

TRANSIENT RESPONSE

MODELLING OF A VEHICLE

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1. ABSTRACT

This paper deals with the operation of spring mass damper system and how it is used to assess the transient response of a vehicle undergoing a standard lane change procedure. SMD models varying from 1-DOF to 7-DOF were developed using MATLAB and Simulink and the lateral and longitudinal acceleration data from a Formula Ford Race Car simulation were fed into the models, after acquiring them using MoTeC. The lane change manoeuvre was done for three setups, each having a different front and rear spring stiffnesses: Setup 1- 14 N/mm and 20 N/mm, Setup 2- 18 N/mm and 16 N/mm and Setup 3- 23 N/mm and 12 N/mm. for the given vehicle, it was determined that Setup 2 had the lowest overall pitch angle and lowest roll with less oscillations. Also, the graph for roll and lateral acceleration graphs for all the setups were compared to see the correlation between them.

2. INTRODUCTION

A spring mass damper (SMD) system (Edwards, 2001) is a model that is made up of discrete mass nodes of an object that are connected together through a system of multiple springs and dampers. The springs in this system provide a restoration force in the direction opposite to its displacement, i.e., if a spring experiences a compression due to external force, it reacts to create a counter force to expand again to its original position and vice versa. This restoration force can be defined as the product of the spring rate of the spring and the amount of deflection of the spring.

$F_s = k * x$, where F is the force, k is the spring rate of the spring and x is the displacement of the spring.

The dampers in this system are used to regulate the oscillations that the object undergoes. This is done using the damping force of the dampers. The damping force is used for slowing down or stopping the motion of the object. This force can be given as the product of the damping coefficient of the damper and the velocity of its motion.

$F_d = c * V$, where F is the damping force, c is the damping coefficient of the damper and V is the velocity of the motion of the damper. A damper is only effective when there is motion as the damping force depends on the velocity of its motion.

In this assignment, this spring mass damper system is used to understand the transient response of a vehicle when the vehicle undergoes a standard lane change procedure. For this, the basic SMD models are developed for multiple degrees of freedom, starting from 1-DOF to 7-DOF using MATLAB and Simulink (Sharma et al., 2014). Upon building the models for different degrees of freedom, the data from a Formula Ford Race Car is used to assess the dynamic response when it undergoes through pitch and roll. The data of the car such as mass,

spring rates, track of wheels, COG etc. are given. For the pitch and roll values, the data was gathered from the simulator using MoTeC.

3. Theory

3.1. 1-DOF MODEL

A 1-DOF model consists of a mass that only has the ability to move to and fro in one direction or axis. All the forces that act on the body are only affected by the mass of the body. This mass is attached to a set of spring and damper as shown in Fig. 1. below.

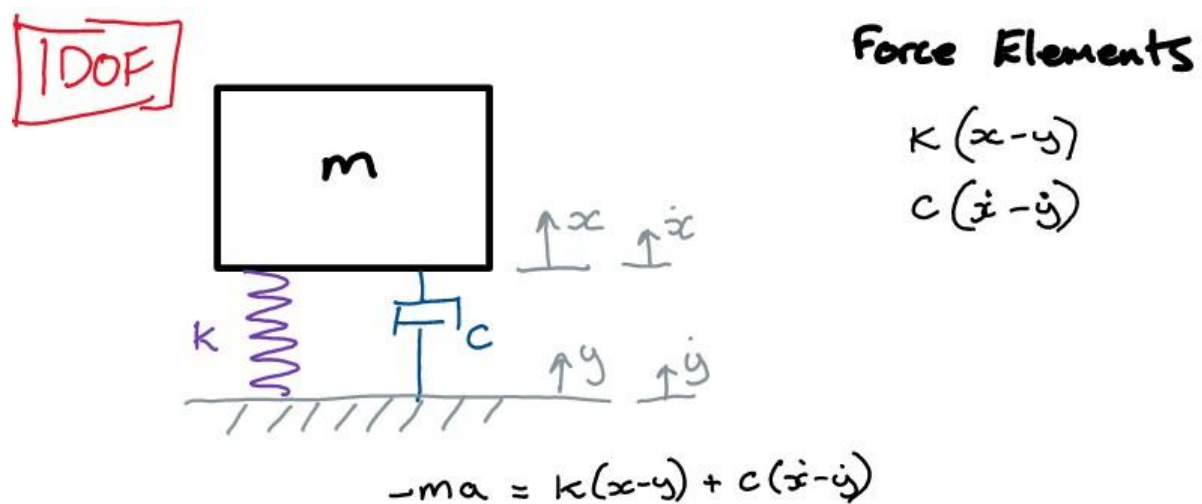


Fig. 1: 1-DOF Model

Here, m represents the mass of the body.

We know, the force acting on a body can be given as the product of mass and the acceleration of the body, i.e., $F_m = m * a$, where a is the acceleration of the body. Also, we know the forces exerted by the spring and the damper, as mentioned in Section 2.

For equilibrium-

$$F_m + F_s + F_d = 0$$

As the forces exerted by the spring and the damper are in the direction opposite to the movement of the body, hence-

$$F_m - F_s - F_d = 0$$

- $F_m = F_s + F_d$

But the acceleration of the body is negative.

- $-ma = k(x-y) + c(\dot{x}-\dot{y})$

Where, $(x-y)$ is the displacement and $(\dot{x}-\dot{y})$ is the velocity of the body.

3.2. 4-DOF MODEL

The 4-DOF model is also known as the proper half car model. In this model, the masses of the car are divided into sprung and unsprung mass. The body of the car is considered to be the sprung mass. The wheels are considered to be the unsprung mass. The sprung mass has one degree of freedom. The front wheels and the rear wheels are paired together in this model and constitute the two unsprung masses, giving two more degrees of freedom. Since the front and rear tyres can move independently of each other in this model, hence this gives the car the ability to pitch up and down. The pitch angle is given as ' θ '. This gives the car another degree of freedom, completing the 4-DOF. The point about which the car pitches is known as the pitch centre of the car. The location of the pitch centre can be given in relation to the two unsprung masses. The distance between the front wheels and pitch centre is given as ' a ' and the distance between the rear wheels and pitch centre is given as ' b ' the same can be seen below in Fig. 2.

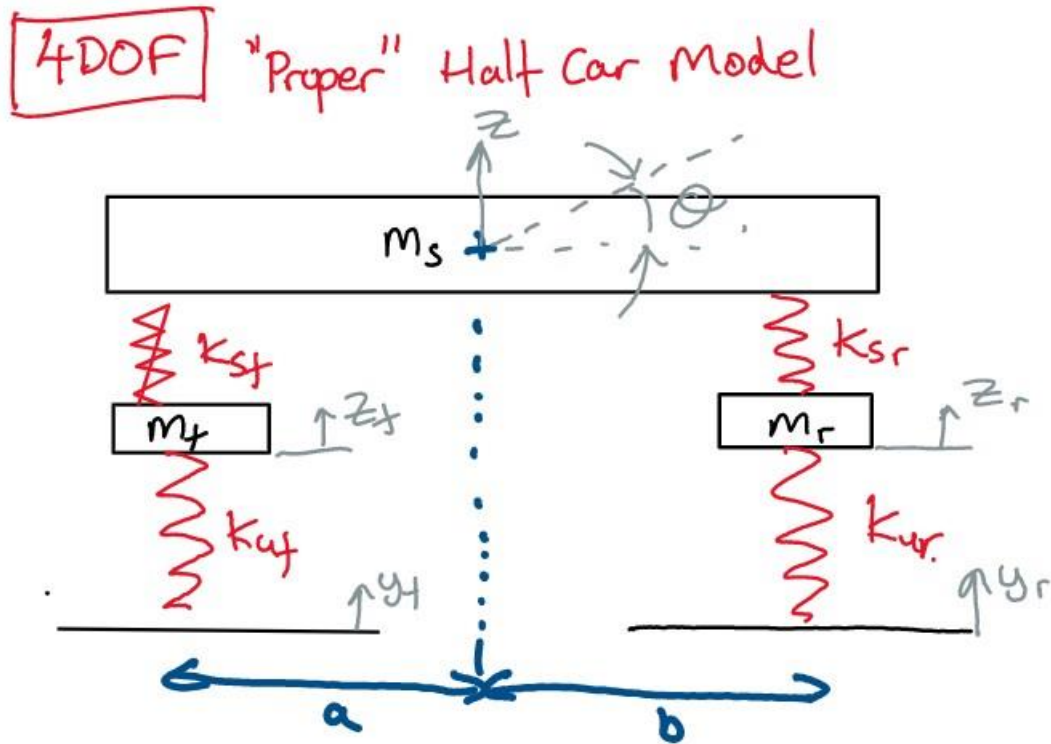


Fig. 2: 4-DOF Proper Half Car Model (without dampers)

Here, the suspension of the vehicle is considered to be the spring and damper system connecting the unsprung masses to the sprung mass, i.e., the wheels to the body of the vehicle. Since the tyres of the vehicle have the ability to deform, they act as a spring and damper system anyway. Hence, they are considered to be the spring and damper system that connects the wheels to the road surface, as shown in Fig. 3.

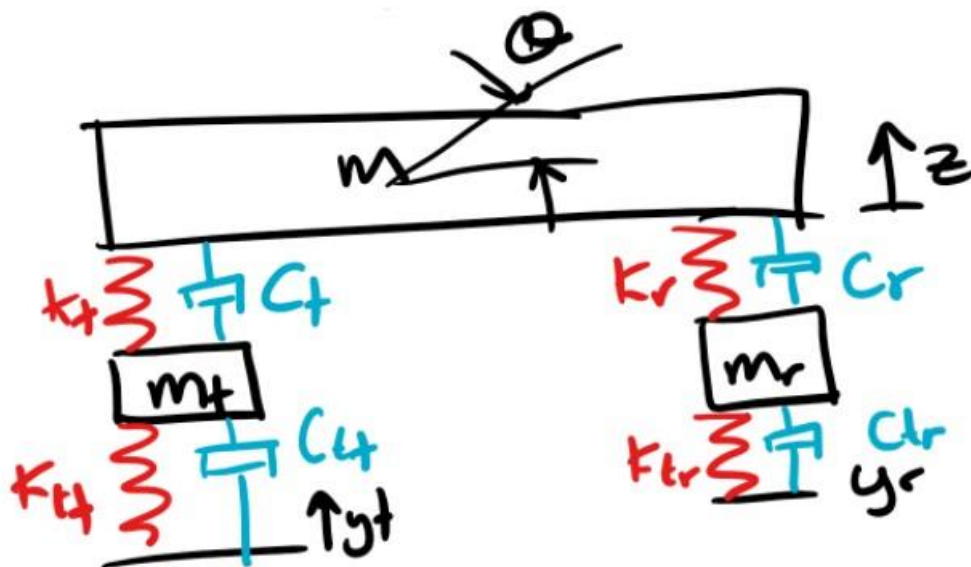


Fig. 3: Proper Half Car Model (with dampers)

The equations for the 4-DOF model are:

- $-M_s \ddot{z}_s = k_f (z_s - z_f - a\theta) + k_r (z_s - z_r + b\theta) + c_f (\dot{z}_s - \dot{z}_f - a\dot{\theta}) + c_r (\dot{z}_s - \dot{z}_r + b\dot{\theta})$
- $-I\ddot{\theta} = b k_r (z_s - z_r + b\theta) - a k_f (z_s - z_f - a\theta) + b c_r (\dot{z}_s - \dot{z}_r + b\dot{\theta}) - a c_f (\dot{z}_s - \dot{z}_f - a\dot{\theta}) + M_s a_x h$
- $-M_f \ddot{z}_f = k_{tf} (z_f - y_f) - k_f (z_s - z_f - a\theta) + c_{tf} (\dot{z}_f - \dot{y}_f) - c_f (\dot{z}_s - \dot{z}_f - a\dot{\theta})$
- $-M_r \ddot{z}_r = k_{tr} (z_r - y_r) - k_r (z_s - z_r + b\theta) + c_{tr} (\dot{z}_r - \dot{y}_r) - c_r (\dot{z}_s - \dot{z}_r + b\dot{\theta})$

3.3. 7-DOF MODEL

The 7-DOF model is also known as the full car model (Mitra et al., 2013). In this model, all the four wheels are considered to be independent. This gives us four unsprung masses along with the body of the car being the sprung mass. Since each of the front and rear wheels are independent, this gives the vehicle the ability to roll sideways along with the pitching. The roll is given as ' ϕ '. As a result, all four unsprung masses now have one degree of freedom each with the sprung mass having another degree of freedom. Lastly the pitch and roll make up the last two degrees of freedom, totalling to 7-DOF. A diagram of the full car model can be seen below in Fig. 4.

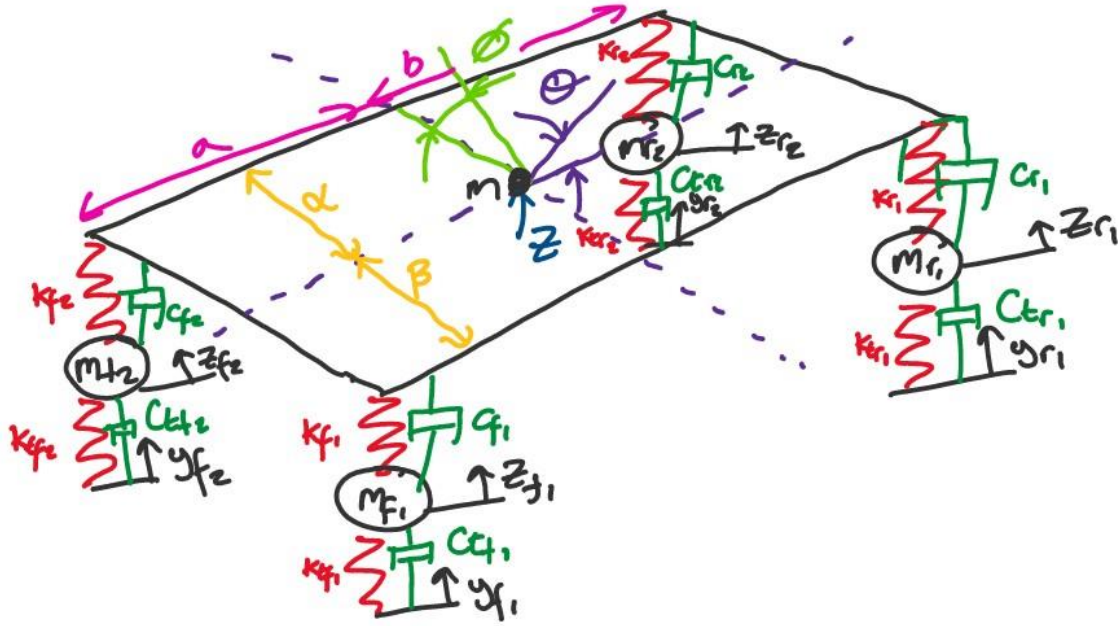


Fig. 4: 7-DOF (Full Car) Model

The equations for the 7-DOF model are:

- $-M \ddot{z} = k_{f1} (z - z_{f1} - a\theta - \beta\phi) + k_{f2} (z - z_{f2} - a\theta + \alpha\phi) + k_{r1} (z - z_{r1} + b\theta - \beta\phi) + k_{r2} (z - z_{r2} + b\theta + \alpha\phi) + c_{f1} (\dot{z} - \dot{z}_{f1} - a\dot{\theta} - \beta\dot{\phi}) + c_{f2} (\dot{z} - \dot{z}_{f2} - a\dot{\theta} + \alpha\dot{\phi}) + c_{r1} (\dot{z} - \dot{z}_{r1} + b\dot{\theta} - \beta\dot{\phi}) + c_{r2} (\dot{z} - \dot{z}_{r2} + b\dot{\theta} + \alpha\dot{\phi})$

- $-I_p \ddot{\Theta} = b k_{r1} (z - z_{r1} + b\Theta - \beta\phi) + b k_{tf2} (z_{f2} - y_{f2}) a k_{f1} (z - z_{f1} - a\Theta - \beta\phi) - a k_{f2} (z - z_{f2} - a\Theta + \alpha\phi) + b c_{r1} (z - z_{r1} + b\Theta - \beta\phi) + b c_{r2} (z - z_{r2} + b\Theta + \alpha\phi) - a c_{f1} (z - z_{f1} - a\Theta - \beta\phi) - a c_{f2} (z - z_{f2} - a\Theta + \alpha\phi) + M_s a_x h$
- $-I_r \ddot{\phi} = \alpha k_{f2} (z - z_{f2} - a\Theta + \alpha\phi) + \alpha k_{r2} (z - z_{r2} + b\Theta + \alpha\phi) - \beta k_{f1} (z - z_{f1} - a\Theta - \beta\phi) - \beta k_{r1} (z - z_{r1} + b\Theta - \beta\phi) + \alpha c_{f2} (z - z_{f2} - a\Theta + \alpha\phi) + \alpha c_{r2} (z - z_{r2} + b\Theta + \alpha\phi) - \beta c_{f1} (z - z_{f1} - a\Theta - \beta\phi) - \beta c_{r1} (z - z_{r1} + b\Theta - \beta\phi) + M_s a_y h$
- $-M_{f1} \ddot{z}_{f1} = k_{tf1} (z_{f1} - y_{f1}) - k_{f1} (z - z_{f1} - a\Theta - \beta\phi) - c_{f1} (\dot{z} - \dot{z}_{f1} - a\dot{\Theta} - \beta\dot{\phi}) + c_{tf1} (\dot{z}_{f1} - \dot{y}_{f1})$
- $M_{f2} \ddot{z}_{f2} = k_{tf2} (z_{f2} - y_{f2}) - k_{f2} (z - z_{f2} - a\Theta + \alpha\phi) - c_{f2} (\dot{z} - \dot{z}_{f2} - a\dot{\Theta} + \alpha\dot{\phi}) + c_{tf2} (\dot{z}_{f2} - \dot{y}_{f2})$
- $M_{r1} \ddot{z}_{r1} = k_{tr1} (z_{r1} - y_{r1}) - k_{r1} (z - z_{r1} + b\Theta - \beta\phi) - c_{r1} (\dot{z} - \dot{z}_{r1} + b\dot{\Theta} - \beta\dot{\phi}) + c_{tr1} (\dot{z}_{r1} - \dot{y}_{r1})$
- $M_{r2} \ddot{z}_{r2} = k_{tr2} (z_{r2} - y_{r2}) - k_{r2} (z - z_{r2} + b\Theta + \alpha\phi) - c_{r2} (\dot{z} - \dot{z}_{r2} + b\dot{\Theta} + \alpha\dot{\phi}) + c_{tr2} (\dot{z}_{r2} - \dot{y}_{r2})$

4. LIMITATIONS

4.1. 1-DOF MODEL

The 1-DOF model is good when it comes to understanding the basics of how a spring mass damper system works in conjunction with a body of a certain mass, in this case the body of the vehicle. But the limitation of this model is that it considers the entire car to be a rigid body. It also assumes that the forces acting on the car is primary due to the undulations or bumps on the road. It does not account for the many individual parts that collectively make up the vehicle.

4.2. 4-DOF MODEL

The 4-DOF model is a huge improvement in comparison to the 1-DOF model when it comes to understanding the dynamics of a vehicle. This is because it considers the front and rear wheels to be unsprung masses. But the limitation with this model is that there are just two unsprung masses. This leads to pairing the front wheels together as one unsprung mass and the rear wheels are paired as the other unsprung mass. This only allows the pitch of the vehicle to change, giving us data of how the vehicle reacts to longitudinal accelerations only. It does not take into account the lateral accelerations that the vehicle undergoes.

4.3. 7-DOF MODEL

The 7-DOF model enable us to understand the dynamics of a vehicle better as it considers all the four wheels to be independent unsprung masses. This lets the wheels on the right and left sides to undergo different forces at the same time. As a result, this does not limit the

vehicle to pitch only but also gives it the ability to roll. While this gives a comprehensive understanding of how a vehicle behaves under different forces, there are still limitations to this model. For the sprung mass, it only accounts for pitch, roll and vertical motions and not its yaw, longitudinal and lateral motions. For the unsprung masses, i.e., the wheels, only the vertical motion is taken into account and not the wheel spin. (Setiawan et al, 2009)

5. SIMULINK MODELS

5.1. 1-DOF MODEL

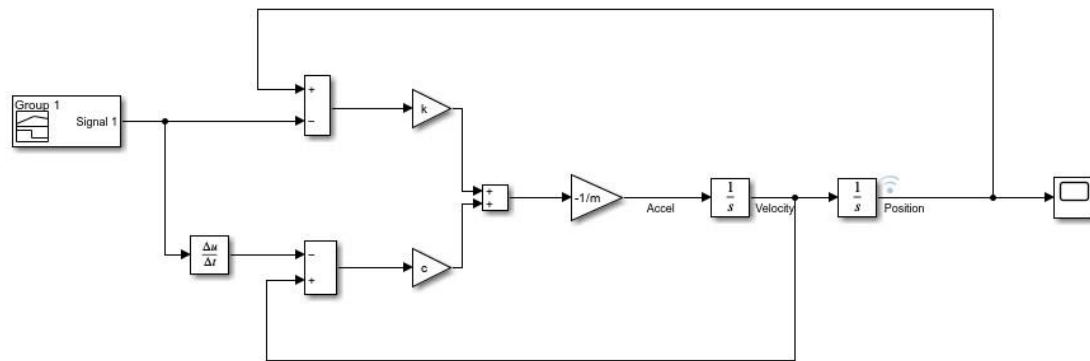


Fig. 5: 1-DOF Simulink Model

Using the equation for 1-DOF model: $-ma = k(x-y) + c(\dot{x}-\dot{y})$; the Simulink model is built.

We start with the terms on the right hand side. We have $(x-y)$ giving position and $(\dot{x}-\dot{y})$ giving velocity. For these, two sum blocks are added and their polarities are assigned. A signal builder is added as well, which gives the value of y , the amplitude of bump, along with $\dot{y}$, the speed of the vertical movement, which comes through a derivative. Two gain blocks are added and are assigned as k and c for the spring stiffness and damping coefficient respectively. These are connected to the two sum blocks. The signal from the k and c gain blocks go into a new sum block with positive polarities. Now, since we want the acceleration of the body, hence,

$$a = (-1/m)(k(x-y) + c(\dot{x}-\dot{y}))$$

So, another gain block is added and assigned $(-1/m)$, which gives us the acceleration, $\ddot{x}$. Hence, two integrators are added to give us the values of velocity, $\dot{x}$ and position, x . These are then

fed back into the two initial sum blocks. In the end, a scope is added which gives us the graph for Position vs Time.

With this, the 1-DOF Simulink model was completed. The same can be seen in Fig. 5.

5.2. 4-DOF MODEL

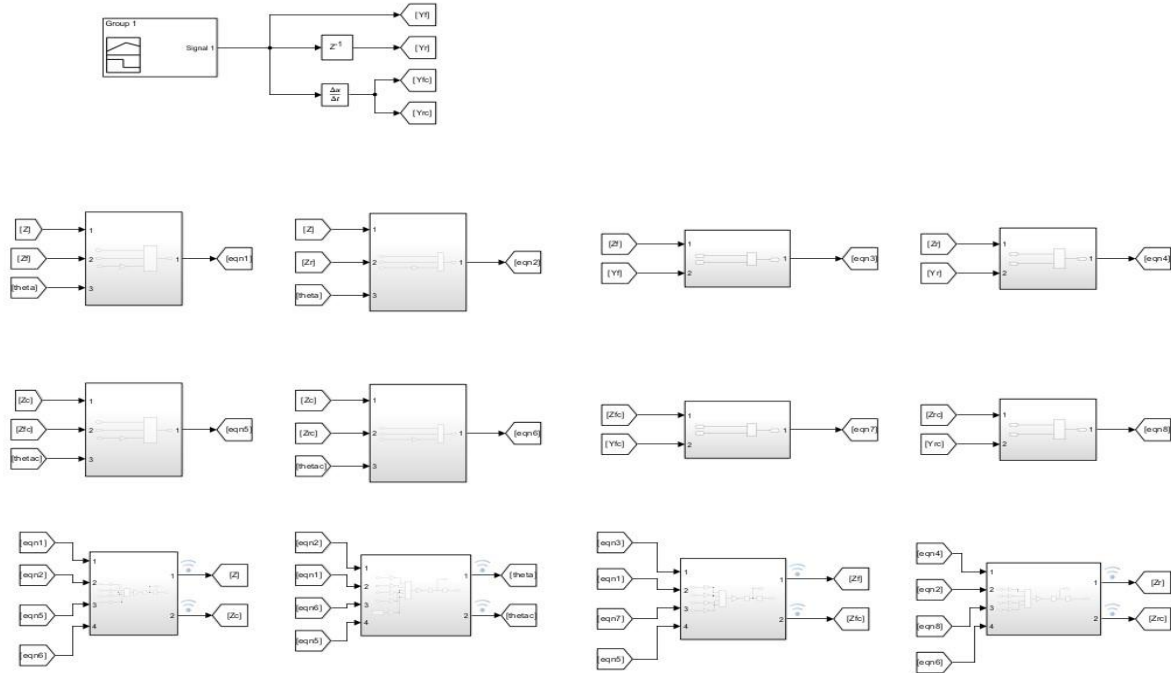


Fig. 6: 4-DOF Simulink Model

From the 4-DOF model equations,

- $-M_s \ddot{z}_s = k_f (z_s - z_f - a\theta) + k_r (z_s - z_r + b\theta) + c_f (\dot{z}_s - \dot{z}_f - a\dot{\theta}) + c_r (\dot{z}_s - \dot{z}_r + b\dot{\theta})$
- $-I\ddot{\theta} = b k_r (z_s - z_r + b\theta) - a k_f (z_s - z_f - a\theta) + b c_r (\dot{z}_s - \dot{z}_r + b\dot{\theta}) - a c_f (\dot{z}_s - \dot{z}_f - a\dot{\theta}) + M_s a_x h$
- $-M_f \ddot{z}_f = k_{tf} (z_f - y_f) - k_f (z_s - z_f - a\theta) + c_{tf} (\dot{z}_f - \dot{y}_f) - c_f (\dot{z}_s - \dot{z}_f - a\dot{\theta})$
- $-M_r \ddot{z}_r = k_{tr} (z_r - y_r) - k_r (z_s - z_r + b\theta) + c_{tr} (\dot{z}_r - \dot{y}_r) - c_r (\dot{z}_s - \dot{z}_r + b\dot{\theta})$

Since there are so many terms in the equations, the model is built in the form of multiple subsystems. These subsystems work together to give us the dynamic response of the vehicle.

For example, from the first equation, all the terms in the brackets, $(z_s - z_f - a\theta)$, $(z_s - z_r + b\theta)$, $(\dot{z}_s - \dot{z}_f - a\dot{\theta})$ and $(\dot{z}_s - \dot{z}_r + b\dot{\theta})$, are grouped into different subsystems. Each subsystem has sum and gain blocks to form the terms in the brackets. This is done for all the four equations. Since most of the terms repeat, we get eight different subsystems. This can be seen in the top eight subsystems of Fig. 6.

Next, to complete each equation, four more subsystems are added to represent each equation. Fig. 7 shows the subsystem representing the second equation as it has the input for longitudinal acceleration and gives the pitch of the vehicle.

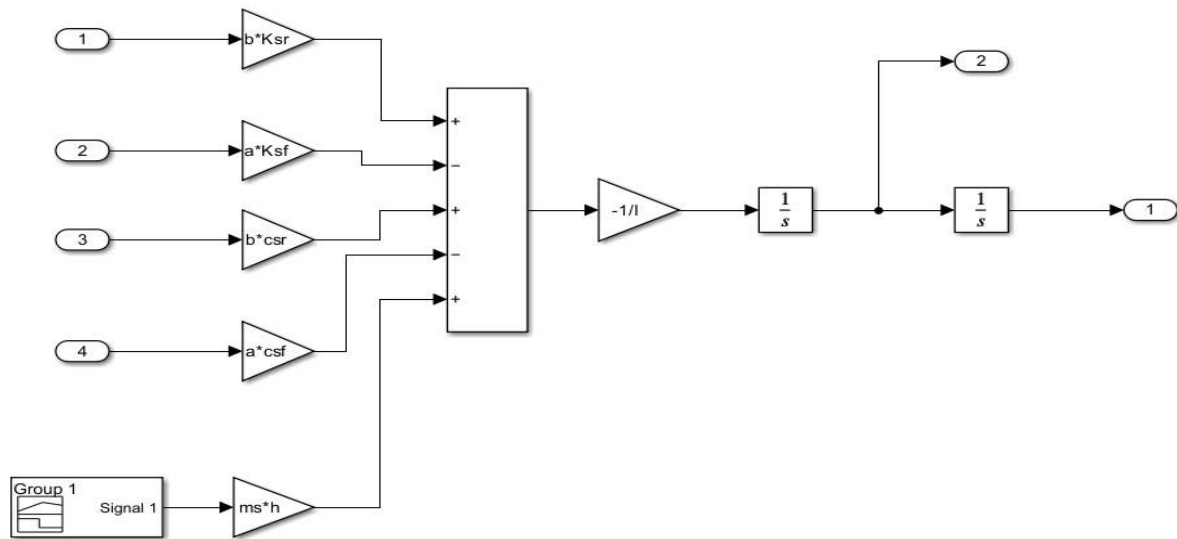


Fig. 7. Subsystem for Equation Giving Pitch in 4-DOF Model

To input the longitudinal acceleration of the vehicle, a signal builder is added along with a gain. This is then added to the sum block. The longitudinal acceleration data acquired from MoTeC is then added to the signal builder, which is discussed in Section 6.

5.3 7-DOF MODEL

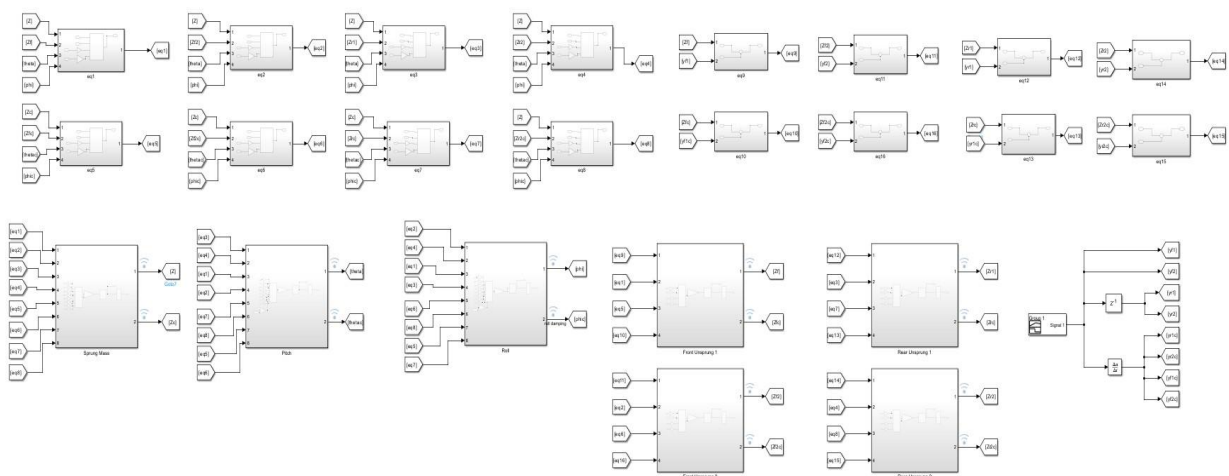


Fig. 8: 7-DOF Simulink Model

Just like the 4-DOF model, multiple subsystems are built for the terms within the equations of the 7-DOF model as well. Each term is added using the From and Goto blocks. After this is done, seven more subsystems are added, one for each equation defining the 7-DOF model.

In this model, the third equation (given in Section 3.3) gives the values related to the roll of the vehicle. This is done by adding a signal builder as well. The data relating to the lateral acceleration of the vehicle is added from MoTeC. The subsystem for this is shown in Fig.9.

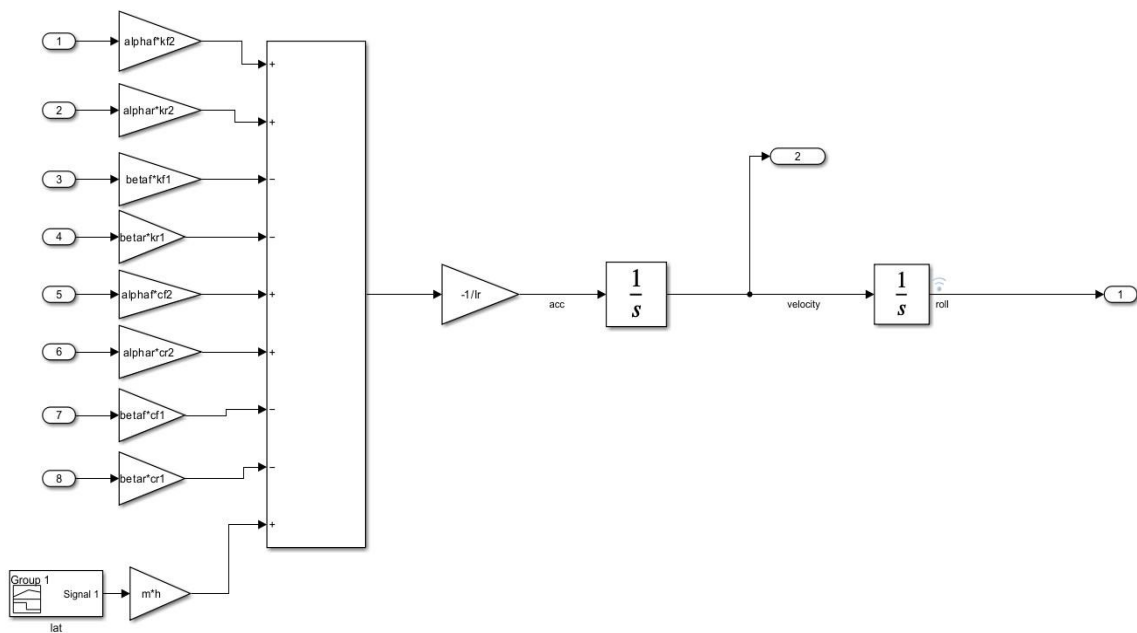


Fig. 9: Subsystem for Equation Giving Roll in 7-DOF Model

6. ACQUIRING DATA FROM MoTeC

The data for the longitudinal and lateral accelerations were produced in the simulator and stored in the form of MoTeC files. These files contain the data from a simulated test of a vehicle performing a standard lane change dynamic benchmarking manoeuvre. This test was repeated three times, each with a different front and rear spring rates as shown in Table 1.

Table 1:

Springs Investigation		Setup 1	Setup 2	Setup 3
Front Spring Rate (N/mm)	K_{sf}	14	18	23
Rear Spring Rate (N/mm)	K_{sr}	20	16	12

The process to acquire the data is explained next. This was done for all the three setups but is explained for just one here.

First, MoTeC is opened and the file containing the longitudinal and lateral accelerations are loaded. Fig. 10 shows the loaded MoTeC screen.



Fig. 10: MoTeC Data

Next, the line was moved to the section containing the data from the lane change manoeuvre. In the bottom part containing the data for the G forces, the manoeuvre section was highlighted and exported. While exporting, the option to export only the 'Visible' section was chosen along with export file type to 'MATLAB File'. This file was then stored in the same folder as the MATLAB code and the Simulink model.

7. MATLAB CODE

Once the model was built in Simulink and the simulator data was acquired from MoTec, the MATLAB code was written. This primarily consisted of the defining all the terms used in the model along with storing the simulator data in the form of matrices. All the specifications of the vehicle are mentioned below in Table 2. The simulator data was loaded and the code was run. Here, it was important to note that the 'time' in the simulator data started upon starting the simulation and the lane change manoeuvre was not done immediately. Hence, the time stamps were adjusted so that the time started when the lane change manoeuvre began.

Table 2: Vehicle Parameters

Vehicle Mass (kg)	MT	514
Front Corner Unsprung Mass (kg)	Musf	22
Rear Corner Unsprung Mass (kg)	Musr	22
Vehicle Sprung Mass (kg)	Ms	426
Centre of Gravity Height (m)	CoG	0.277
Weight Split F:R (%)	-	45:55
Wheelbase (m)	W	2.443
Front Track (m)	Tf	1.62
Rear Track (m)	Tr	1.57
Front Tyre Rolling Radius (m)	rf	0.2667
Rear Tyre Rolling Radius (m)	rr	0.2794
Front Motion Ratio	MRf	1
Rear Motion Ratio	MRr	1
Front Default Spring Rate (N/mm)	Ksf	18
Rear Default Spring Rate (N/mm)	Ksr	16
Front Roll Centre Height (mm)	hf	0.9
Rear Roll Centre Height (mm)	hr	59
Front Slow Bump	N/m/s	1660
Front Fast Bump	N/m/s	511
Front Slow Rebound	N/m/s	1915
Front Fast Rebound	N/m/s	511
Rear Slow Bump	N/m/s	1477
Rear Fast Bump	N/m/s	537
Rear Slow Rebound	N/m/s	1612
Rear Fast Rebound	N/m/s	537

8. VALIDATION

There were no papers found online for the validation of a 7-DOF model completing a standard lane change manoeuvre with all the vehicle specifications mentioned properly. Hence, for validation purposes, the roll angle graph from another paper is looked into to compare it to the nature of results achieved for the Formula Ford Race Car.

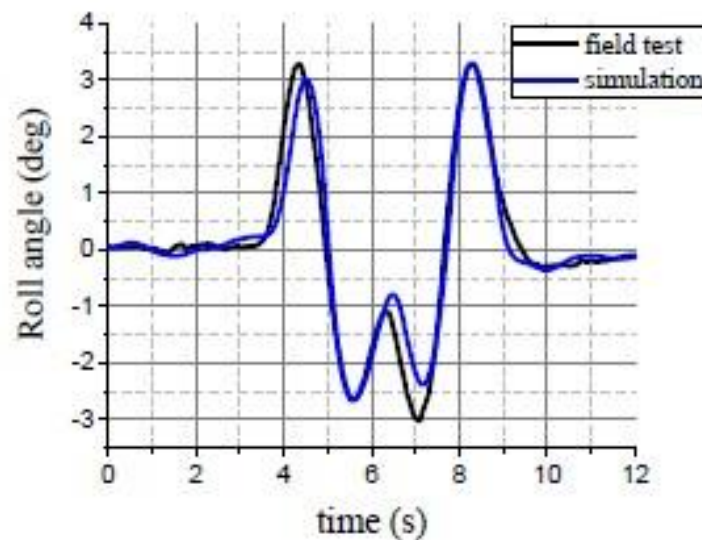


Fig. 11: Roll angle Graph for Validation Comparison (Bindac et al., 2022)

In section 9.3, it can be seen that the graphs obtained for the roll angles for all three setups were similar to Fig. 11.

9. RESULTS

9.1 1-DOF MODEL

Even though the 1-DOF model was built, it is out of scope to obtain results from the given data. The 1-DOF model responds only to undulations on the road surface. The details of the road surface are not available in the given data. Only the lateral and longitudinal accelerations of the body are available through the MoTeC data.

9.2 4-DOF MODEL

Unlike the 1-DOF model, the 4-DOF model has the ability to pitch. As a result, it can react to longitudinal accelerations that the vehicle undergoes. The graphs for the vehicle pitch for the three different spring rates can be seen in Fig. 12.

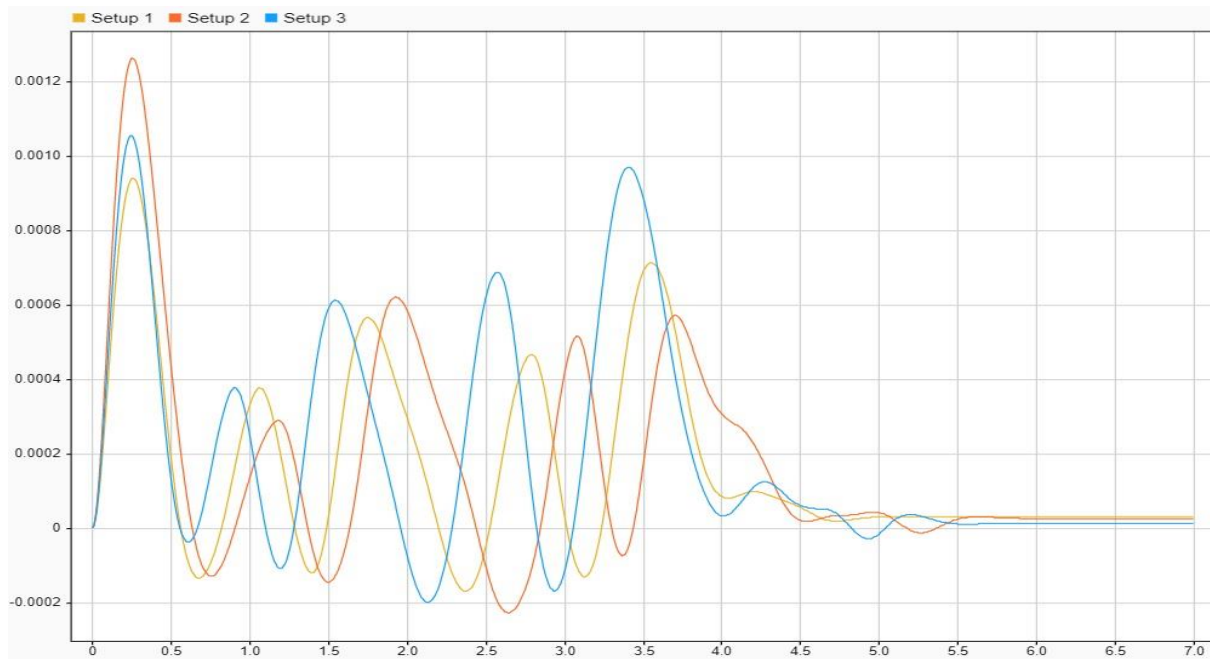


Fig. 12: 4-DOF Graph for Pitch

It can be seen that Setup 1 with the lower front spring rate of 14 N/mm and higher rear spring rate 20 N/mm has relatively lower amplitudes of pitch when compared to Setup 3 which has higher front spring rate of 23 N/mm and lower rear spring rate of 12 N/mm. Setup 2 with medium front and rear spring rates of 18 N/mm and 16 N/mm had the highest pitch amplitude initially but the amplitude in the subsequent peaks were significantly lower when compared to Setup 1 and Setup 3.

9.3 7-DOF MODEL

The 7-DOF model has the ability to roll as well, in addition to its ability to pitch like the 4-DOF model. Hence, it not just reacts to longitudinal accelerations of the vehicle but also to lateral accelerations.

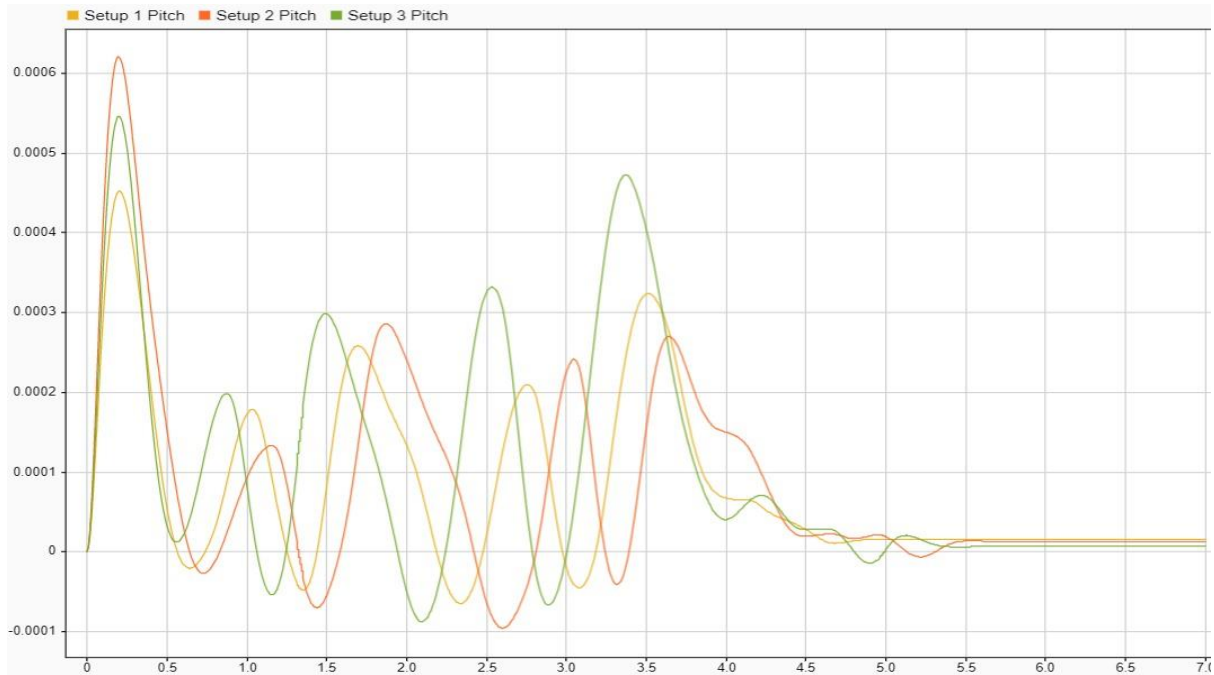


Fig. 13: 7-DOF Graph for Pitch

As we can see in Fig. 13, the 7-DOF graph for pitch looks almost identical to that of the 4-DOF graph for pitch. But the major difference observed here is the amount of pitch the vehicle undergoes.

The amplitudes of the pitch in the graphs for the 7-DOF model was observed to be almost half that of the pitch graphs for 4-DOF model. This could possibly be explained by the fact that the 7-DOF model has got more degrees of freedom, which could lead to the pitching moment being distributed more evenly over the additional unsprung masses. Also, this model is countering the roll of the vehicle, which has a way larger amplitude compared to pitch during a standard lane change manoeuvre, as shown in Fig. 14. As a result, the springs and dampers are already in function and the amount of pitch experienced by the vehicle is lower.

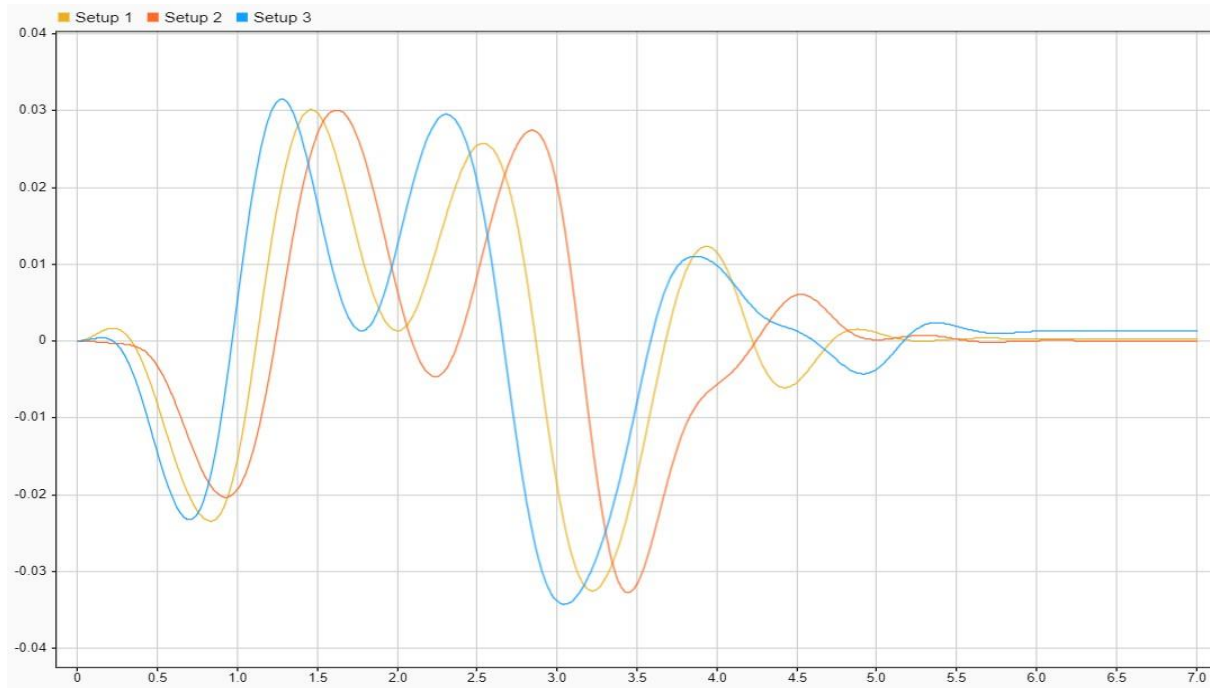


Fig. 14: 7-DOF Graph for Roll

In Fig. 14, we can see that Setup 3 with higher front spring rate of 23 N/mm and lower rear spring rate of 12 N/mm experiences the highest amount of roll, even though marginally, with its highest peaks and troughs. Meanwhile, Setup 2 with medium front and rear spring rates of 18 N/mm and 16 N/mm experiences the lowest amount of roll overall. It is also observed that Setup 2 has one less oscillation after the second lane change manoeuvre is completed. Setup 1 with lower front spring rate of 14 N/mm and higher rear spring rate 20 N/mm experiences similar amount of roll during the first lane change as Setup 3 and less roll during the second lane change manoeuvre, but has more roll at the end of the manoeuvre. Even with the higher roll in the end, Setup 1 takes less time to stabilize compared to Setup 3.

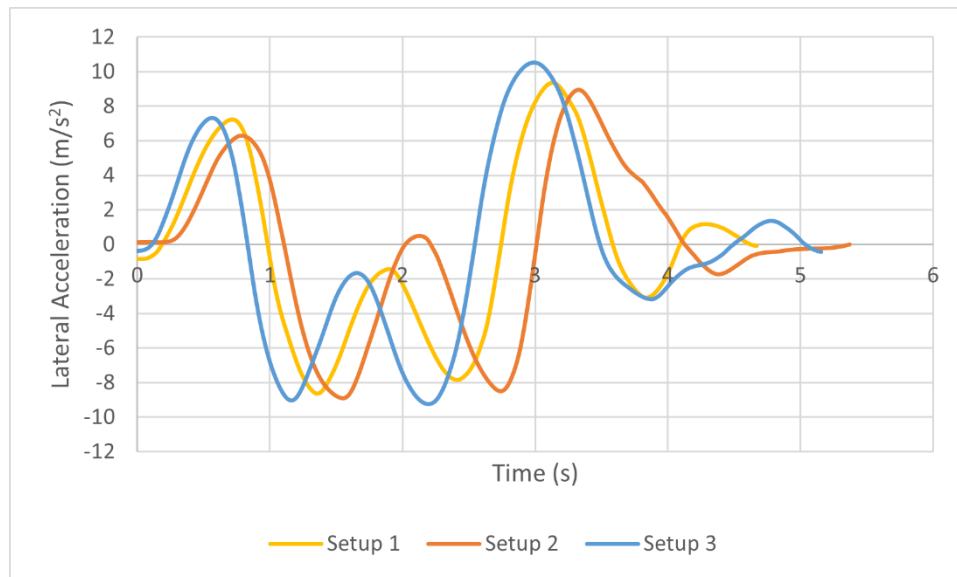


Fig. 15. Lateral Acceleration Simulator Data

Fig. 15 shows the lateral acceleration data that was acquired from the MoTeC file containing simulator data. When we compare the graphs in Fig. 14 and Fig. 15, it is observed that the lateral acceleration graphs are mirror images of the graphs for roll. This is because during roll, the mass of the vehicle is constantly decelerating or is in negative acceleration whereas lateral acceleration is always positive. The change in polarity from positive to negative and vice versa in both the graphs is due to change in direction.

10.CONCLUSION

In this paper, the basics of a spring mass damper system was discussed. This was done in order to understand the transient response of a vehicle when it undergoes certain manoeuvres, focusing on a standard on lane change procedure in this case. For this, different vehicle models with differing degrees of freedom were constructed using MATLAB and Simulink, starting from 1-DOF to 7-DOF. Then, the dynamic response data such as lateral and longitudinal accelerations from a Formula Ford Race Car was used to assess the roll and pitch of the various models when the spring stiffnesses were varied. The data for the Formula Ford Race Car was attained using a simulator and this data was then acquired using MoTeC.

The 1-DOF model is the most primitive of all the models as it only allows vertical movement of the vehicle and the entire vehicle is assumed to be a rigid body. The 4-DOF model is a slight improvement as it considers the front and rear wheels to be two unsprung masses. This allows the vehicle to have a pitching motion. Lastly, the 7-DOF model assumes all the four wheels to

be unsprung masses and as a result, this model allows the vehicle to have a rolling motion in addition to pitching.

For the 4-DOF model, Setup 2 with front and rear spring rates of 18 N/mm and 16 N/mm had a high pitch angle initially but had a lower pitch angle for the rest of the period.

The 7-DOF model had the same pitch characteristics for all the three setups but the amplitudes of the pitch angle were half that of the 4-DOF model. This was assumed to happen because of the increase in the degrees of freedom and the presence of lateral acceleration causing the vehicle to roll as well. For the roll graphs, Setup 2 again had the least amplitudes compared to Setup 1 and Setup 3, in addition of having one less oscillation.

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