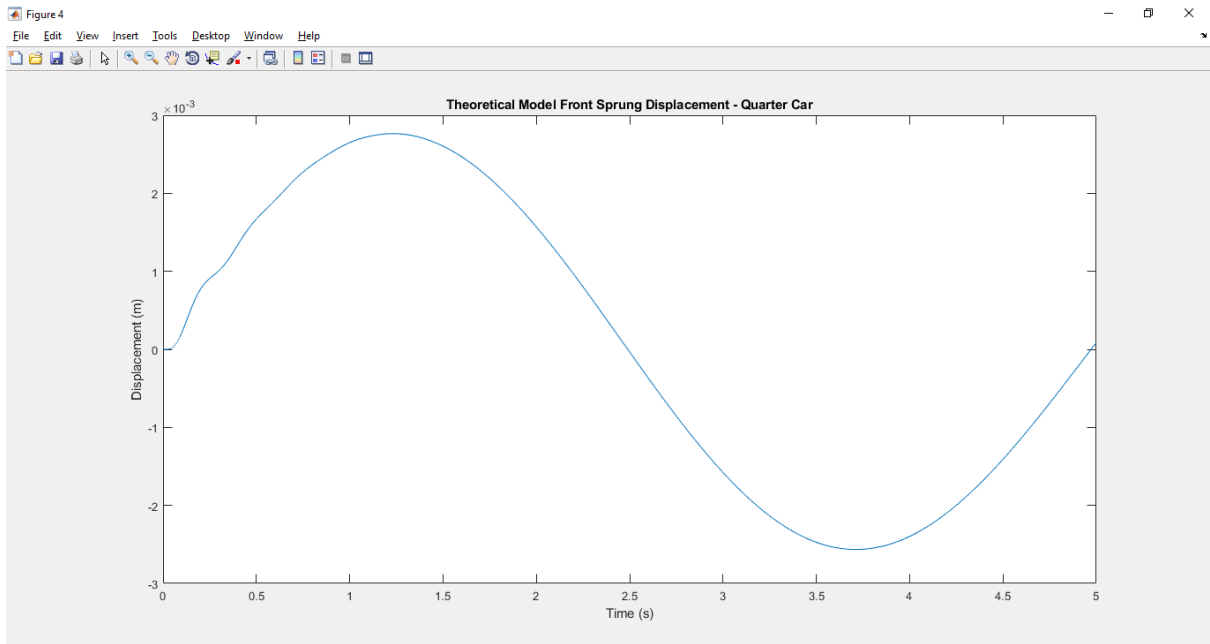


Figure 18

$$m_s \ddot{z} - C_s(\dot{z}_s - \dot{z}_u) - k_s(z_s - z_u) = 0$$

$$m_s \ddot{z} + C_s(\dot{z}_s - \dot{z}_u) + k_s(z_s - z_u) - C_u(\dot{z}_s - \dot{z}) - k_u(Z_u - Z) = 0$$

5.3 Half Car Model Without Pitch

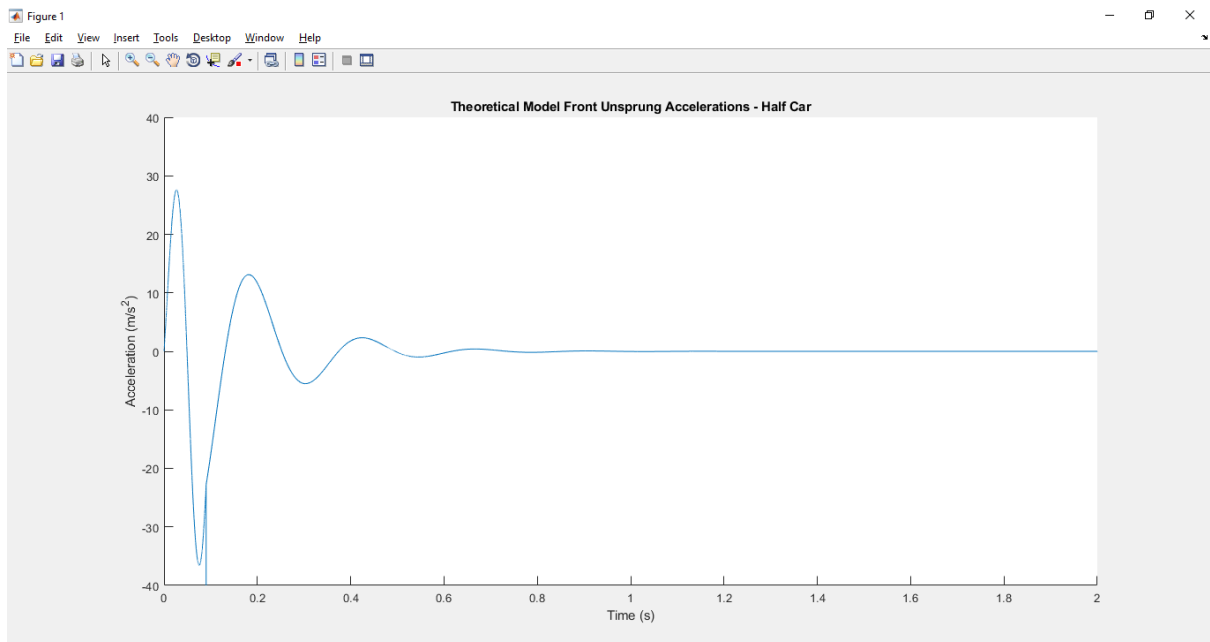


Figure 19

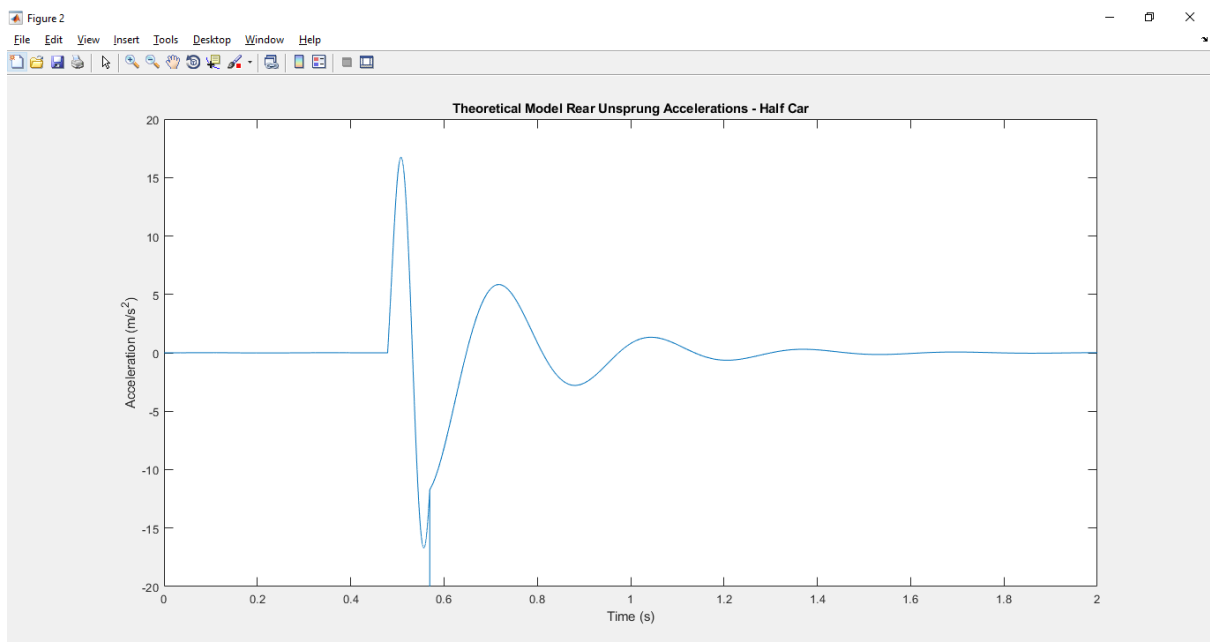


Figure 20

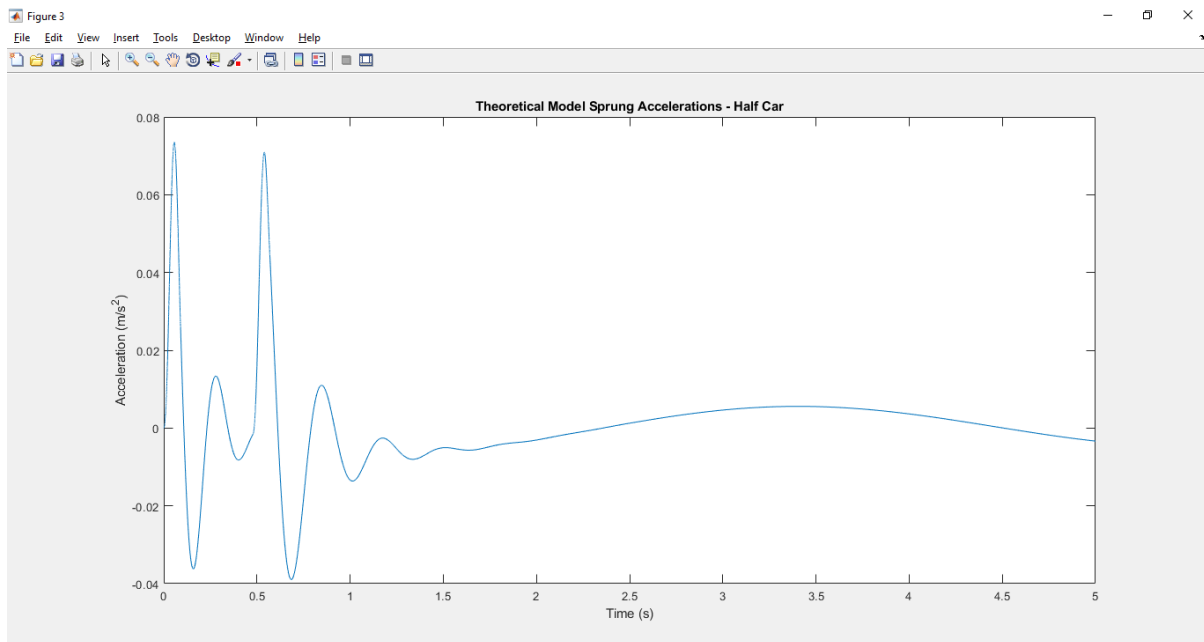


Figure 21

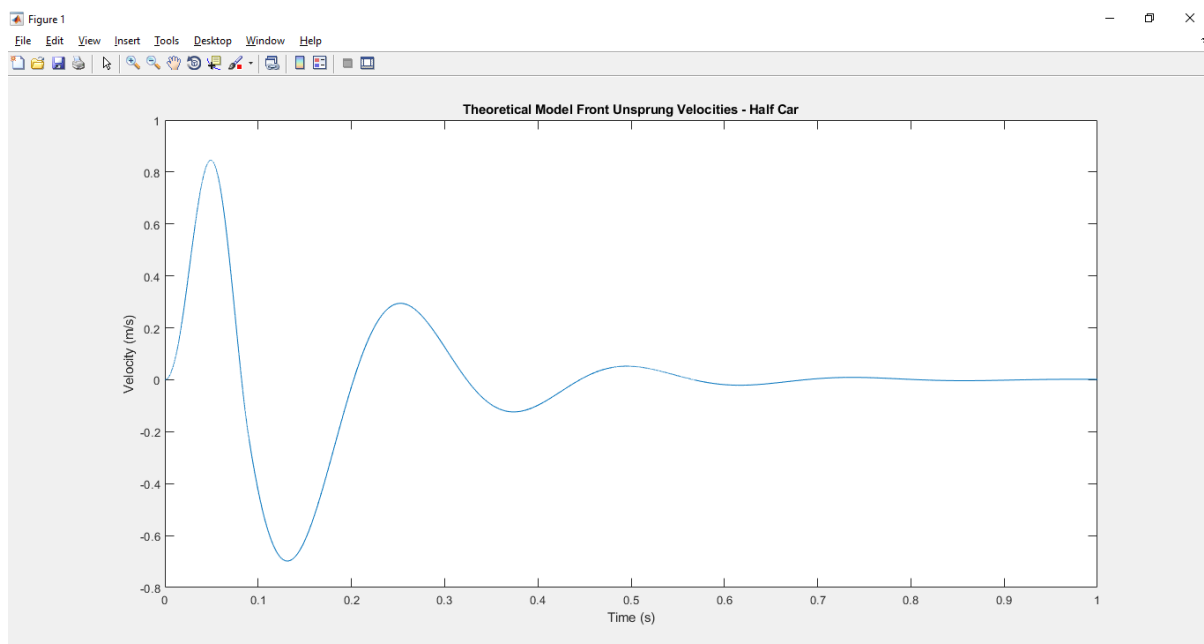


Figure 22

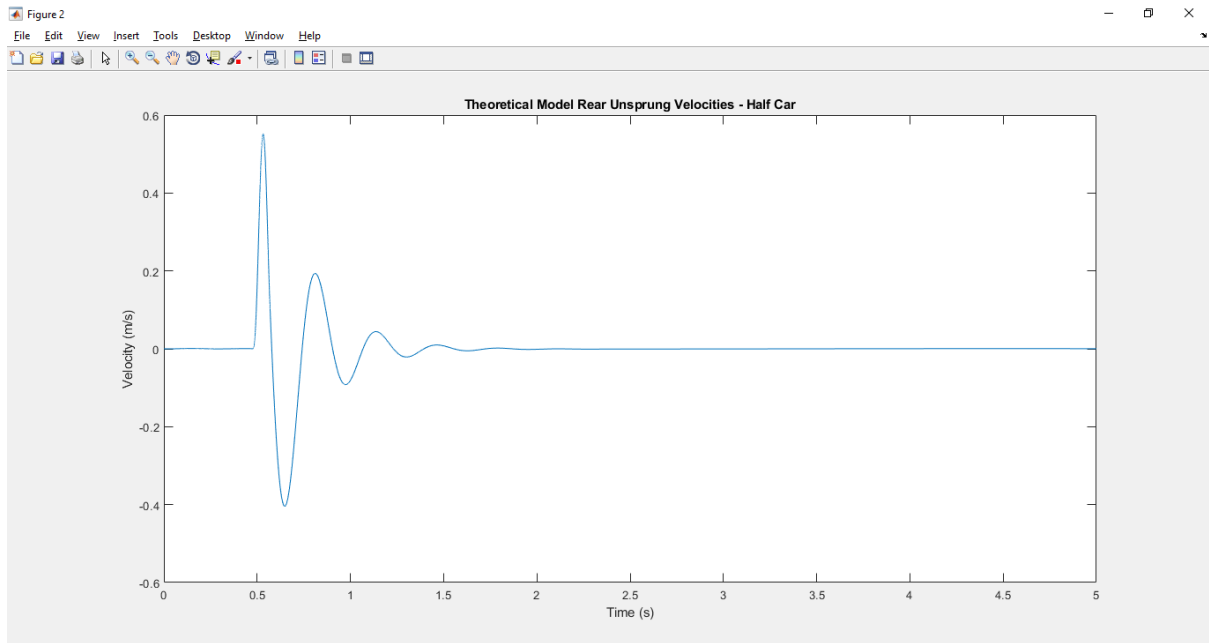


Figure 23

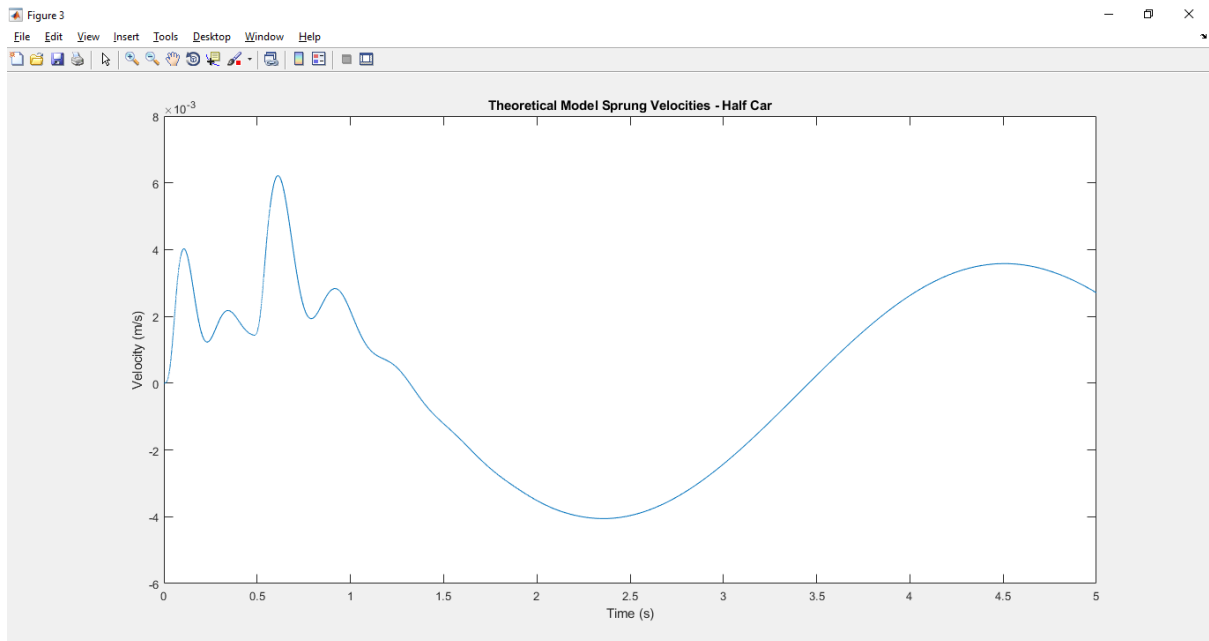


Figure 24

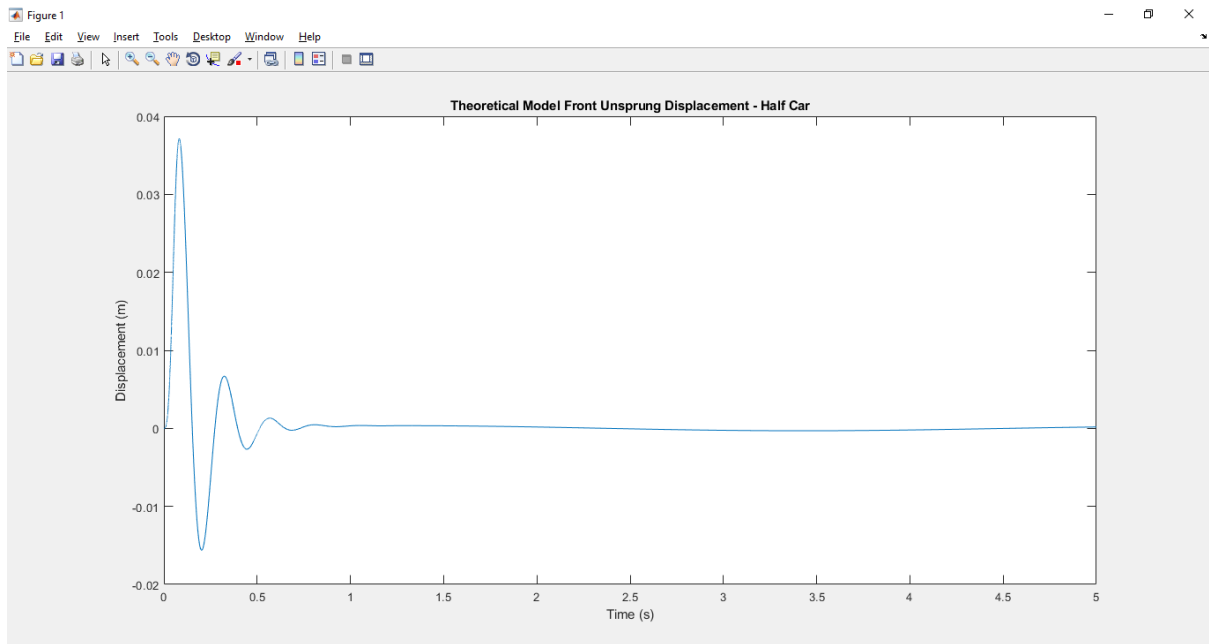


Figure 25

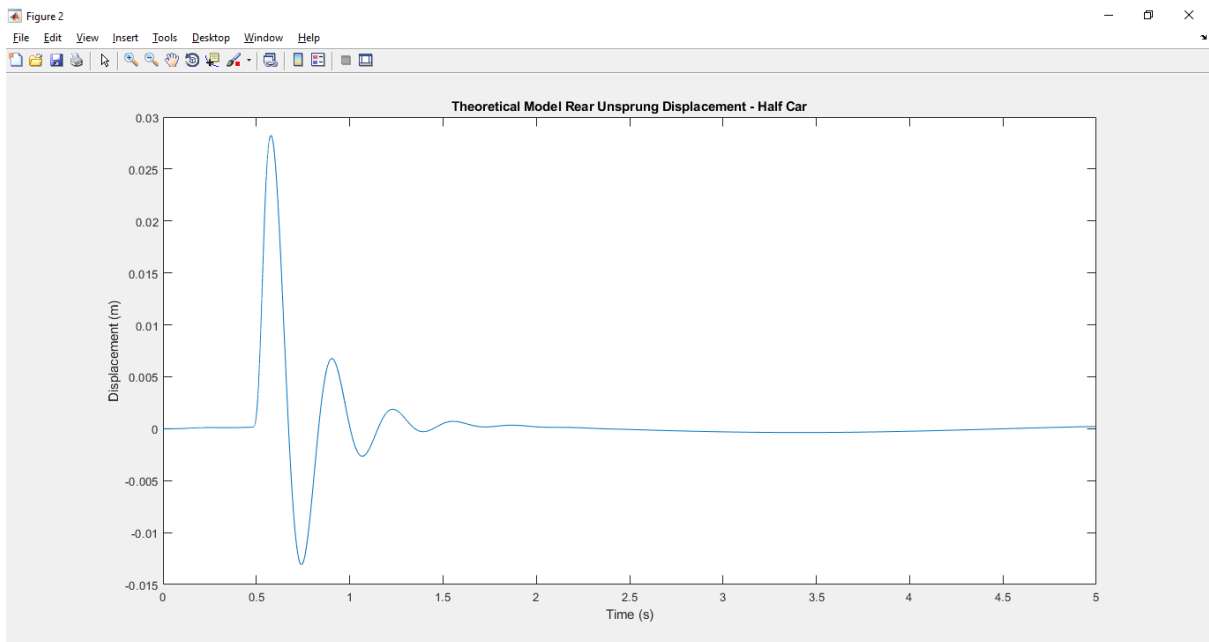


Figure 26

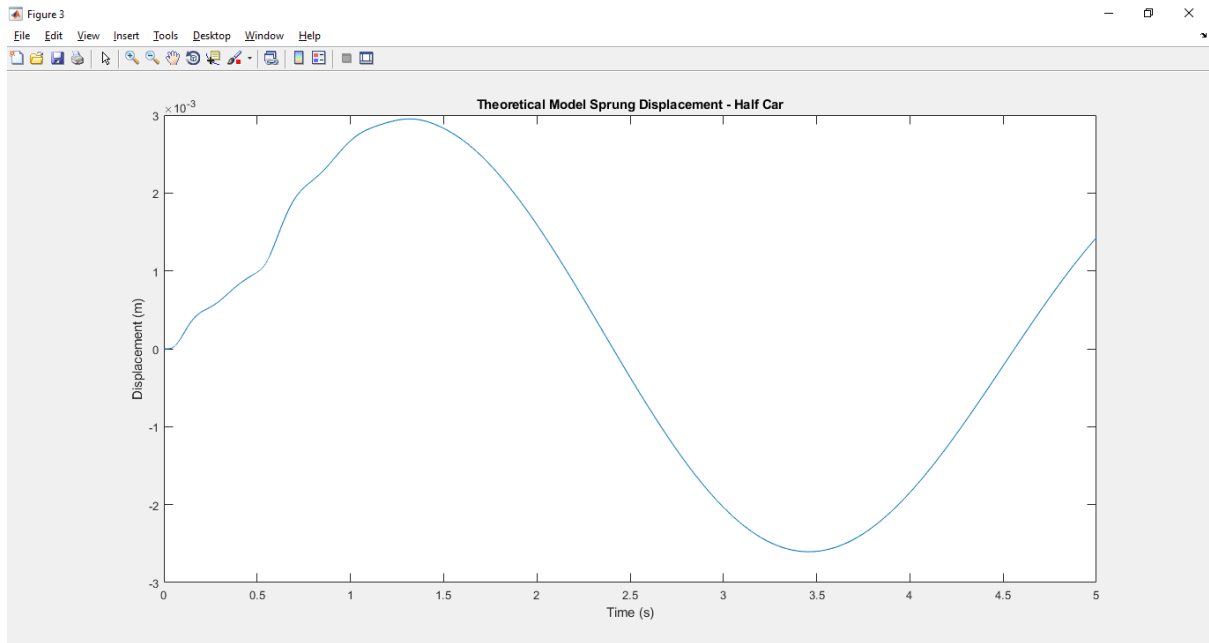


Figure 27

5.4 Half Car with Pitch

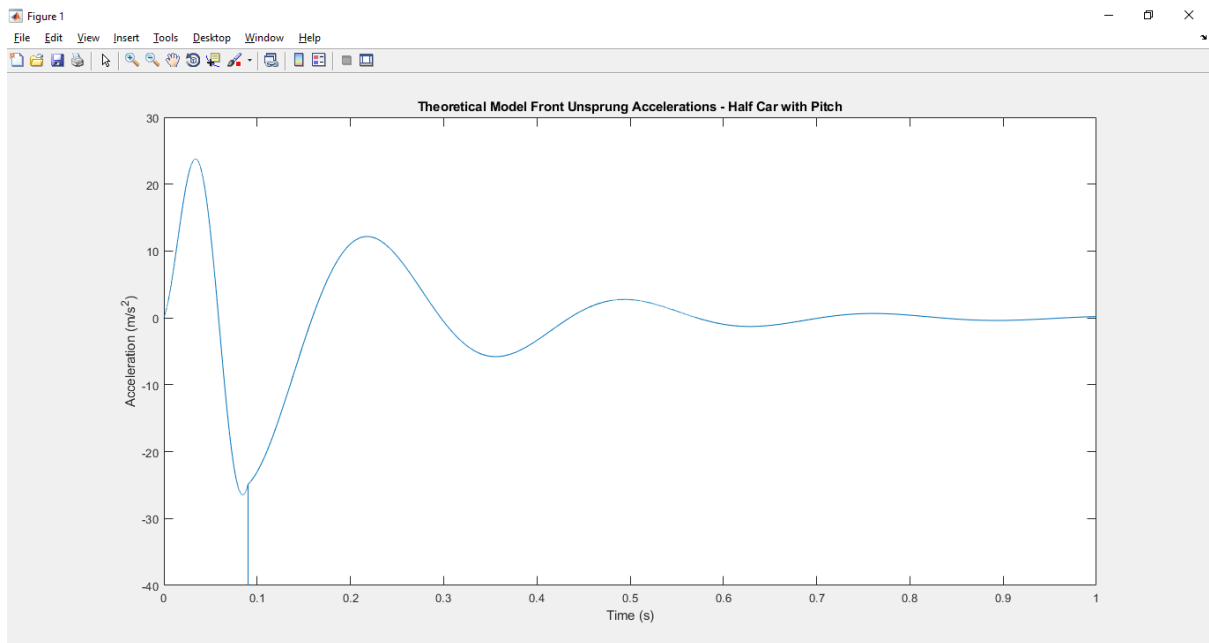


Figure 28

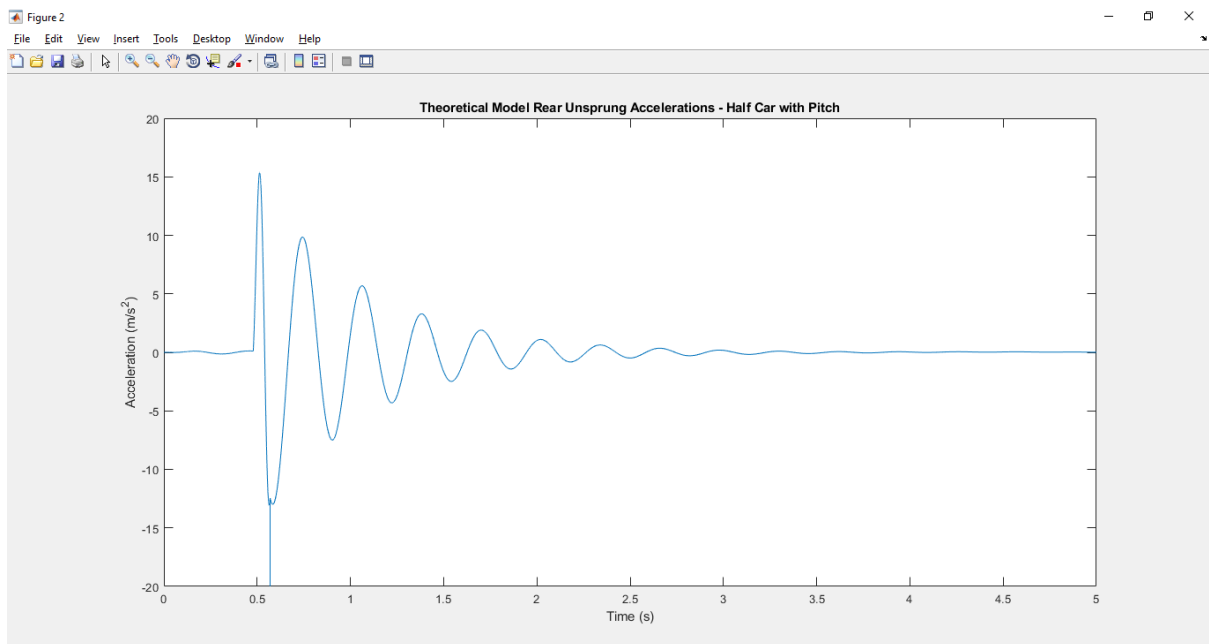


Figure 29

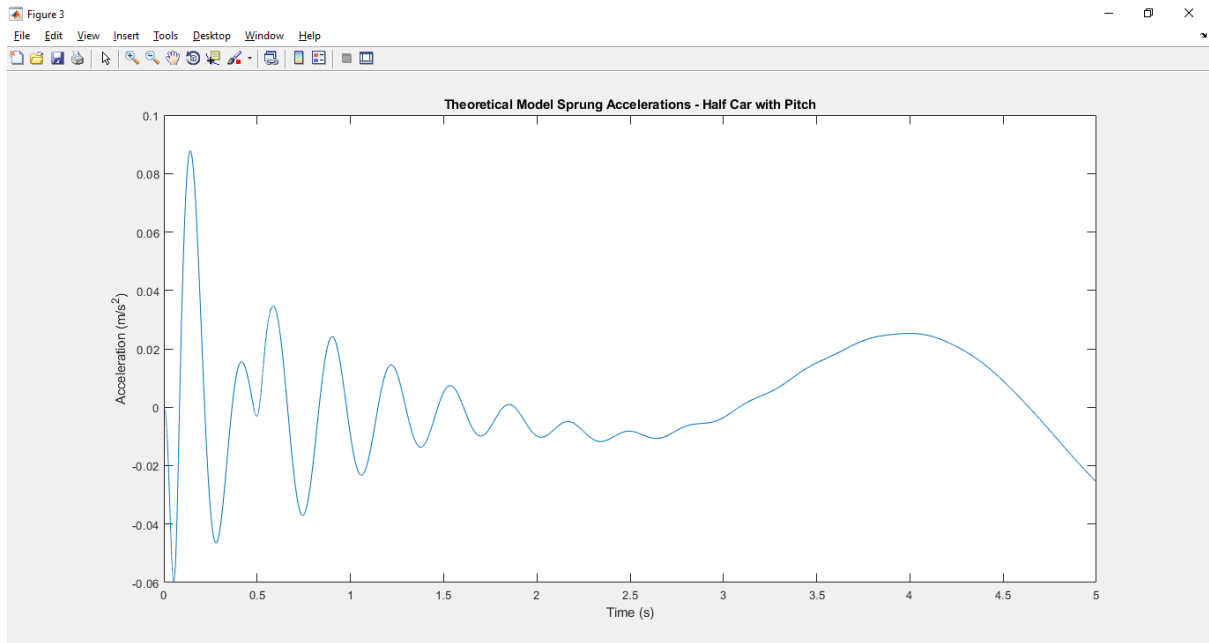


Figure 30

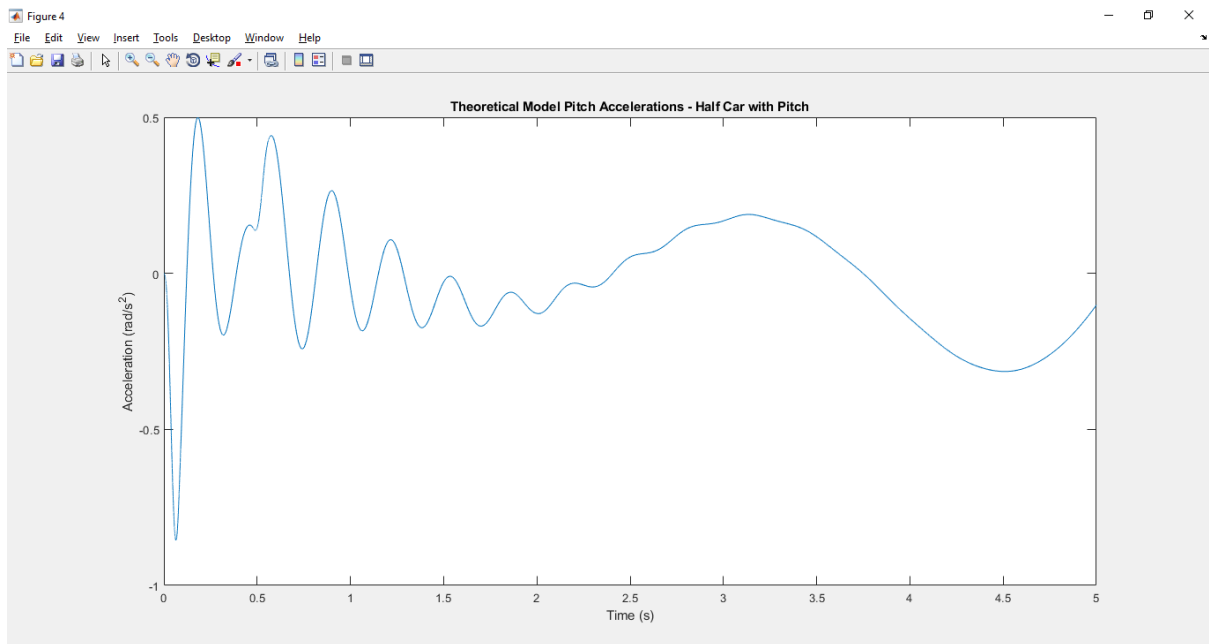


Figure 31

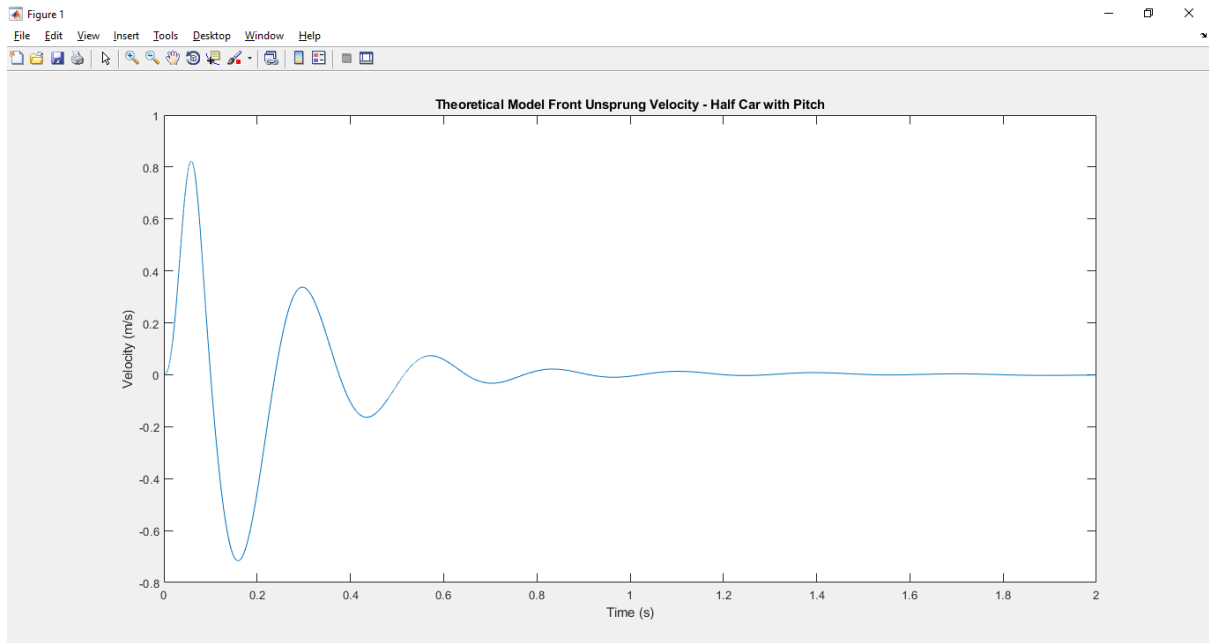


Figure 32

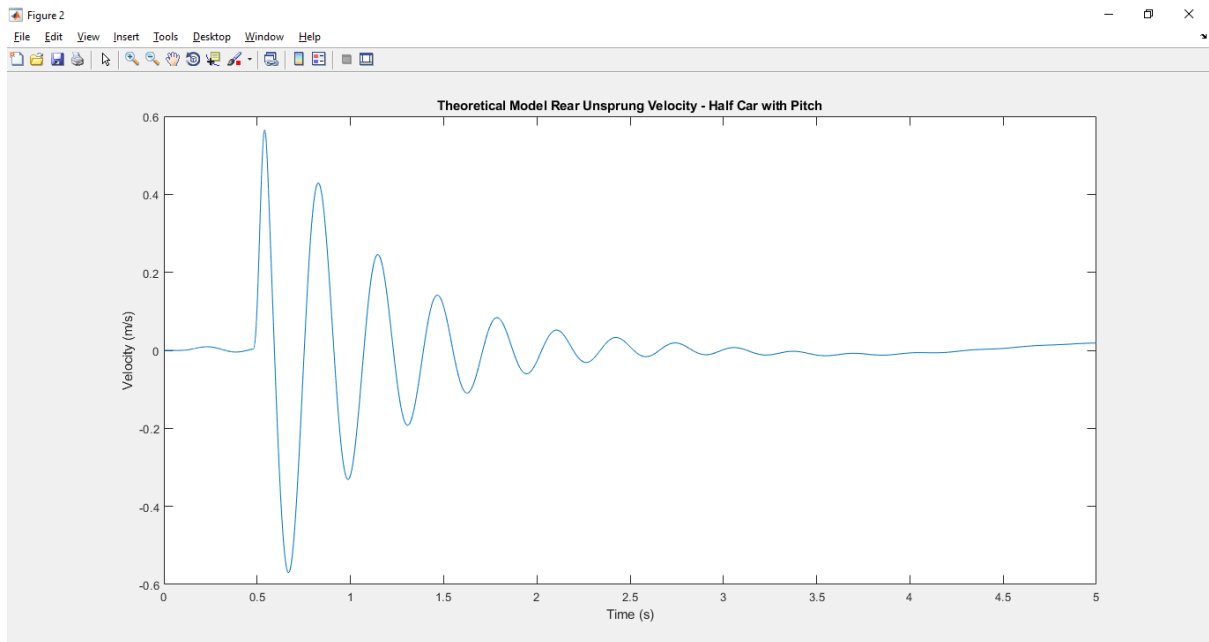


Figure 33

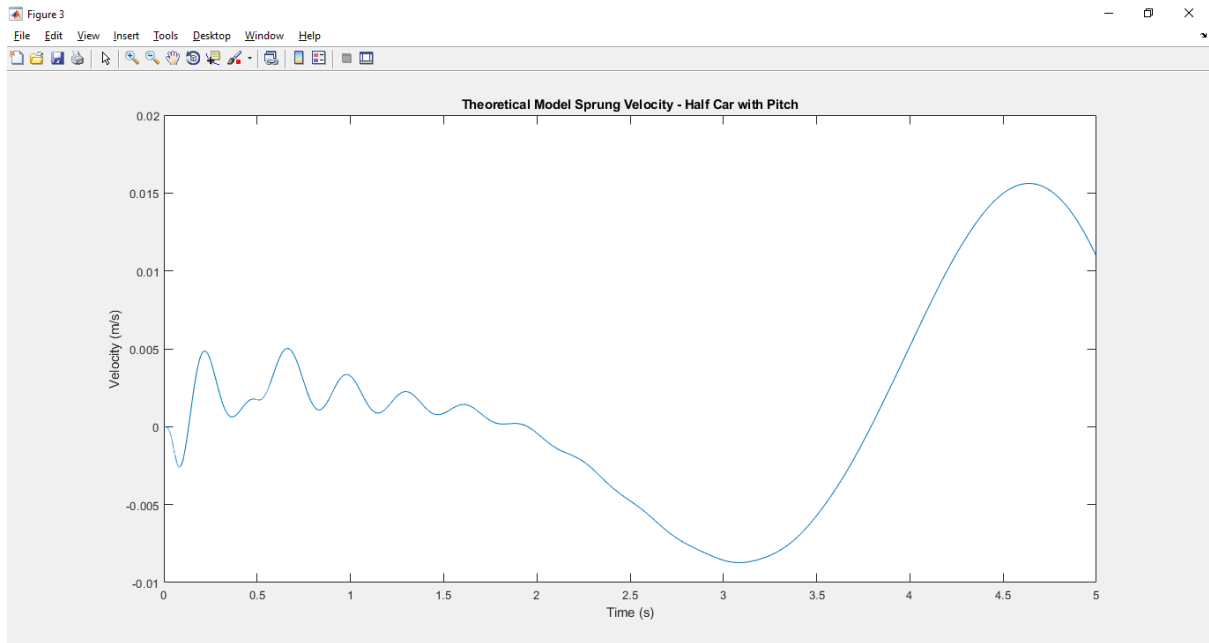


Figure 34

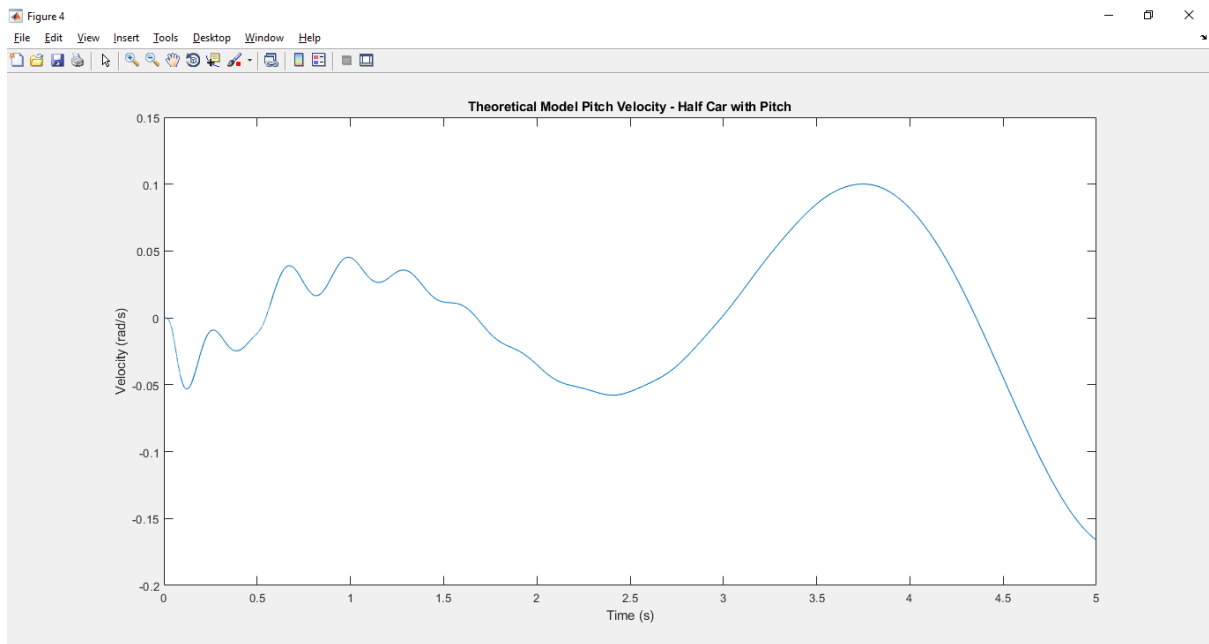


Figure 35

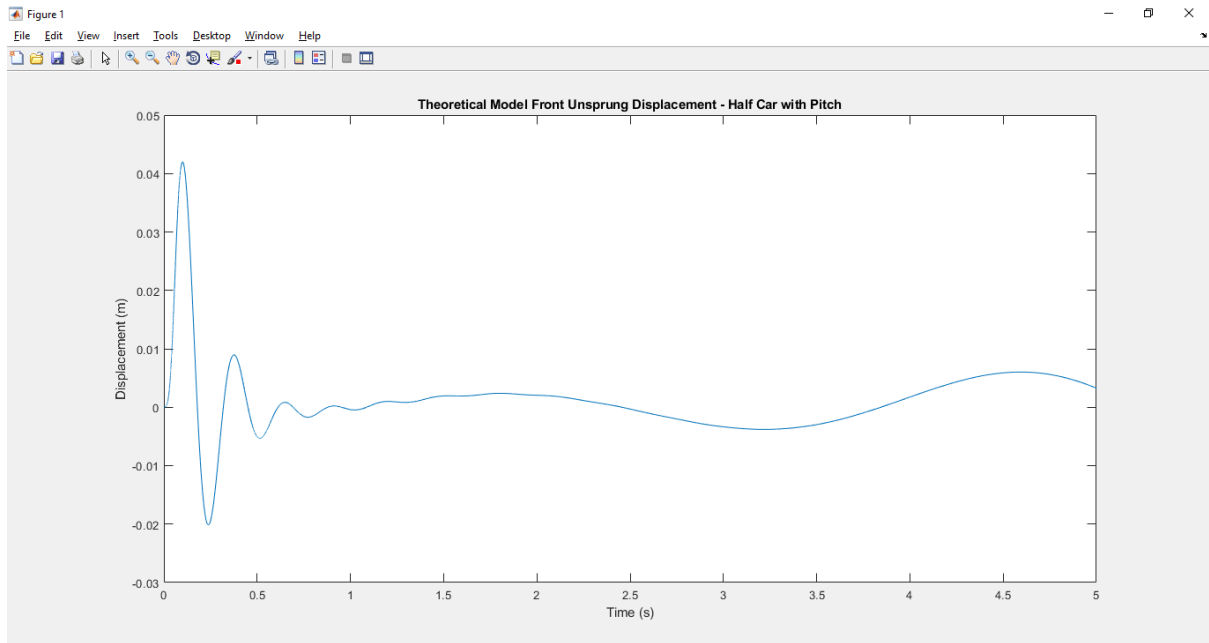


Figure 36

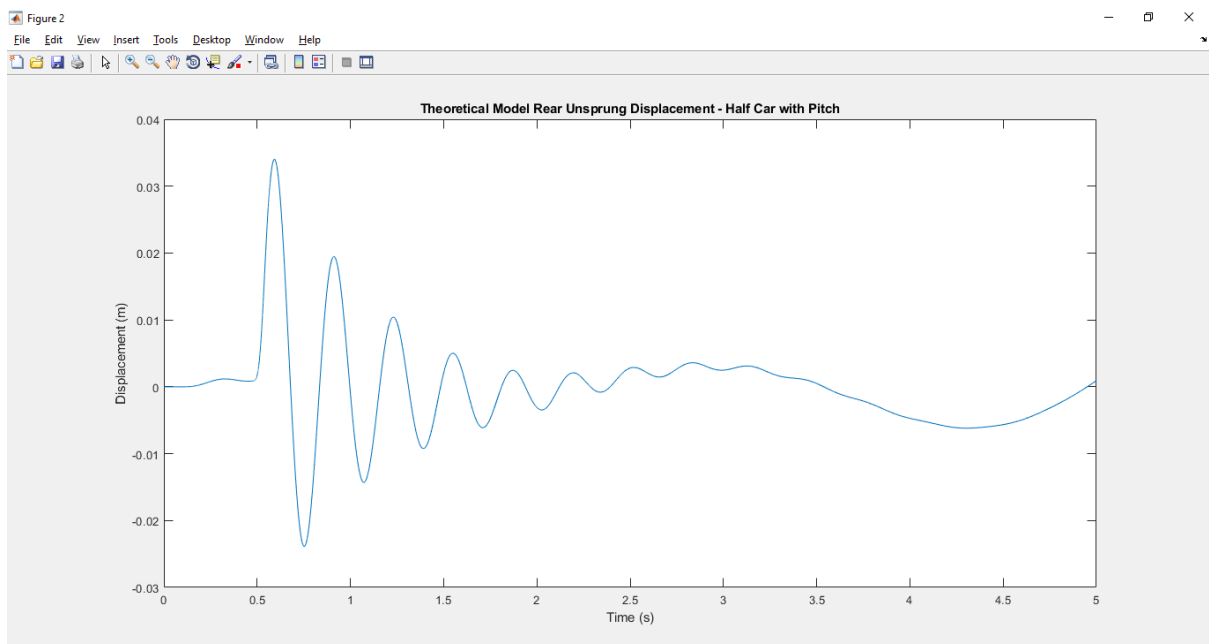


Figure 37

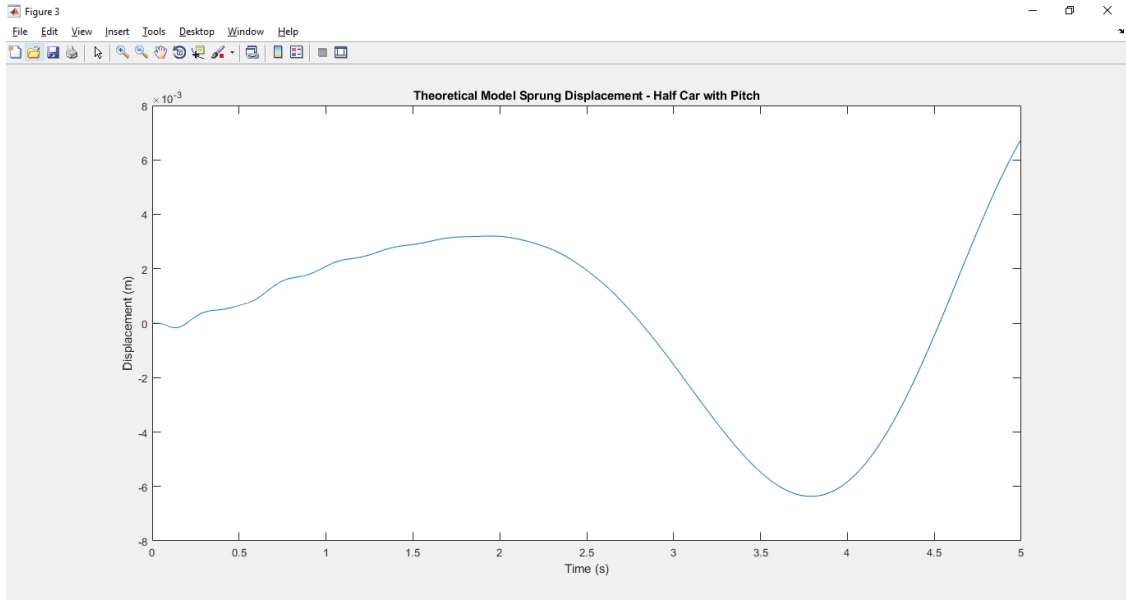


Figure 38

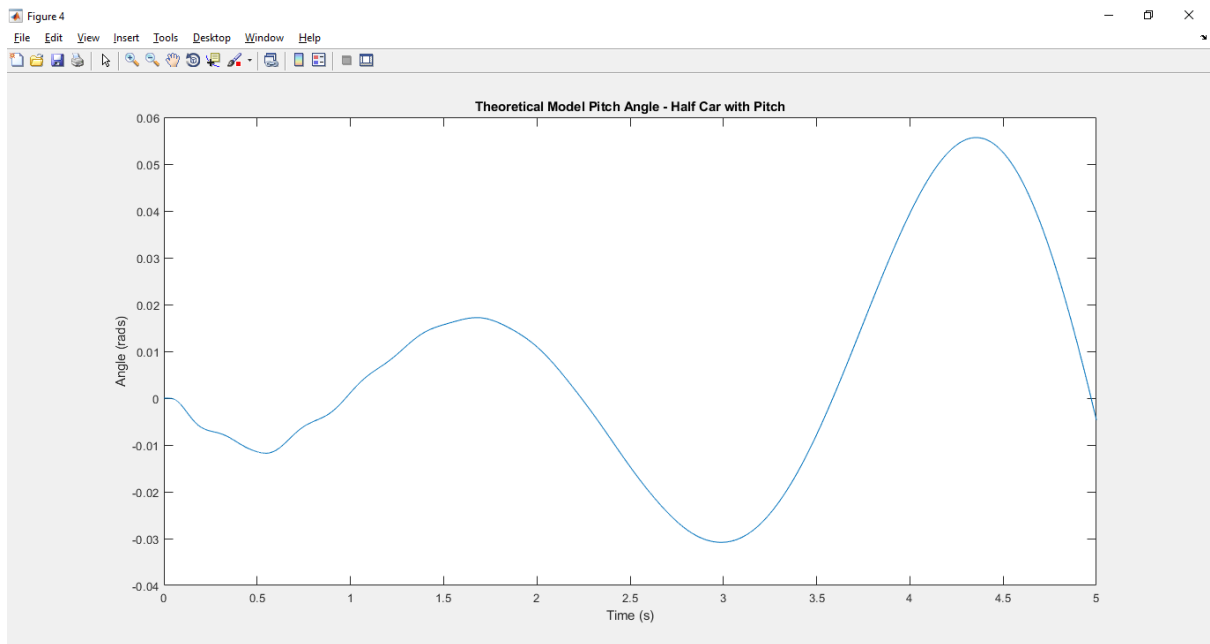


Figure 39

$$m_s \ddot{Z}_s + K_{fs}(Z_s - Z_f) + K_{rs}(Z_s - Z_r) + C_{fs}(\dot{Z}_s + \dot{Z}_f) + C_{rs}(\dot{Z}_s - \dot{Z}_r) + K_{fs}(p - b)^2 \theta - K_{rs}(b^2 \theta) + C_{fs}(p - b)\dot{\theta} - C_{rs}b\dot{\theta} = 0$$

$$I\ddot{\theta} + K_{fs}((Z_s - Z_f)(p - b)) - K_{rs}((Z_s - Z_r)b) + C_{fs}((\dot{Z}_s + \dot{Z}_f)(p - b)) + C_{rs}((\dot{Z}_s - \dot{Z}_r)b) + K_{fs}(p - b)^2 \theta + C_{fs}(p - b)\dot{\theta} - K_{rs}(b^2 \theta) + C_{fs}(p - b)\dot{\theta} - C_{rs}b\dot{\theta} = 0$$

$$m_f \ddot{Z}_f - K_{fs}(Z_s - Z_f) - C_{fs}(\dot{Z}_s + \dot{Z}_f) - K_{fs}(p - b)^2 \theta - C_{fs}\dot{\theta} + K_{fu}(Z_f - z) + C_{fu}(Z_f - \dot{z}) = 0$$

$$m_r \ddot{Z}_r - K_{rs}(Z_s - Z_r) - C_{rs}(\dot{Z}_s + \dot{Z}_r) - K_{rs}(b)^2 \theta - C_{rs}\dot{\theta} + K_{ru}(Z_r - z) + C_{ru}(Z_r - \dot{z}) = 0$$

6 Analysis

When analysing a complex system such as the 4 DOF half car model including pitch with a computer program such as MATLAB or Simulink there can be some downfalls. One of which is that it is easy to accept the results as true without thoroughly analysing the results and logically testing the output. To this end, the simplest model, The Simulink Quarter Car Model will be used as the backbone of the analysis as it is the easiest to conceptualise the output. The results will then be compared to the other models and any changes will be analysed, including comparing to the recorded experimental data. All the Simulink models had the same basic specifications to remove any differences caused by differences in this area. Table 8 summarises the specifications for each model and compares them to the other models.

Spec.	¼ Car	½ Car	½ Car w Pitch
Ms (kg)	108	216	216
Mfu (kg)	16	16	16
Mru (kg)	-	18	18
Kfs (N/m)	21568	21568	21568
Krs (N/m)	-	26500	26500
Kfu (N/m)	163000	163000	163000
Kru (N/m)	-	163000	163000
Cfs (Ns/m)	1108	1108	1108
Crs (Ns/m)	-	1757	1757
Cfu (Ns/m)	2296	2296	2296
Cru (Ns/m)	-	2621	2621
I (kg m ²)	-	-	40
B (m)	-	0.736	0.736
P (m)	-	1.38	1.38

Table 8 - Specification Data For Models

As can be seen, some of the rows in the quarter car model column are empty. This is due to the quarter car model being applied only to the front as a means of validation. It can however, also be used to model the rear. The above data was inputted into a MATLAB .MAT file and used as an input to the Simulink models.

6.1 Forward Velocity Implications

An issue that was encountered both in the theoretical modelling, and with the experimental data, was the combination of forward velocity and the step dimensions. During experimentation it was found that even at slow speed when at the peak of the step the front wheel would leave contact with the surface of the step. It is seen that at just before the peak of the step the wheel is still in contact (figure 40). However, a short period later when the centreline of the tyre is in line with the end of the step, the tyre is visibly airborne. The last snapshot in figure 42 is the tyre first contacting the ground again, a considerable distance from the end of the step.



Figure 40 - Tyre momentarily before peak



Figure 42 - Tyre after peak and airborne

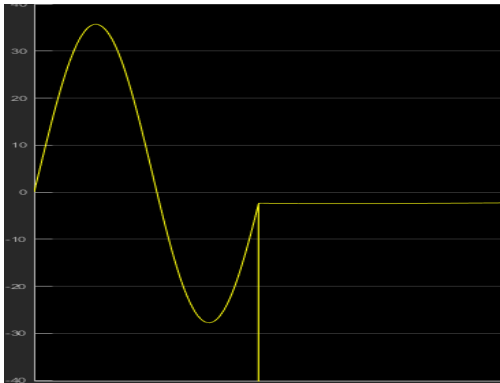


Figure 41 - Graph of negative spike in data

The above development also has an impact on the theoretical models. As low as 4 m/s it is noticeable that the expected second peak of the unsprung acceleration plot is not visible, compared to the 2 m/s model. When running the Simulink simulation, the runtime diagnostics produced multiple warning messages saying it was unable to reduce the minimum to step size more than 4.805 E -16

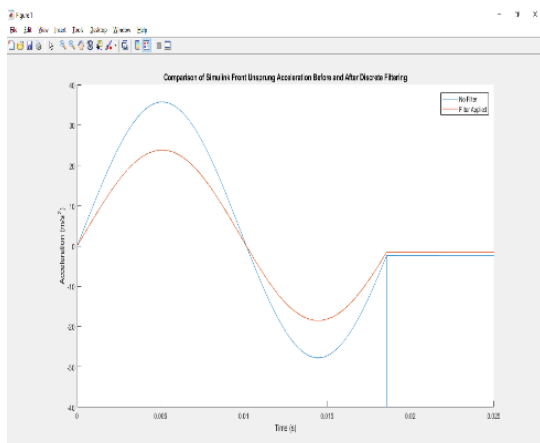


Figure 43- comparison of before and after filtering

seconds. This warning for the 12 m/s simulation, specified that the violation occurred at a time of 0.1352 seconds. From basic calculation, taking the wheelbase equal to 1.38m, the step to be symmetrical and 0.26m long, with the forward velocity of 12 m/s to be accurate, placing the rear tyre exactly at the peak of the step.

Equation 8 - Time Wheel Reaches Peak of Step

$$\begin{aligned} \text{Step Peak at } 12 \text{ ms}^{-1} &= \frac{p + (0.5 * \text{step } L)}{V} \\ &= \frac{1.64}{12} = 0.1366 \text{ s} \end{aligned}$$

To verify that the error occurs at the peak of the step, the simulation was run again at 14 m/s. This again resulted in an error, at a time of 0.1171 seconds. Repeating the calculation in equation 8 for step peak time, using 14 m/s results in a time that matches the error to 15 decimal places.

The graphical output of the unsprung accelerations after the simulation has been run at velocities matching above, shows an extremely large negative spike ($-7.121\text{E}+6 \text{ m/s}^2$) at the peak of the step, shown in figure 41.

It is possible to remove this spike in Simulink using a discrete filter before the graphical output. This does not change the simulation values, only the graph. Through applying the filter, the peak values of the graph changes, excluding the negative spike. This can be seen by figure 43, which shows a visible difference in the peaks graphically, with Table 9 showing the results numerically (minimum values excluding negative spike).

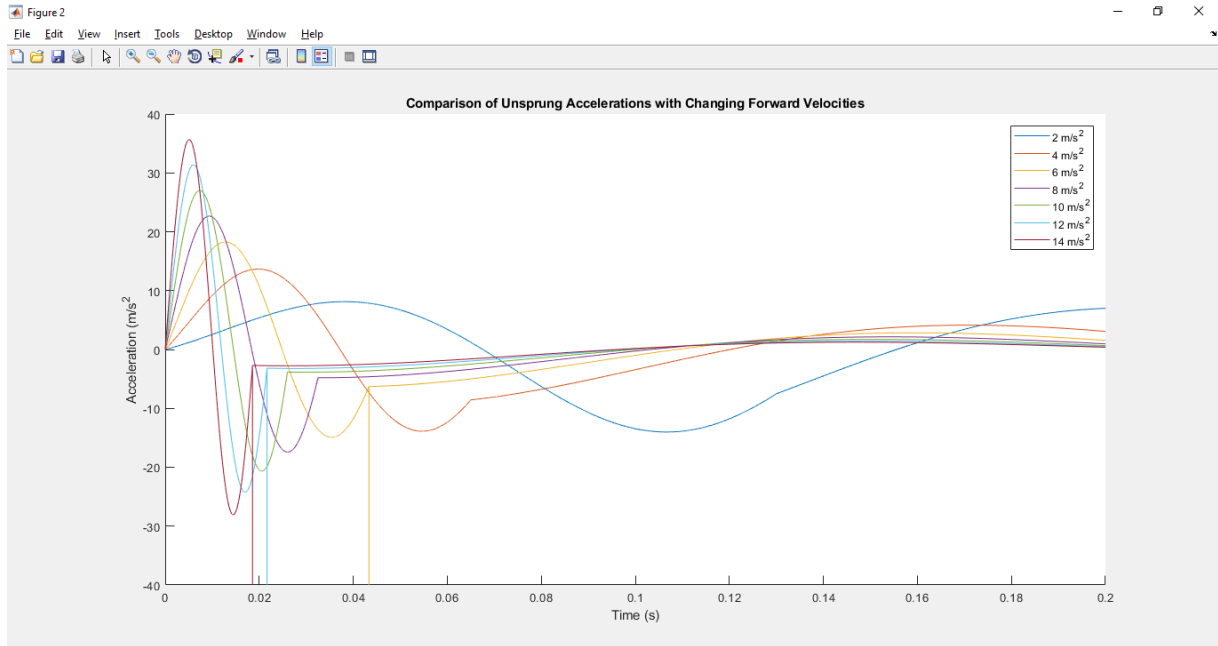


Figure 44 - Comparison of Front Unsprung Accelerations with Different Forward Velocities

Parameter	Value (m/s ²)
Filtered Maximum Value	23.80
Filtered Minimum Value	-18.51
Unfiltered Maximum Value	35.70
Unfiltered Minimum Value	-27.76
Difference in Max Peaks	11.90
Difference in Min Peaks	9.25

Table 9 - Comparison of Maximum and Minimum Values Before and After Filtering

Figure 45 shows the effect the different forward velocity has on the magnitude, length and shape of the simulation outputs. Figure 45 is taken from the quarter car simulation, with forward velocities ranging from 2 to 14 m/s in increments of 2 m/s. As can be seen from the unsprung accelerations, (Figure 45) the faster the forward velocity, the higher the peak accelerations. However, even at 4 m/s there is a drastic change in direction of the data. It is noticeable that it converges rapidly towards zero and plateaus, with the rate of

change being larger as the forward velocity increases.

These points of high divergence from the trend of the simulation are indicative of the points at which the simulation is unable to solve the combined parameters. This can be proven by inspection of the data for 6 m/s, where the negative spike is at 0.0433s (13/300 seconds). This is the calculatable end of the step, as shown in equation 9.

Equation 9 - Equation to Calculate Time at End of Step

$$\begin{aligned} \text{End of Step @ } 6 \text{ m s}^{-2} &= \frac{0.26 \text{ m}}{6 \text{ m s}^{-1}} \\ &= 0.043\bar{3} \text{ s} \end{aligned}$$

This implies that there is an issue with either the velocity that is modelled and experimented at, or the dimensions of the step. Due to the velocities being tested still being modest (14 m/s is only approximate to 31 mph) compared to the possible velocities of a motorcycle, reason suggests that the step is a logical area for modification. To this end, the length of the step in the input parameters was adjusted to 1 metre. This value was chosen as a large increase over the original length, but not so large that it causes contact with both wheels at the same instance. The

step must be less than the wheelbase of 1.38m to eliminate interference.

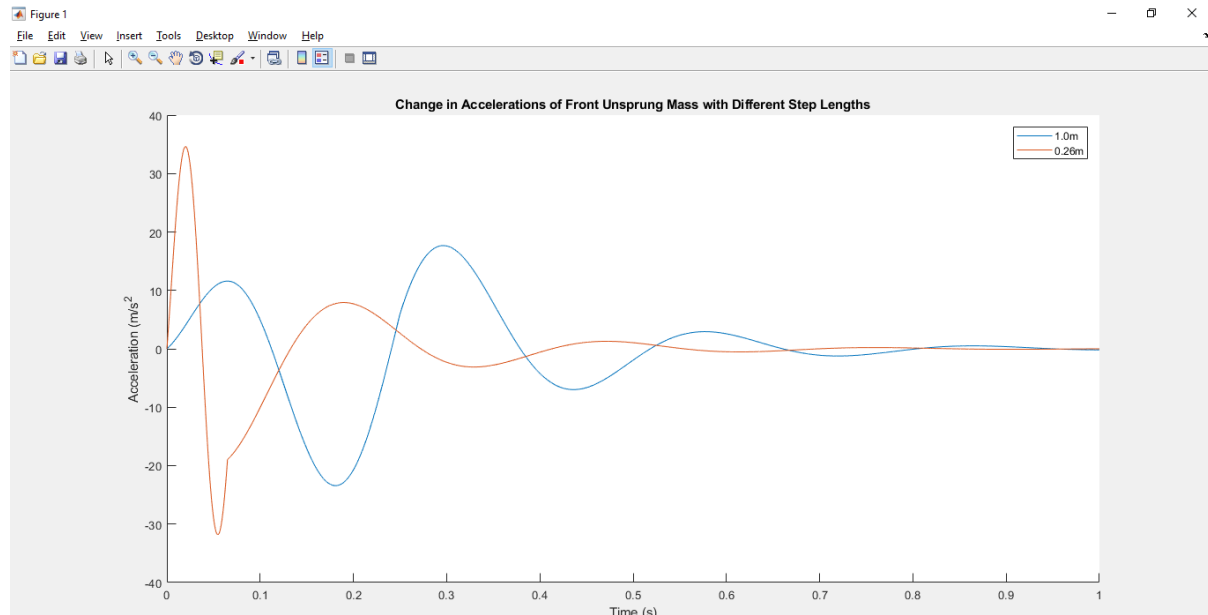


Figure 45 - Graph of Unsprung Accelerations at 4 m/s with different step lengths

Figure 46 shows plots of the front unsprung acceleration at a forward velocity of 4 m/s, with step lengths of 0.26m and 1m. As can be seen, increasing the step length allows the simulation to solve the system without errors, sharp changes in gradient and provides an output that is logical when considering the dynamics involved.

6.2 Measuring and Modelling Pitch

As explained in the methodology, due to hardware restrictions (limited number of controller input pins), the accelerometer that was placed at the COG had to be removed resulting in two issues. Firstly, the ability to measure the vertical change in COG (sprung) position was withdrawn. Secondly, it was then not possible to easily calculate the pitch of the sprung mass.

To circumvent these issues, both values were found using calculations from the front and rear sprung accelerometers. Below are the calculations that were used to find both the

COG displacement and the pitch.

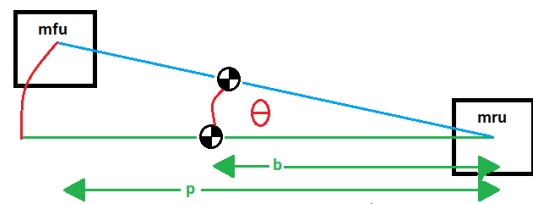


Figure 46 - Diagram Used To Calculate Pitch

By using the small angle approximation, it was possible to use the difference in front/rear displacements to find the overall vertical displacement. Then, using the known value of the wheelbase, it was assumed that the difference in vertical displacements was equal to the length of beta (Figure 48).

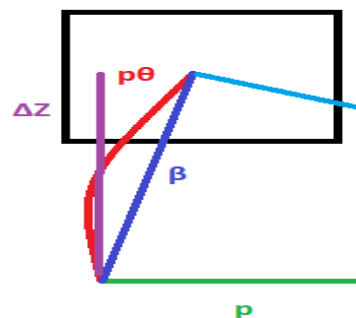


Figure 47 - Diagram Including Beta

Though assuming beta is equal to the vertical displacement does induce some error, it is

visually analysed to find variances that will help determine the usefulness and

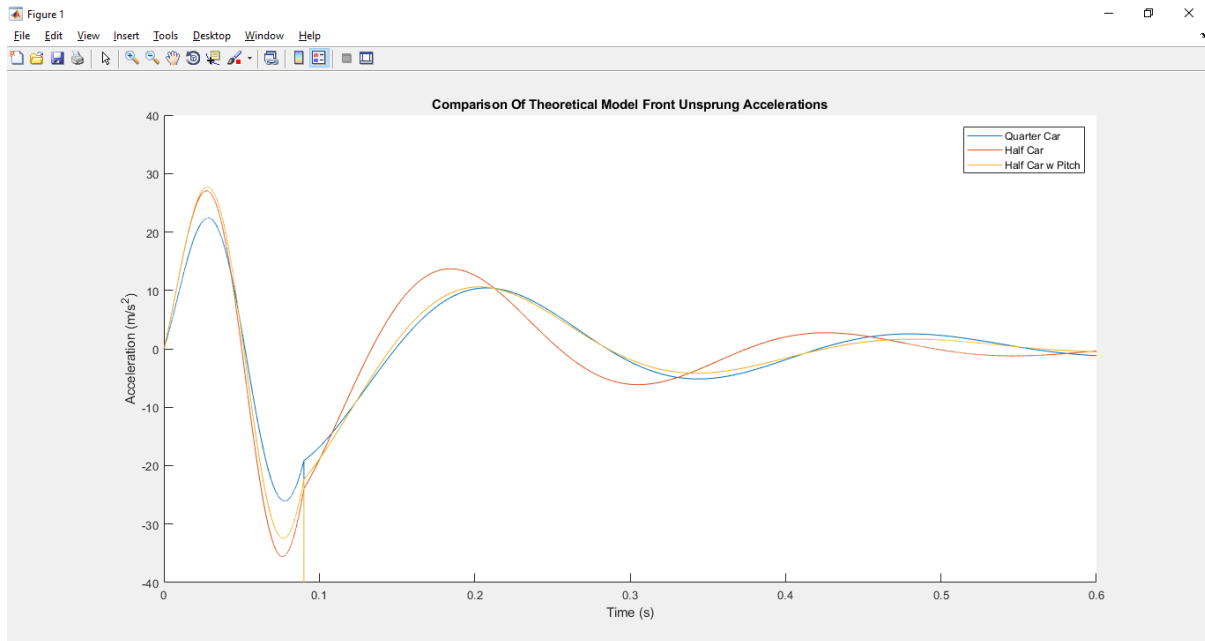


Figure 48 - Comparisons of Unsprung Accelerations from All Three Models

minimal. It was calculated to only have a percentage error of 0.38% up to a value for the pitch of 10 degrees. Once the pitch was calculated, the vertical displacement of the COG was able to be deduced. Once again using the small angle approximation, (error of 0.13%) and the value of b (horizontal distance of rear spindle to COG position), it is possible to find the value of α and hence the vertical sprung displacement.

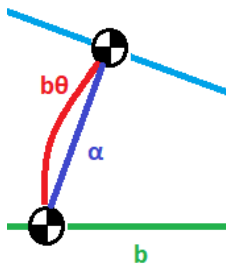


Figure 49 - Diagram Including alpha for calculating Z sprung

6.3 Model Output Comparisons

To understand the potential difference in the output of each of the models, they must be

applicability of each. This needs to be completed before comparing to experimental data, to finally determine the accuracy and advantages of each model.

6.3.1 Accelerations

As the experimental data is found through using accelerometers, acceleration is the most important parameter to be analysed. Velocity and displacement calculations are found through integration of the accelerometer output, meaning they are very sensitive to small changes in data. These changes in data, which are most prominent in unfiltered data, can be accentuated due to these operations, leading to compound errors.

Figure 51 is a graph of the three models front unsprung acceleration output. As can be seen by the data, the differences in maximum and minimum values of the two half car models is extremely small for the peak positive acceleration. However, a slightly larger

difference is apparent in the peak negative acceleration (3 m/s^2).

For the quarter car model, the peak values are inconsistent with the other models. Compared to the half car models they are considerably lower, with a maximum difference on the negative peak of 6.4 m/s^2 .

As well as analysing the maximum and minimum values, it is important to take note of the frequency differences. Although the half car and half car with pitch model agree on the magnitude of the maximum acceleration (within a 0.67 m/s^2 difference), they calculate different frequencies that would be caused by the step input. The only difference in the two models is the introduction of the pitch motion in the

When comparing the velocity of the front to the rear unsprung masses such as in figure 52, it is seen that they are very close to being equal in shape. This is with the frequency of the rear being slightly lower, as well as the magnitude of the velocity of the rear being less than that of the front unsprung mass.

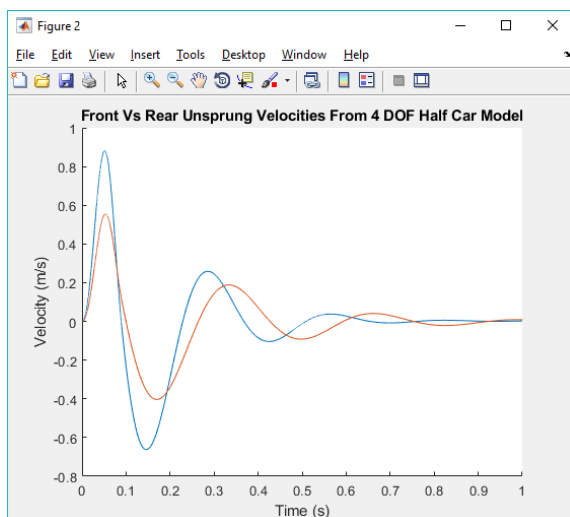


Figure 51 - Graph of Front Vs Rear Unsprung Velocities

calculations. Therefore, it is reasonable to deduce that the addition of pitch causes a lower front unsprung frequency than that of the half car with pitch not included.

6.3.2 Velocities

Through integration of the accelerometer data it is possible to obtain the velocity in the Z direction for the suspension system. This is important as for the majority of motorcycles, the damping force is proportional to the velocity. Figure 52 shows the velocity trace for the relative velocity between the sprung and unsprung mass. This is the velocity that the damper will experience and that when multiplied by the damping coefficient will determine the damping force.

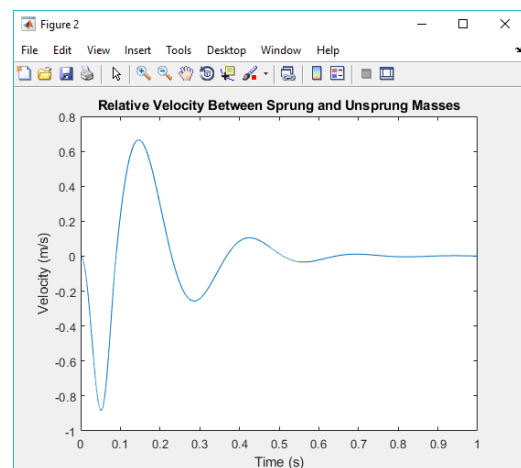


Figure 50 - Relative Velocity Between Front Sprung and Unsprung Mass

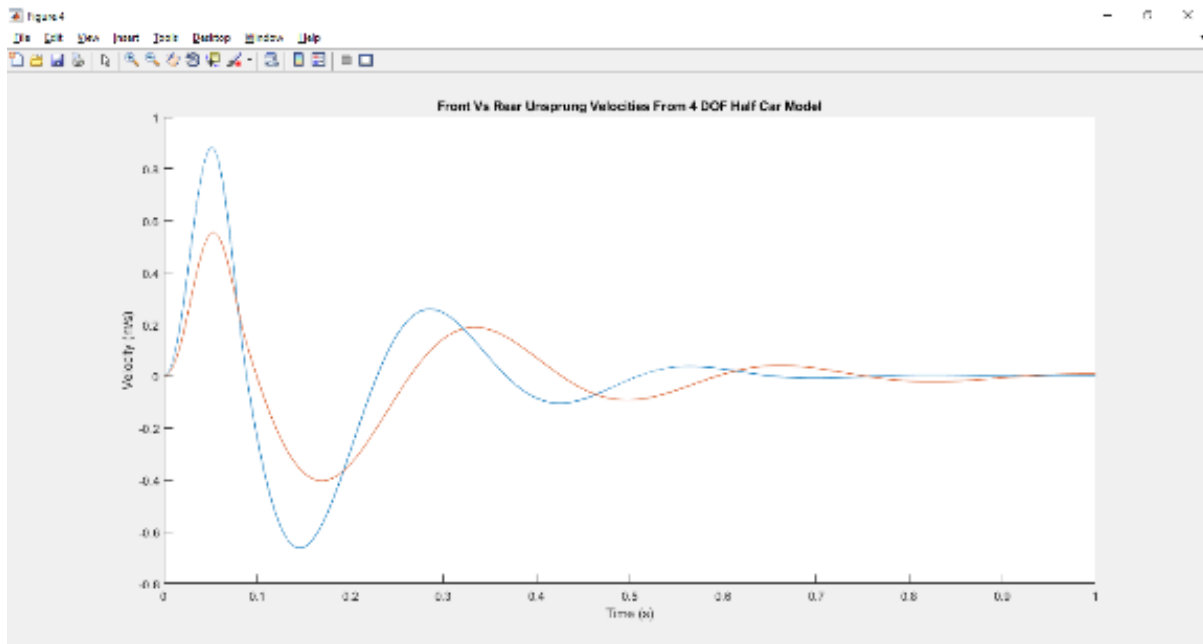


Figure 52 - Front Vs Rear Unsprung Accelerations

When analysing the front sprung mass, it is immediately apparent that the pitch of the sprung mass has a large impact on the initial oscillations of the motorcycle being studied which can be in figure 54, where it shows the first oscillation is affected by introducing pitch.

Figure ?? displays the angular velocity of the pitch of the motorcycle at the test velocity. Superimposed is the location of the start, peak and end of the step, for both the front wheel climbing the step as well as the rear.

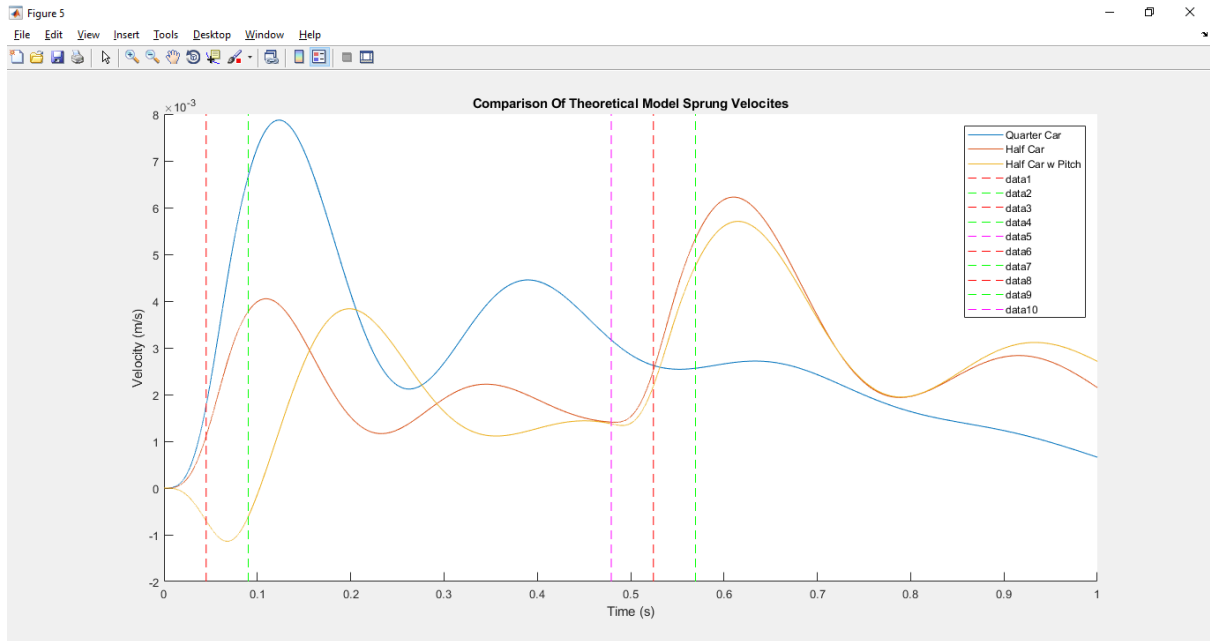


Figure 53 - Comparison of the Sprung Velocities of the Three Models

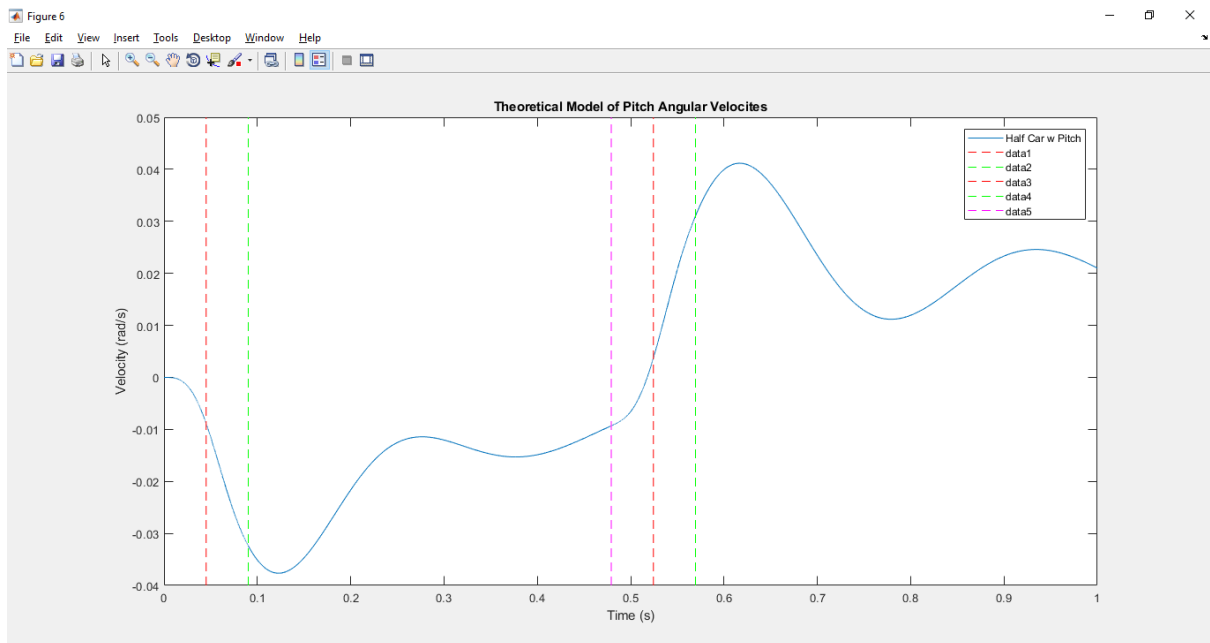


Figure 54 - Graph of Pitch Velocity with Start, Mid and End of Step Labelled

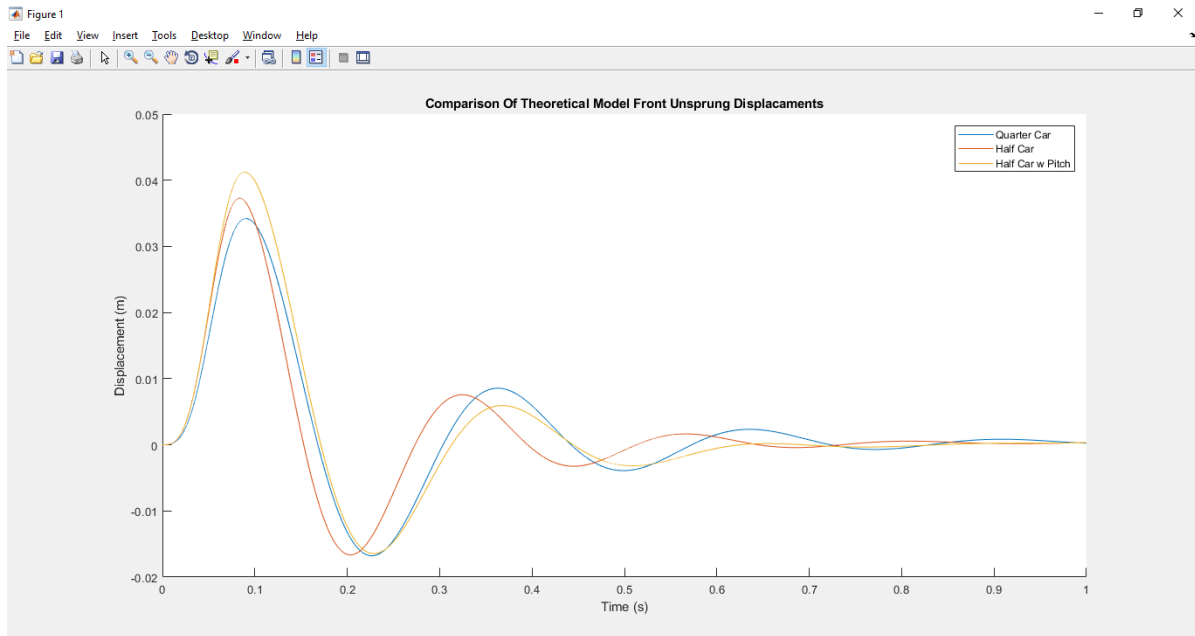


Figure 55 - Comparison of the Front Unsprung Displacements of the Three Models

6.3.3 Position

When comparing the displacement of the front and rear unsprung masses, there is not a large degree of difference between the models, as can be seen by figure 55. For the negative portion of the first oscillation, the difference between the models is within a range of 0.33mm. Compared to this, the positive portion of the oscillation stated before, the maximums of the quarter car and the half car with pitch differ by over 7mm at their peaks. This can be accounted for by the lower sprung mass of the quarter car, resulting in larger motion of the same sprung mass. This is shown by the velocities reached, and therefore the kinetic energy shared between the two masses. Leading to a higher proportion distributed to the sprung mass, giving rise to a larger displacement.

As the pitch of the motorcycle has not been directly measured using gyroscopes, it needs to be calculated. Figure 56 shows the pitch angle with respect to time. This can be

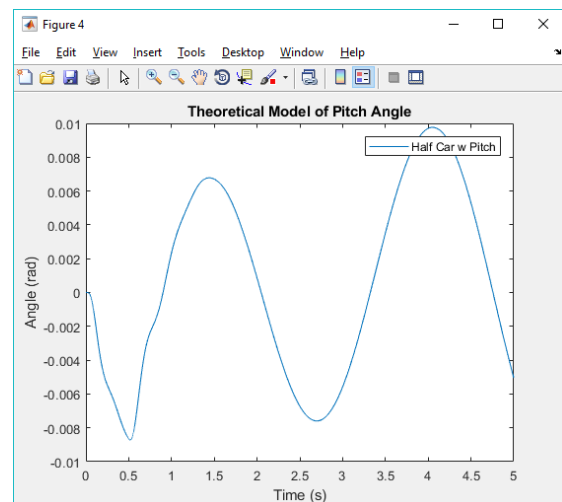


Figure 56 - Pitch angle Found from Half Car Model

completed using the small angle approximation. This approximation is based on the small difference in values obtained between assuming that for a small angle, the chord length is equal to the arc length.

To obtain the equivalent front sprung displacement, the difference is found between the front and rear sprung masses (the numerator). Using equation 10 and rearranged for theta, the pitch angle can be found.

Equation 10 - Equation for Pitch Angle

$$\theta = \frac{Z_{fu} - Z_{ru}}{p}$$

Equation 11 shows by using length 'b' (the linear horizontal distance from the rear contact patch to the COG) and the pitch angle, the displacement of the COG can be found.

Equation 11 - Equation for COG Displacement

$$Z_s = b \cdot \theta$$

It can be seen from figure 57 that before the front tyre loses contact, the pitch angles are predicted and those found by measurement are close with a small amount of error. Although not matching exactly, the general trend is very similar.

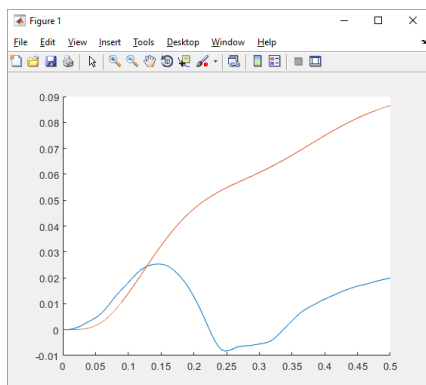


Figure 57 - Comparison of Calculated Pitch vs Model Pitch

6.4 Second Oscillation from Impact

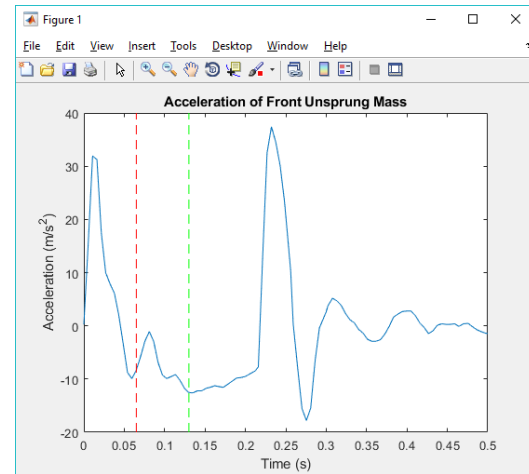


Figure 58 - Acceleration of the Front Unsprung Mass from Experimental Data

On inspection of the data, and as discussed in section 6.1, it is seen that the half car with pitch model and the experimental data do not match up as expected. There is a second oscillation that does not correspond to a continuation of the first oscillation (referred to as peak or oscillation 'A') due to contact with the step. At the point of the second peak (occurring 0.2322s later) the damping effect has had enough time to reduce the size of the initial oscillation to a much lower magnitude than is found from the unexpected peak (hereon referred to as peak 'B'), and what is predicted by the models. This can be seen from Figure 58, where it can be observed that the magnitude of the acceleration of peak B is larger than the acceleration caused by the initial contact with the first half of the step. Superimposed on the graph are dashed lines with red indicating the peak of the step and green corresponding to the end of the step.

The first theory that was analysed to explain peak B was that it was caused due to the pitch of the motorcycle from the rear tyre hitting the step, creating an interaction with the front unsprung mass. However, this was soon ruled out as for the slow speed, mid damping experiment, the forward velocity of the

motorcycle would have to equal 6.23 m/s (equation ??), which does not match the data from the front Hall sensor (2.89 m/s).

Equation 12 - Calculation for Velocity from Between Front and Rear Wheels Contacting the Step

$$V = \frac{p}{t_B - t_A} = \frac{1.38}{0.2322 - 0.0108} = 6.23 \text{ m s}^{-1}$$

To resolve this anomaly, a slow-motion camera was used to record the wheels interaction with the step. This soon showed

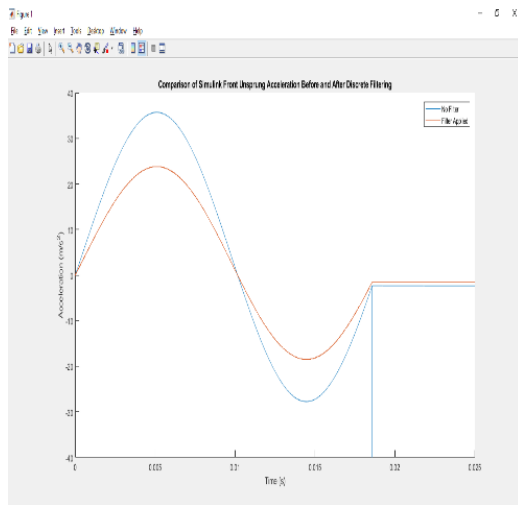


Figure 59 - Comparison of Model Data With and Without Filtering

how the tyre, after reaching the peak of the step, would leave contact with the step, and land after the end of the step on the floor at a vertical displacement of zero. After analysing the time difference between the tyre reaching the peak of the step and contacting the floor again, the times were found to coincide with the experimental data. Due to this second oscillation corrupting the last portion of the data, only the first half of the oscillation can be analysed. This causes errors to arise, as the frequency of the data can no longer be calculated to a reasonable degree of accuracy, and hence have been removed from proposed calculations.

6.5 Numerical Model Analysis

To get an accurate result from the models, without loss of data through post-simulation filtering, the negative spikes induced in the simulation have not been removed. This is because when comparing the experimental data to the results of the models, the peak values have a lower difference without filtering, than compared to filtered data. For example, figure 59 shows the same model, with the only change being a filter applied before the output to the scope in Simulink.

The data will not be filtered, and the negative spike will be removed manually by deleting the corresponding element in the output vector, as the spike is composed of only one anomalous data point. The data will only be modified this way if numerical data analysis requires it. For basic comparisons of minimum and maximum points, the minimum will be found by inspection.

As a basic comparison of the models and data, the minimum and maximum values were found to compare the accelerations, velocities and positions that are achieved. Below is the maximum and minimum acceleration data for the models.

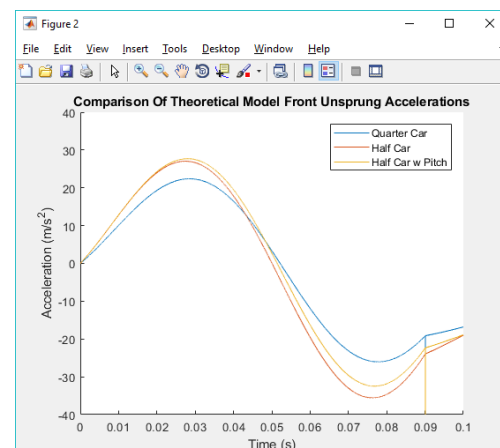


Figure 60 - Comparison of Front Unsprung Accelerations Between the Three Models

MAX	Quarter Car	Half Car	Half Car w Pitch
Front US	22.39	27.04	27.71
Sprung	0.1246	0.0726	0.065
Pitch	N/A	N/A	0.6858
Rear US	N/A	16.76	16.83

Table 10 - Table of Maximum Values Reached By Each Model

MIN	Quarter Car	Half Car	Half Car w Pitch
Front US	-26.04	-35.56	-32.47
Sprung	-0.066	0.0391	-0.0346
Pitch	N/A	N/A	-0.6057
Rear US	N/A	-16.69	-16.64

Table 11- Table of Minimum Values Reached By Each Model

noticeably different on the graphs in figure 61.

The difference between the sprung accelerations of the two half car models equals a 17.2% difference. This equates to the quarter car model being 4.65 m/s^2 less than the half car without pitch, and 5.32 m/s^2 less than the half car with pitch included as shown in Figure 60.

When comparing the differences in the models unsprung accelerations to the data's peak accelerations, the half car models are a lot closer to each other, as well as the peak. This can be seen as the maximum of the data after filtering is in between the maximums of the two half car models, this is shown in figure 62.

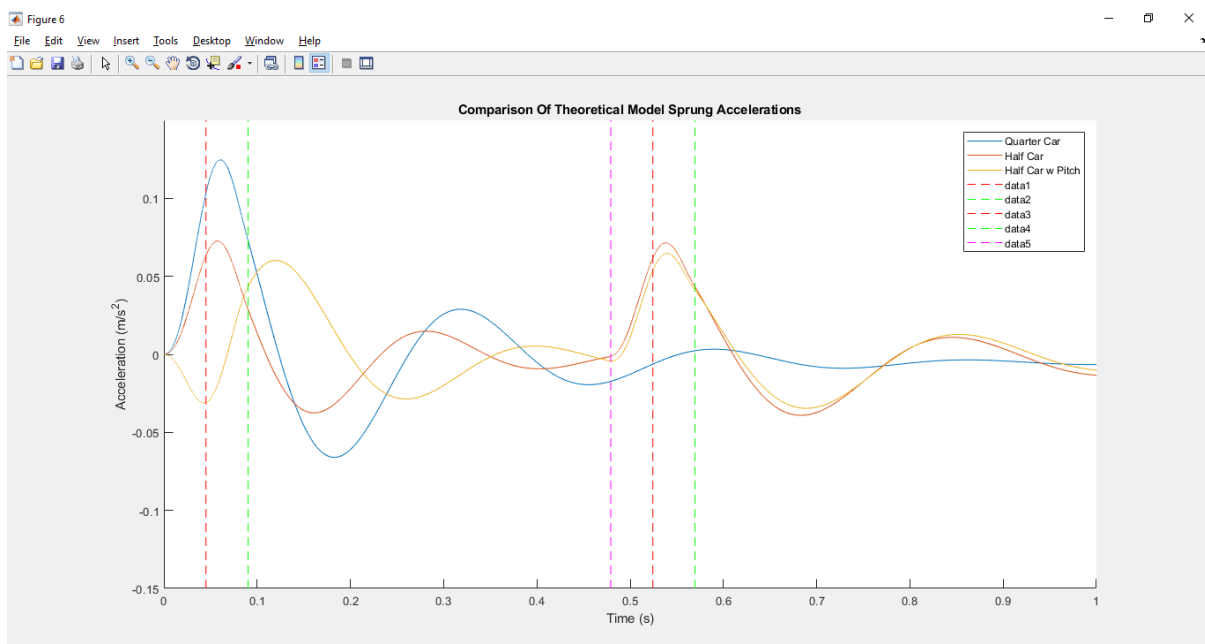


Figure 61 - Graph of Sprung Accelerations from the Three Models

As can be seen, the difference between the front unsprung accelerations of the half car models is very small, with a percentage difference of less than 2.5%. The difference between the half car without pitch and the quarter car models is much larger, and

This trend continues in the sprung accelerations, as well as the velocity and position datasets. Interestingly, when looking at the data for sprung accelerations, even

though the maximum and minimum values are closer in the two half car models than the quarter car, the graphs do not follow the same general trend. This can be seen in figure 61 where the two half car models are seemingly out of phase with each other until 0.5 seconds. This graph is visible proof of the effect of pitch on the sprung mass. At 0.05 seconds, the half car with pitch model is a negative. This part of the oscillation reaches a largest magnitude at the point of the wheel reaching the top of the step (shown by the red dashed vertical line).

phase but with general equal maximums in the sprung mass, continues with the velocities. In the trace for the unsprung mass, the quarter car model still reaches a lower maximum than the half car models, which again are within 5% of each other. However, the visible difference in shape and magnitudes is lower in the velocity data than the acceleration data. Regarding the sprung mass, the quarter car velocities are much higher than those of the half car models. This may be due to the lower sprung mass, but when modifying the spec sheet to change the sprung mass of the quarter car model to 218kg (the full sprung mass), the velocity trace drops significantly, as shown in figure 63. This suggests that there is a mass between 109kg and 218kg, that would give a similar output to the half car models.

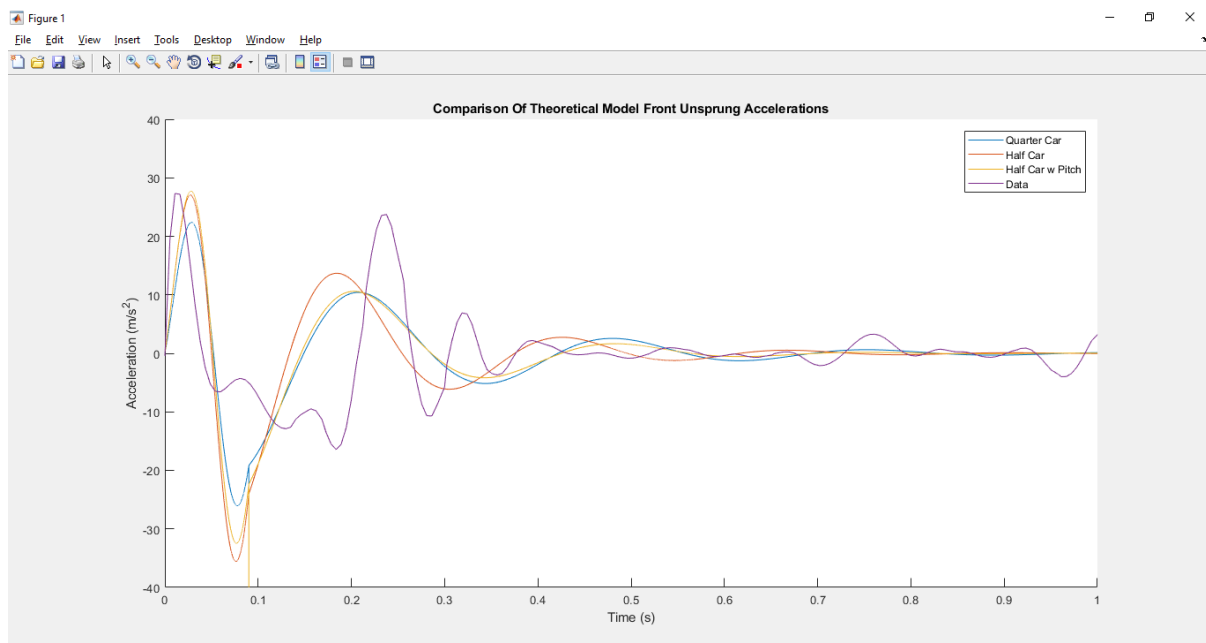


Figure 62 - Models Vs Experimental Data for Unsprung Accelerations

Both the trends stated above, regarding the two half car models being closer in shape and magnitude in the unsprung mass, and out of

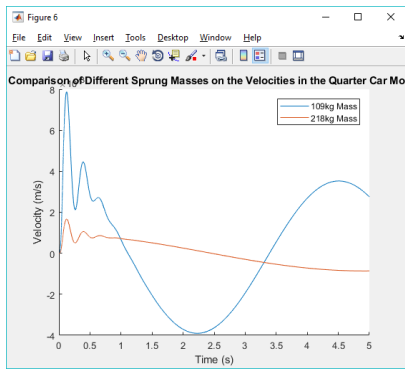


Figure 63 - Full vs Half Sprung Mass Quarter Car Velocities

Once again, when considering the sprung mass, there is a difference when including and excluding pitch. When looking at figure 62 it is visible that the velocities are out of phase, matching the equal trend in the accelerations. The trend between the acceleration and velocity even matches to the point that the phase shift reduces to a minimum at the same point, at the point of the rear beginning to climb the step. This trend that is visible in acceleration and velocities, is not visible in the displacement. However, there is a slight negative displacement on the graph at the same point as the negative velocity. Although, this is small enough that compared to the general difference in the two datasets, it cannot be considered a trend. In the unsprung displacements the phase shift trend is still visible, but much later in the displacement than the acceleration, with the separation occurring at 0.1 seconds in the acceleration and 0.2 seconds for the position.

6.6 Induced Errors, implications and solutions

Due to time constraints and issues with the experiments, direct data for the values of the rear tyre stiffness and both tyre damping coefficients were assumed based on data acquired from Dr Owen Williams (2018). The data that was acquired was from a damping test conducted on a set of tyres equal to those used in this experiment and of similar construction.

The front tyre was physically tested to find the stiffness. This was conducted using a hydraulic press to compress the tyre with a set of scales underneath. This was to find the force exerted by the press as the pressure gauge was not precise enough for the required calculations. A floor standing Vernier was used to measure the compression of the tyre. This gave the result of 163000 N/m to 3 significant figures. This value was used for the rear tyre due to there being a fitment issue with the hydraulic press making it impossible to test the tyres the same way. This could induce error into any related calculations.

For the tyre damping data, the data from the tyre test conducted by Dr Williams was used. The value for zeta (damping factor to determine the degree to which the tyre is underdamped) was then used with the theoretical calculations in Equation 2 for a vertical ideal damper to obtain a damping coefficient value. The value of zeta was theoretically verified using the accelerometer data for the front unsprung mass. The acceleration data was integrated to give the velocity (Figure 64) and position (Figure 65).

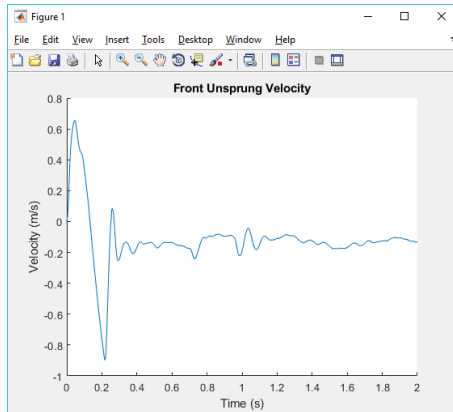


Figure 64- Graph of Velocity of Front Unsprung Mass

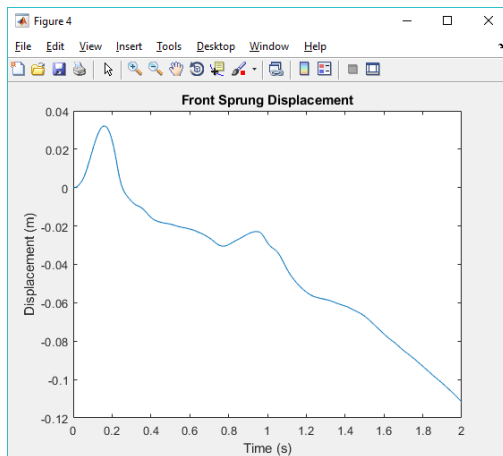


Figure 66 - Front Sprung Displacement

This data was then plotted against data created to be likely values for velocity encountered (0 to 0.08 m/s, similar to velocities achieved on the damper dynamometer). The derivative was found and averaged to give a single value for zeta. This was then inputted into the Simulink model and the changes in the model analysed. In future work, a variable value of zeta for each velocity could be used to further improve the accuracy of the model.

To determine the forward velocity of the motorcycle, a hall sensor was used as a wheel speed sensor. This proved to give a wide

range of different wheel speed velocities to those realistically achieved. In figure 66 you can see the graph for the wheel speeds calculated from every third of a rotation of the front wheel.

It is easy to see that the results vary. To try and get a useable value that agrees with logic, the wheel speed was averaged across the entire test. This value agrees with the speed taken from the slow-motion video capture of the wheel going over the step. The value also directly contradicts the values obtained using the hall sensor. This is shown in table 12 where the minimum, mean, maximum and average over the entire time taken are compared, with the last point being different to the mean.

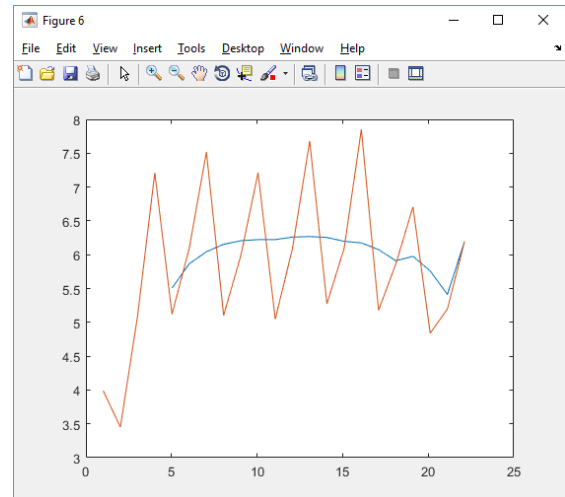


Figure 65 - Graph of Velocities Derived from Hall Sensor

Method Obtained	Velocity
Maximum	7.86
Mean	6.25
Minimum	4.36
Average Across Test	2.89

Table 12

As can be seen, the data from the hall sensor when obtained using an instantaneous calculation is unreliable, and hence it was not used. Instead, all calculations for forward velocity was used giving more accurate results for 2.89 m/s.

7 Discussion & Evaluation

During this research, certain issues presented themselves which complicated the matter of obtaining the results required. One of the most important issues, was that it was realised that the dimensions of the step were too aggressive to obtain results for a controlled climb and fall of the step. Instead, the wheel would reach the top of the step, and continued on a trajectory that was not the same as the step, leading to loss of contact. This presented an issue as the data could not be obtained for the second half of the step as the suspension and the tyres were extending. In terms of the data recorded, half of the graphs were not useable, and this also effected the ability to visualise the decay of the oscillation.

The step dimensions did not only cause issues with the experimentation but also with the modelling. The tyres needed to be modelled as at the top of the step, a large negative spike would be calculated instead of the controlled change from positive to negative gradient. This would cause Simulink to result in an error, and the graph would change direction sharply, and return to zero in a way not fitting with the other models. The models that did not exhibit this error in the calculation phase were all at lower velocities. This led to the conclusion that for accurate results to be obtained in future work, a different step with a lower length to height ratio would have to be used. This conclusion was tested by repeating the mathematical model with a step that was 1m long compared to the original 0.26m but using the same height. At the velocity used for the first set of models an error would be produced. However, after changing the length of the step and repeating the models the error was no longer produced.

This shows a positive correlation between the length of the step, and the velocity achievable before computation becomes too complex for reasonable error tolerances.

Being able to calculate what the velocity of the bike was at the point of reaching the step also proved difficult. The original plan was using a hall sensor with magnets attached to the wheel every third of a revolution, from which the velocity could be derived. However, this method proved problematic as the hall sensor was picking up extra signals, leading to higher velocity readings, outside what was attainable using the experimental method. To circumvent the problem, an average velocity was taken from the first signal received by the sensor to the last and averaging across the time taken between these two signals. This gave an average velocity attained throughout the test which closely agreed with that found through using the slow-motion camera. Although this method does reduce the accuracy of the final output, when using similar values, it was found that through the two methods it was deemed acceptable and would not have a large effect on the final output. Although, if the experiment was to be repeated, a professional wheel speed sensor would be used. This would be done with a reluctor ring, similar to that used in ABS systems as they are required to determine the wheel velocity for small rotations of the wheel.

7.1 Assumptions

To be able to attain both experimental results and results through mathematical modelling in the timeframe provided, certain assumptions were made. Some of these assumptions were only temporary and were corrected later such as the

damping values, whereas others were used throughout the research.

Below are the general assumptions made in developing the model;

- Forward velocity was equal to average velocity
 - The difference in front/ rear displacement is equal to the wheelbase multiplied by the pitch angle
 - The COG displacement is equal to b multiplied by the pitch angle
 - Tyre damping values are equal to that of other equal sized tyres at equal pressures
 - Tyre stiffnesses are equal front and rear
 - Motorcycle can only move in the X, Z and Pitch DOF
 - Linkage ratio is equal throughout stroke of rear shock absorber
 - Dynamic rake angle stays equal throughout test
-
- What could be changed to improve error

8 Conclusion

Although there were issues with both the experimentation and the mathematical modelling, it has not stopped important information from being gained. Through the numerical analysis it has been shown that there are strong correlations between the results of the experiments and the models. It has also shown that as well as being able to determine the pitch, which is considered an important benefit, the 4 DOF model is also the closest related to the results obtained. This

has been shown through analyses of the acceleration, velocity and displacements.

Application of this research is wider spread than what might first be considered. Currently motorcycle manufacturers are still using 'trial and error' methods to determine the best set ups for road going motorcycles. Given that, it has been shown that even a relatively simple model can give close results to those gained from research using sensors and electronics. This suggesting that it has a place in the development of motorcycles. This development can either be in the race sector or the OEM manufacturing sector.

As experimentation to a high degree of accuracy can be time consuming and expensive to get reliable results, any method that can reduce that time is well invested. Multi body codes are already developed in the research sector but not widely used by OEM manufacturers. This research suggests that theoretical specifications and changes could be inputted into a mathematical model and the differences analysed.

One of the novel uses of the experimental data acquired was to determine the damping coefficients that are being used in the shock absorbers currently in use. This has yet to be verified using equipment such as a damper dynamometer but was tested by comparing the results with assumed damping coefficients. The results of the model using the derived coefficients made the model and experimental results much closer in various critical areas. This, and the sensor system used to obtain them, has applications such as the use of magnetorheological dampers, active and semi active suspension systems that are currently being applied to new motorcycles.

The area that has been chosen for this research is vast and ever expanding. Much interest has arisen in motorcycle dynamics, active suspension and other safety systems. The research conducted here could lead on to various avenues with applications across the

industry. Future research on this subject however, would include using more sophisticated, accurate and precise measuring systems to be fitted to a motorcycle being used both in and out of plane conditions. When factoring in out of plane dynamics more applications and techniques are available for analysis. These include frame flexibility, which has been shown to influence the vibration performance on track of race motorcycles due to the different needs of a motorcycle frame in lateral and longitudinal

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