



Prifysgol Cymru
Y Drindod Dewi Sant
University of Wales
Trinity Saint David

Transient Response Modelling of a Vehicle

ACEA7003: Advanced Vehicle Dynamic Behavior
MSc Automotive Engineering

Student: Jaimish V. Navadiya (2109787)

Lecturer: Andrew Noble

Date: 29 Jan. 2023



Abstract

This report is about design a 7 Degree of Freedom (DOF) Spring-Mass-Damper (SMD) model of the Formula Ford race car and characterise its transient response behaviour. The vehicle model is constructed using MATLAB and SIMULINK. This report represents the path of how I create 7 DOF model from 1 DOF of simplest Spring-Mass-Damper model. First, I built 1 DOF simple spring mass harmonic motion model with damping and after that I add more degree of freedom like secondary mass linear translation (no heave and pitch) which is known as Quarter car model. After completion of this model, I tried to build 2 DOF model with rotational motion (Heave and Pitch). Heave and pitch model is used for me to understand how to apply roll and pitch in the 7 DOF model. After successfully completion of the 2 DOF linear and rotational model, I built half car model which is 4 DOF vehicle model. In the last, completion of all these models, I built 7 DOF car model. In 7 DOF model, I import the longitudinal and lateral accelerations from simulation of the vehicle performing a standard lane change dynamic benchmarking manoeuvre. I compare MoTech simulation data and my Simulink model result to verify the accuracy of my model. furthermore, I alter the variables like spring rate, damping coefficient to predict the behaviour of the vehicle.

Introduction

Transient response modelling of a vehicle is the process of analysing how a vehicle responds to sudden changes in its operating conditions, such as sudden acceleration or braking, lane change, cornering. Transient response models are used to predict the vehicle's behaviour under these conditions, including its acceleration, velocity, and position. They can also be used to predict the vehicle's suspension, steering, and braking response.

The transient response model is typically based on a set of differential equations that describe the dynamics of the vehicle's motion. These equations consider the vehicle's mass, suspension stiffness, tire properties, and other factors that affect its behaviour. The model can be used to simulate the vehicle's response to different inputs, such as changes in the accelerator pedal position or brake pedal pressure.

The transient response model can be used to optimize the vehicle's performance under different driving conditions, including high-speed and low-speed handling, braking performance, and ride comfort. It can also be used to predict the vehicle's behaviour under extreme conditions, such as emergency braking or sudden steering manoeuvres.

It's important to note that this model is again a simplified model, and it does not consider all the factors that affect a vehicle's behaviour. However, it can still provide useful insights and predictions that can help improve the vehicle's performance.

The modelling of the vehicle is nothing but solution of some complex mathematical equations that might be solved to predict the response of the vehicle. The model can be created by using MATLAB and SIMULINK software where these mathematical equations can be implemented into block diagram to solve these equations. A 7DOF (degree of freedom) spring mass damper model of a Formula Ford race car is a mathematical representation of the dynamics of the car's suspension system. It considers seven different variables that affect the car's behaviour, including the position and velocity of the car's suspension, as well as the forces acting on the car. These variables are used to create equations that describe the car's motion, which can then be used to predict how the car will respond to different driving conditions and to optimize the car's performance.

The 7 degrees of freedom in this model typically include:

The longitudinal motion of the car's centre of gravity- 1

The longitudinal motion of the car's suspension system- 4

The roll angle of the car- 1

The pitch angle of the car- 1

These variables can be used to calculate the car's suspension deflection, roll angle, pitch angle, as well as position of the COG of the car. This information can be used to optimize the car's suspension geometry, spring rates, and damper settings to improve its handling and stability.

Methodology

To design 7 DOF model, first it is important to successfully design of 1 DOF and then gradually import the more DOF into the model. From 1 DOF to 7 DOF model is described in the following method.

1. 1 Degree of Freedom

The simplest model of the Spring-Mass-Damper represent by the equation $M\ddot{x} = K_s(x - y) + C(\dot{x} - \dot{y})$. Where $\ddot{x}$ is the acceleration of the system. $\dot{x}$ is the velocity of the system and x is the displacement of the system. y is disturbance given to the system. that means by some external force is applied to the system. $\dot{y}$ is the velocity of the disturbance. For solving this equation in Simulink first we need to add two different forces, one is spring force and other one is damping force. Then divided by the variable M . But here, the x and y both are dependent on time, that means every smallest interval of the time it will change, and this change is depend on the just before value. So at time $t + \Delta t$, value of x and y are depend on the value x and y at time t . So, for solving equation at time $t + \Delta t$ we need to input of values at time t . The output of velocity and displacement signal is going back for further calculation. 1 DOF SMD model is shown in the figure (1).

Consideration for 1 DOF SMD model,

mass= 514 kg,

spring rate K_s = 18000 N/m,

damping rate= 1660 N/m/s.

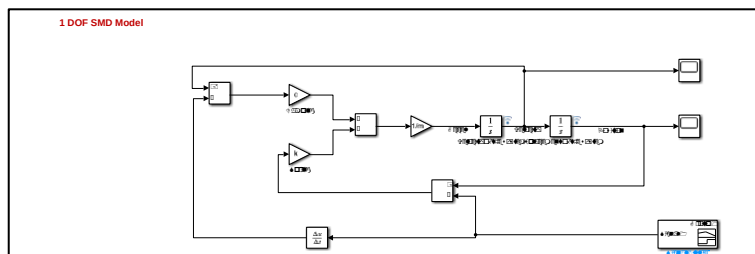


Figure 1: Simulink 1 DOF SMD model

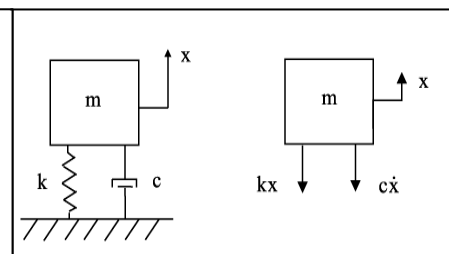


Figure 2: Freebody diagram of 1 DOF SMD model

Now if we give 10 cm disturbance to the system by signal building block. The response of the system is shown in figure (2). This graph is comparison of 2 different condition. In one, there is no any disturbance ($y=0$) then there is no any response in the system. in second, I gave 10 cm disturbance for 1 second ($y=10$) then mass is vibrating and then is get back to the original position. This model result is satisfied. So after the successfully completion of 1 DOF SMD model, let us add one unsprung mass between two separate springs and dampers as shown in figure (3).

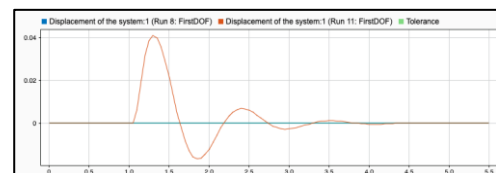


Figure 3: Response of 1 DOF SMD model

2. 2 Degree of Freedom (Quarter Car Model)

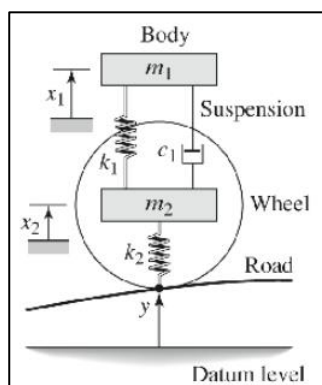


Figure 4: FBD of 2 DOF SMD

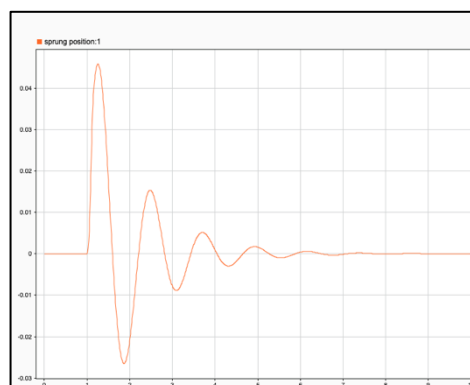


Figure 6: Response of 2 DOF SMD model

2 DOF SMD model is shown in the figure (4) and (5) and (6). Variables for this system.

m_1 = 512 kg,

m_2 =29 kg,

k_1 = 18 N/mm,

k_2 = 50 N/mm,

c_1 = 1660 N/m/s.

from signal builder, 10 cm of disturbance has been given to the system. the maximum displace of the sprung mass is 4 cm and it is gaining its original position after 4 seconds. In

designing of this model, it helps to understand *FROM* and *GOTO* block.

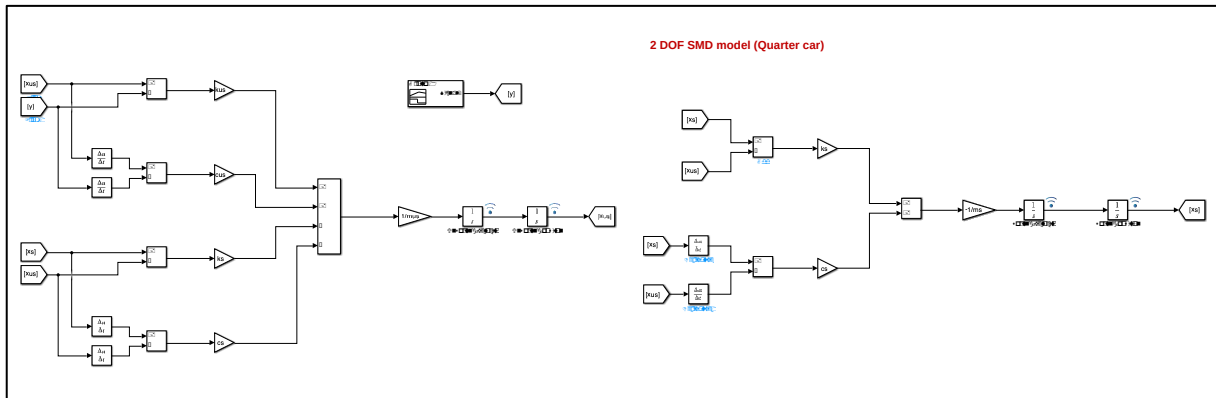


Figure 5: Simulink 2 DOF SMD model (Quarter car model)

3. 2 Degree of Freedom (Heave & Pitch)

From 2 DOF heave and pitch model, it helps to understand torque about centre of gravity, which will further help in 7 DOF to calculate roll and pitch angle. Simulink model for heave and pitch is shown in the figure (7).

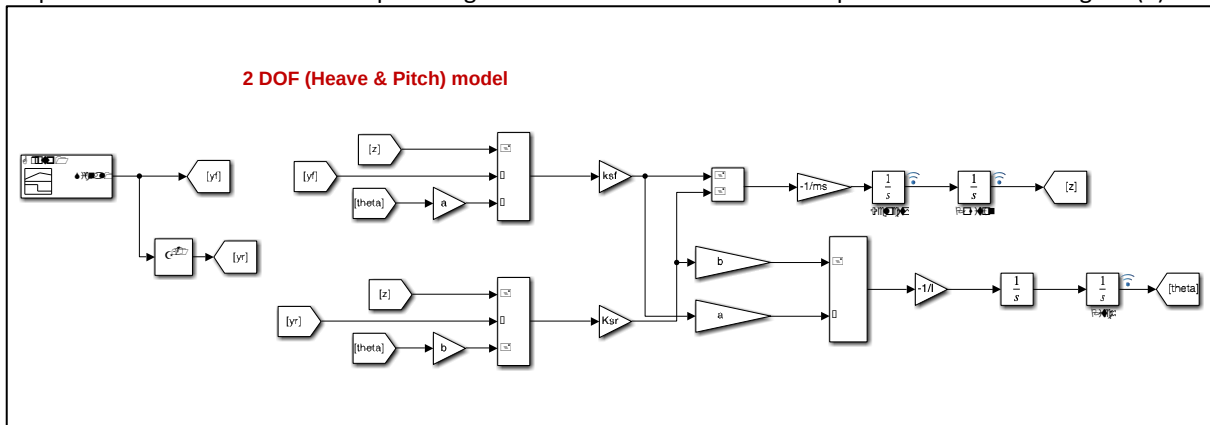


Figure 7: Simulink 2 DOF SMD model (Heave and Pitch model)

The variables for 2 DOF heave and pitch model, there are 2 different simulation results. Input variables for both results are different. For Run 47 (blue), moment of inertia about pitch axis (longitudinal axis) is taken 1000 kgm^2 and for the Run 48 (red), moment of inertia is taken as 500 kgm^2 .

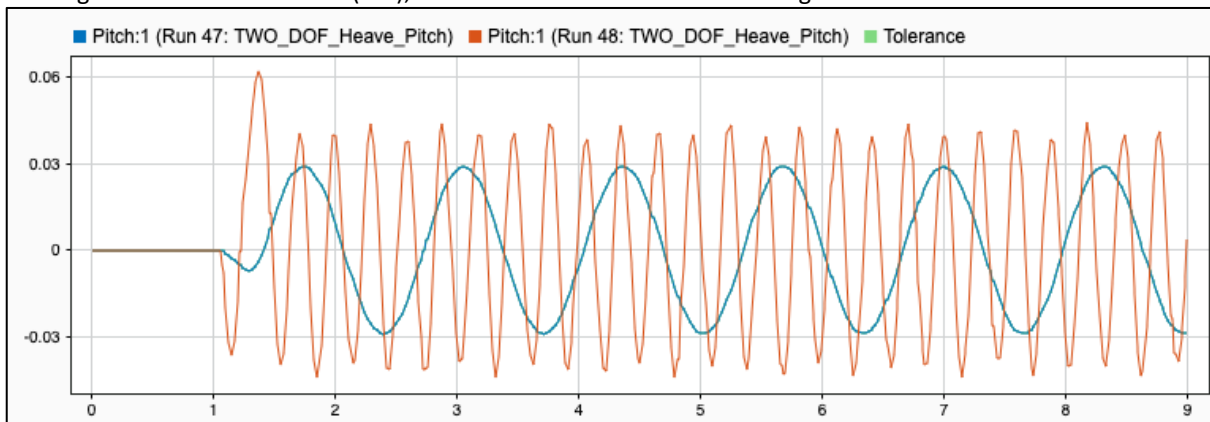


Figure 8: Pitch response of 2 DOF SMD

variables for this simulation are, $m_s = 100$ kg, $K_{sf} = 50$ N/mm, $K_{sr} = 62$ N/mm, $a = 1.6$ m, $b = 1.2$ m and I (Run 47) = 1000 kgm^2 and I (Run 48) = 500 kgm^2 . If increase the moment of inertia, the frequency of pitch is decrease. If decrease MI half, the frequency of the pitch will decrease by 4 times. Here, pitch angle does not decrease, or system is not going to its original position after giving 10 mm disturbing to the system. Even, the system is continuously vibrating. This is because the input variables like mass, spring rate are not match with the real model.

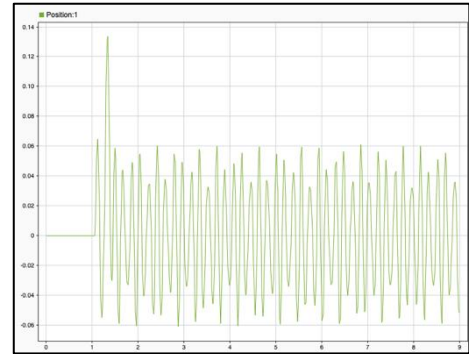


Figure 9: Position response of 2 DOF SMD

4. 4 Degree of Freedom (Half Car Model)

After successfully modelling of 1 and 2 DOF SMD model, I made 4 DOF (half car) model. 4 DOF model is generally front view model of the car. 4 degrees in this model are roll angle, sprung mass position, and two front unsprung mass position. Freebody diagram for 4 DOF half car model is shown in the figure (10). Undamped system equation without pitch torque Simulink model is shown in the figure (11). The input parameters for this system are,

Figure 10: Freebody diagram of 4 DOF SMD

```
clc
clear
ms=380;
```

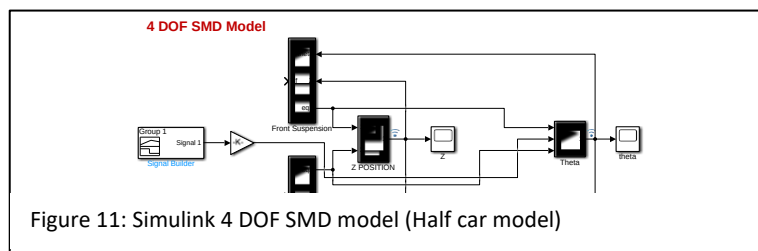


Figure 11: Simulink 4 DOF SMD model (Half car model)

```
Ksf=55000;
Ksr=62000;
Ktf=280000;
Ktr=320000;
a=1.6;
b=1.2;
mf=26;
```

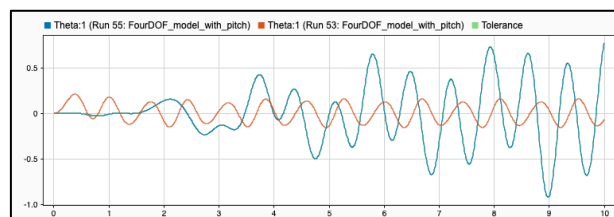
```
mr=29;
l=400;
```

```
%% load Lat accel data
load Setup1_Slalom.mat,G_Force_Lat,G_Force_Long
load Setup1_Slalom.mat,Suspension_Pos_FR,Suspension_Pos_RR;
aa=Suspension_Pos_FR.Time;
xx= diff(aa);
deltaT=xx(1);
T=deltaT*length(xx)
time=0:deltaT:T;
ax= G_Force_Lat.Value;
ay= G_Force_Long.Value;
```

```
h=0.9;
```

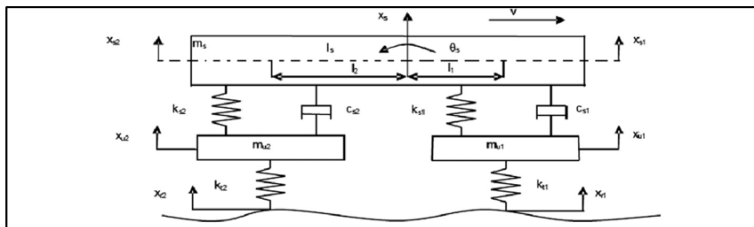
By considering this input variables for half car model the pitch response can be seen in the figure (12). I applied 2 lane driving maneuverer longitudinal acceleration as well as slalom driving manoeuvre longitudinal acceleration to verify my 4 DOF model. here, as we can see in response graph, 2 lane change (blue) pitch angle is gradually increase with respect to time. While in slalom (red) driving maneuverer, pitch angle is changing by sinusoidal fashion.

Figure 12: Pitch response of 4 DOF SMD



5. 4 Degree of Freedom (Half Car Model)

Finally, after verifying all the previous models, I design 7 DOF SMD model in simulink. Input design parameters for Formula Ford Race Car is given in the table 1.



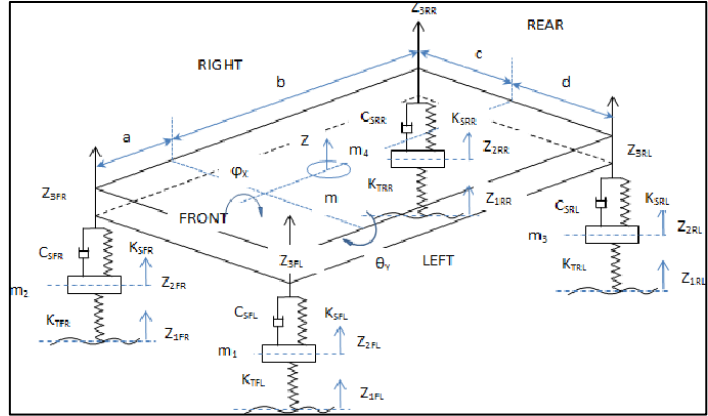
Vehicle Mass (kg)	MT	514
Front Corner Unsprung Mass (kg)	M _{usf}	22
Rear Corner Unsprung Mass (kg)	M _{usr}	22
Vehicle Sprung Mass (kg)	M _s	426
Centre of Gravity Height (m)	CoG	0.277
Weight Split F:R (%)	-	45:55
Wheelbase (m)	W	2.443
Front Track (m)	T _f	1.62
Rear Track (m)	T _r	1.57
Front Tyre Rolling Radius (m)	r _f	0.2667
Rear Tyre Rolling Radius (m)	r _r	0.2794
Front Motion Ratio	MR _f	1
Rear Motion Ratio	MR _r	1
Front Default Spring Rate (N/mm)	K _{sf}	18
Rear Default Spring Rate (N/mm)	K _{sr}	16
Front Roll Centre Height (mm)	h _f	0.9
Rear Roll Centre Height (mm)	h _r	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

For designing of the formula ford 7 dof model, some assumptions have been considered due to lake of official data availability.

- (1) Moment of inertia about Roll axis (I_{xx}) = 575.3 kgm²
- (2) Moment of inertia about Pitch axis (I_{yy}) = 120.1 kgm²
- (3) Here, location of roll centre height and pitch centre height both are consider same point.
- (4) $h=h_2=0.0221$ m from centre of gravity (COG).
- (5) Slow bump and rebound only considered in the 2-lane change driving maneuverer. In slalom driving maneuverer, only fast bump and rebound considered.

Mathematical Model of Formula Ford Race Car

Figure 13: 7 DOF SMD vehicle model



Sprung mass position equation:

$$m_s \ddot{Z} = K_{f1}(Z - \alpha_f \theta - b\phi) + K_{f2}(Z - \alpha_f \theta + a\phi) + K_{r1}(Z + \beta_r \theta - b\phi) + K_{r2}(Z - \beta_r \theta + a\phi) + C_{f1}(\dot{Z} - \alpha_f \dot{\theta} - b\dot{\phi}) + C_{f2}(\dot{Z} - \alpha_f \dot{\theta} + a\dot{\phi}) + C_{r1}(\dot{Z} + \beta_r \dot{\theta} - b\dot{\phi}) + C_{r2}(\dot{Z} - \beta_r \dot{\theta} + a\dot{\phi})$$

Pitch angle equation:

$$I_{xx} \ddot{\theta} = \beta_r K_{r1}(Z + \beta_r \theta - b\phi) + \beta_r C_{r1}(\dot{Z} + \beta_r \dot{\theta} - b\dot{\phi}) + \beta_r K_{r2}(Z - \beta_r \theta + a\phi) + \beta_r C_{r2}(\dot{Z} - \beta_r \dot{\theta} + a\dot{\phi}) - \alpha_f K_{f1}(Z - \alpha_f \theta - b\phi) - \alpha_f K_{f2}(Z - \alpha_f \theta + a\phi) - \alpha_f C_{f1}(\dot{Z} - \alpha_f \dot{\theta} - b\dot{\phi}) - \alpha_f C_{f2}(\dot{Z} - \alpha_f \dot{\theta} + a\dot{\phi}) + m_s h a_x$$

Table 1: Formula Ford Race Car parameters

Roll angle equation:

$$I_{yy} \ddot{\phi} = \alpha_f K_{f2}(Z - \alpha_f \theta + a\phi) + \alpha_f C_{f2}(\dot{Z} - \alpha_f \dot{\theta} + a\dot{\phi}) + \beta_r K_{r2}(Z - \beta_r \theta + a\phi) + \beta_r C_{r2}(\dot{Z} - \beta_r \dot{\theta} + a\dot{\phi}) - \alpha_f K_{f1}(Z - \alpha_f \theta - b\phi) - \alpha_f C_{f1}(\dot{Z} - \alpha_f \dot{\theta} - b\dot{\phi}) - \beta_r K_{r1}(Z + \beta_r \theta - b\phi) - \beta_r C_{r1}(\dot{Z} + \beta_r \dot{\theta} - b\dot{\phi}) + m_s h a_y$$

Front left suspension position:

$$m_{f1} \ddot{Z}_{f1} = K_{tf1}(Z_{f1} - y_{f1}) - K_{f1}(Z - \alpha_f \theta - b\phi) - C_{f1}(\dot{Z} - \alpha_f \dot{\theta} - b\dot{\phi}) + C_{tf1}(\dot{Z}_{f1} - \dot{y}_{f1})$$

Front right suspension position:

$$m_{f2} \ddot{Z}_{f2} = K_{tf2}(Z_{f2} - y_{f2}) - K_{f2}(Z - \alpha_f \theta + a\phi) - C_{f2}(\dot{Z} - \alpha_f \dot{\theta} + a\dot{\phi}) + C_{tf2}(\dot{Z}_{f2} - \dot{y}_{f2})$$

Rear left suspension position:

$$m_{r1} \ddot{Z}_{r1} = K_{tr1}(Z_{r1} - y_{r1}) - K_{r1}(Z + \beta_r \theta - b\phi) - C_{r1}(\dot{Z} + \beta_r \dot{\theta} - b\dot{\phi}) + C_{tr1}(\dot{Z}_{r1} - \dot{y}_{r1})$$

Rear right suspension position:

$$m_{r2} \ddot{Z}_{r2} = K_{tr2}(Z_{r2} - y_{r2}) - K_{r2}(Z - \beta_r \theta + a\phi) - C_{r2}(\dot{Z} - \beta_r \dot{\theta} + a\dot{\phi}) + C_{tr2}(\dot{Z}_{r2} - \dot{y}_{r2})$$

Matlab coding for 7 DOF formula ford car

```
clc  
clear
```

```
Mt=514;    % total mass - kg  
mf1=22;    % front unsprung mass - kg  
mf2=mf1;  
mr1=22;    % rear unsprung mass - kg  
mr2=mr1;  
ms=426;    % sprung mass - kg  
           % weight split F:R % = 45:55  
Hcg=0.277; % height of center of gravity - m  
W=2.443;   % wheelbase - m  
alpha=W*0.55;  
beta=W*0.45;  
Tf=1.62;   % front track - m  
Tr=1.57;   % rear track - m  
af=Tf*0.55;  
bf=af;  
ar=Tr-af;  
br=ar;  
rf=0.2667; % front tyre rolling radius - m  
rr=0.2794; % rear tyre rolling radius - m  
MRf=1;     % front motion ratio  
MRr=1;     % rear motion ratio
```

```
%% spring rate N/m
```

%SETUP 1

```
Kf1=14000; % front default spring rate - N/m  
Kf2=Kf1;  
Kr1=20000; % rear default spring rate - N/m  
Kr2=Kr1;
```

%SETUP 2

```
% Kf1=18000; % front default spring rate  
% Kf2=Kf1;  
% Kr1=16000; % rear default spring rate  
% Kr2=Kr1;
```

%SETUP 3

```
% Kf1=23000; % front default spring rate  
% Kf2=Kf1;  
% Kr1=12000; % rear default spring rate  
% Kr2=Kr1;
```

```
%% TIRE spring rate
```

```
Ktf1=158000;  
Ktf2=Ktf1;  
Ktr1=Ktf1;  
Ktr2=Ktf1;
```

Tyres spring rates value directly taken from the team SMD lecture notes.

```
%% Roll center height
```

```
hf=0.9/1000; % front  
hr=59/1000; % rear
```

```
%% Fast bump/rebound damping rate
```

```
% Cf1=511; % Front fast bump=rebound  
% Cf2=Cf1;  
%  
% Cr1=537; % Rear fast bump=rebound  
% Cr2=Cr1;
```

There are 3 different springs which are used for simulation and comparison its response in the roll angle and pitch angle as well as sprung mass heave.

%% SLOW bump/rebound damping rate

```
% Cf1=1660; % Front slow bump
% Cf2=Cf1;
%
% Cr1=1477; % Rear slow bump
% Cr2=Cr1;
%
% Cf1=1915; % Front slow rebound
% Cf2=Cf1;
%
% Cr1=1612; % Rear slow rebound
% Cr2=Cf1;
```

```
load Setup1_Slalom.mat,Suspension_Pos_FR,Suspension_Pos_RR;
aa=Suspension_Pos_FR.Time;
xx= diff(aa);
deltaT=xx(1);
T=deltaT*length(xx)
time=0:deltaT:T;
```

```
FR=Suspension_Pos_FR.Value;
RR=Suspension_Pos_RR.Value;
FL=Suspension_Pos_FL.Value;
RL=Suspension_Pos_RL.Value;
```

```
%% SLOW front bump/rebound
for ii=1:length(FR)-1
    if FR(:,ii+1)>FR(:,ii) %rebound
        Cf1= 1915
    else FR(:,ii+1)<FR(:,ii) %bump
        Cf1= 1660
    end
end
```

```
%% SLOW rear bump/rebound
for ii=1:length(RR)-1
    if RR(:,ii+1)>RR(:,ii) %rebound
        Cr1=1612
    else RR(:,ii+1)<RR(:,ii) %bump
        Cr1= 1477
    end
end
%%
Cf2=Cf1;
Cr2=Cr1;
```

%% Tires damping rate

```
Ctf1=16;
Ctf2=Ctf1;    Tyres damping rates value directly taken from the team SMD lecture notes.
Ctr1=Ctf1;
Ctr2=Ctr1;
```

%% moment of inertia kg.m^2

```
lxx=575.3;
lyy=120.1;
```

Due to lake of data and experiment limitation, the MOI values directly taken from the SMD lecture notes.

%% Roll centre height @ CG point

```
h1=Hcg-hf;
h2=Hcg-hr;
i=hr-hf;
```

Slow bump and rebound define form the MoTec simulation data. In 2-Lane changing driving maneuverer, if position at time= T +deltaT is greater than position at time T. it is considered as rebound.

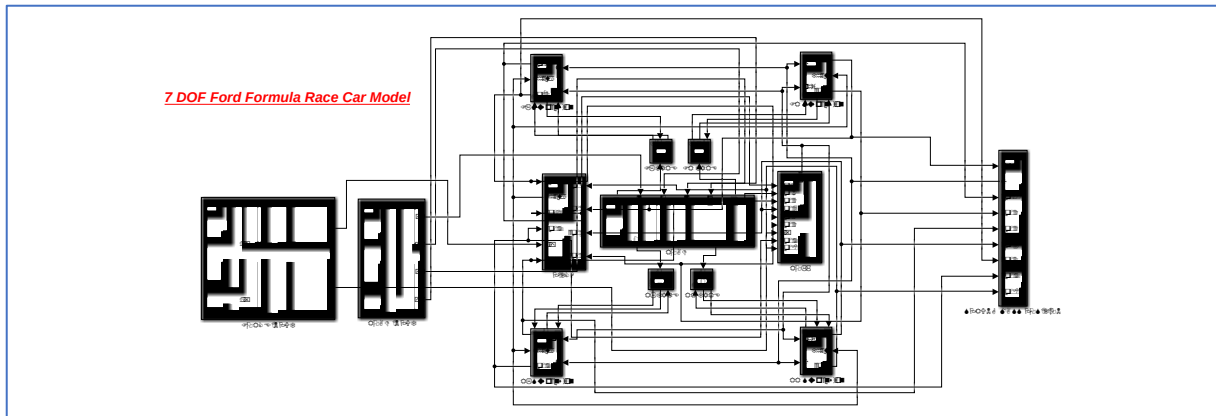


Figure 14: SIMULINK 7 DOF SMD vehicle model

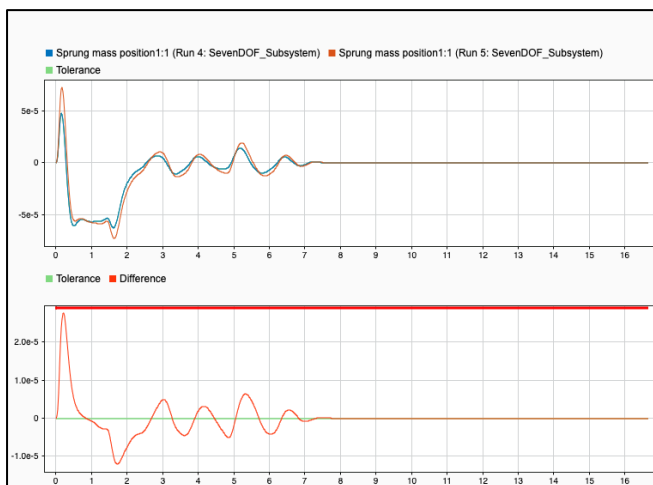
$$\tan X = i/W;$$

$$h = hf + (\tan X * af);$$

```
load ('Setup1_Slalom.mat','G_Force_Lat','G_Force_Long');
```

```
%% define time
aa=G_Force_Lat.Time;
xx=diff(aa);
deltaT=xx(1);
T=deltaT*length(xx);
time=0:deltaT:T;
```

```
%% Lateral acceleration
```



```
ax= G_Force_Lat.Value;
```

```
%% Longitudanal acceleration
```

```
ay= G_Force_Long.Value;
```

```
%% RIDE HEIGHT =Road input
```

```
load Setup1_Slalom.mat Ride_Height_FR
Ride_Height_FL Ride_Height_RR Ride_Height_RL
Hfr=Ride_Height_FR.Value;
Hfl=Ride_Height_FL.Value;
Hrr=Ride_Height_RR.Value;
Hrl=Ride_Height_RL.Value;
```

Since half car model gives 4 dof, I need to model 7 dof model for more accurately analyse the vehicle response. Here all the suspension and wheels are considered independent form each other. This give benefit to incorporate with roll motion of the car.

All independent variables such as tyre, suspension, sprung mass displacement, roll motion and pitch motion model are seen in the appendix.

Validity of Slow bump and rebound

For validation of slow bump and rebound, first I sun the model on same front slow bump value, then I run the model on the condition that is already explained in the matlab coding.

As seen in the figure 15, the difference between sprung mass position in same simulation parameter with only applying slow bump/rebound condition. Blue graph is response of the first condition and the red graph is

response of the system for second condition (in case of applying slow bump/rebound). The difference between these two responses are very small but still there is satisfied that the model is running perfectly.

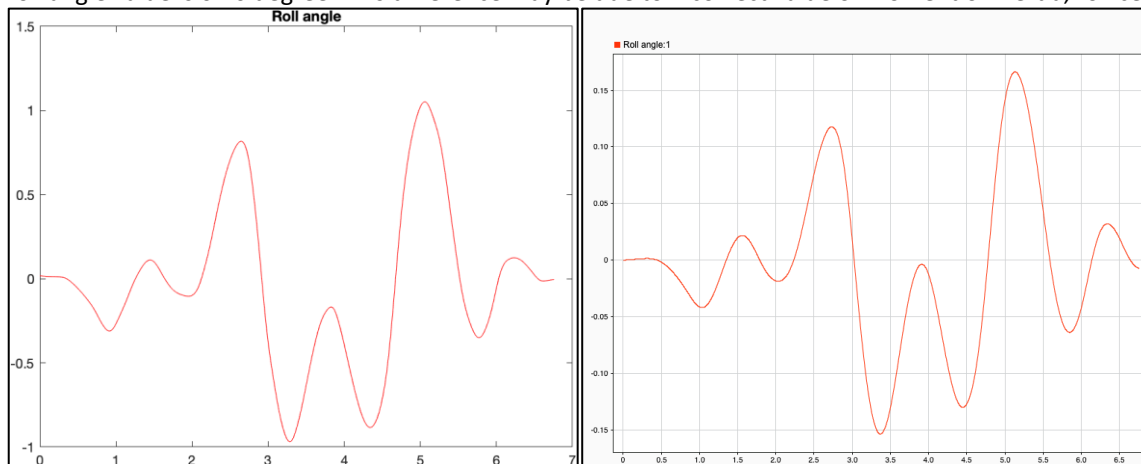
Setup1 2Lane change driving response comparison

Roll Angle:

Figure 16: MoTec Roll angle

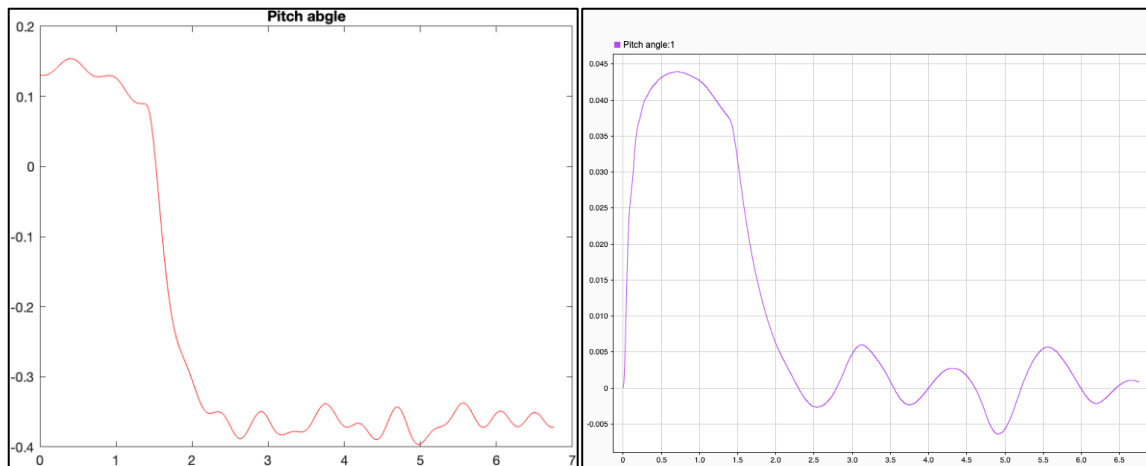
Figure 17: Simulink model Roll angle

As can be seen in the figure 16 (Motec) and 17(Simulink model), roll angle of both the graph is looking like in same fashion. The only difference between them is roll angle values. In the simulation, the max roll angle value is around 1 degree, while in my model, I found the max roll angle value is 0.16 degree. This difference may be due to incorrect value of moment of inertia, roll centre.



Pitch Angle:

Figure 15: Validity of slow bump/rebound



Simulation pitch angle and model pitch angle have some difference in trending. A Simulink graph starting at around 0.13 degree while model graph is starting from 0 at time 0 sec. this difference is supposed from the initial conditions of the vehicle. In the Simulink model I did not consider any initial conditions like at time 0 sec of simulation, the roll position, pitch position, sprung mass position, etc. I consider all these initial conditions from 0. Even in pitch angle, the max pitch angle value from simulation is 0.13 degree while in my model the max pitch angle is seen very small around 0.05 degree.

Suspension position:

Damper velocity:

Figure 18: MoTec Pitch angle

Figure 19: Simulink model Pitch angle

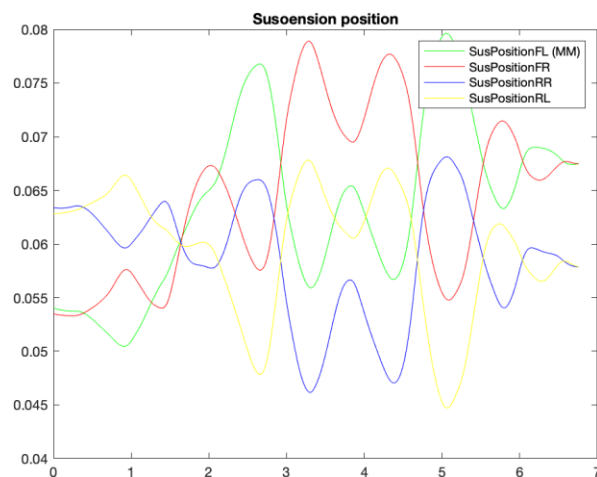


Figure 20: Motec suspension position

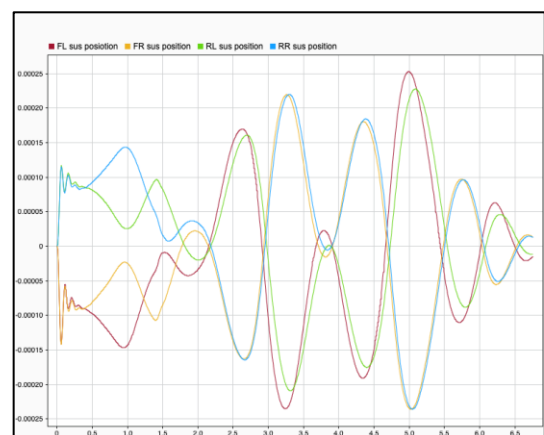
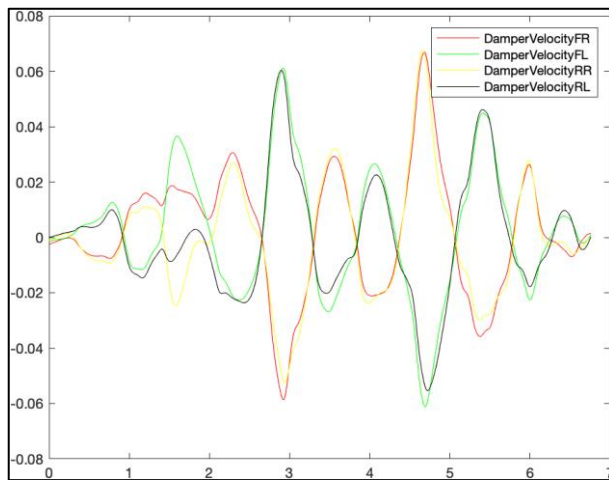


Figure 21: Simulink model suspension position



Setup2 2Lane change driving response comparison

Figure 22: Simulink model damper velocity

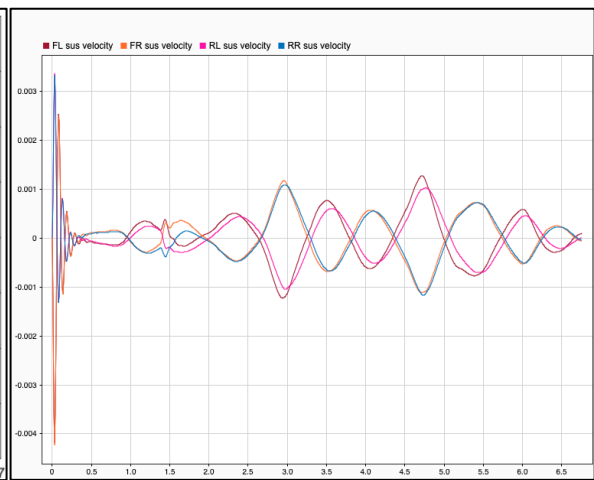


Figure 23: Simulink model damper velocity

Roll Angle:

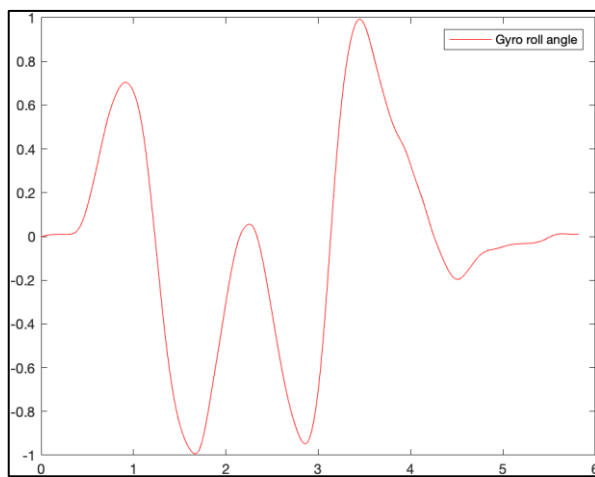


Figure 24: MoTec Roll angle

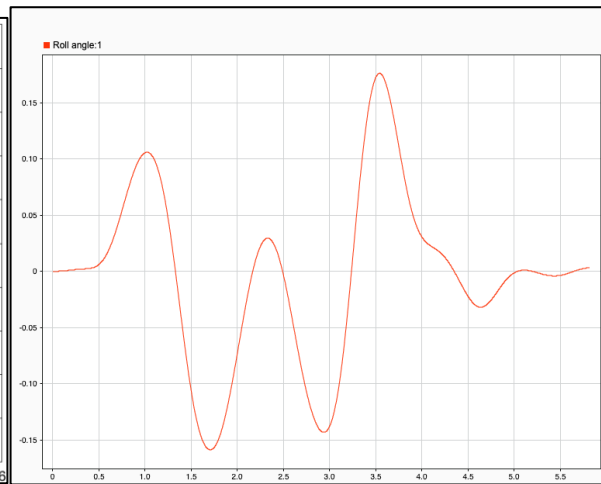


Figure 25: Simulink model Roll angle

Pitch Angle

Suspension position

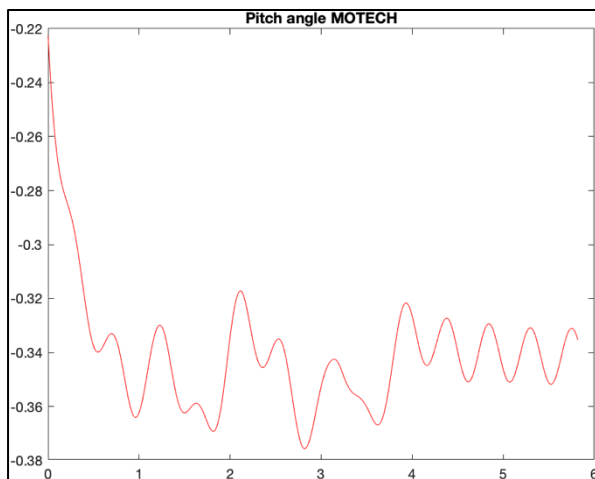


Figure 26: MoTec Pitch angle

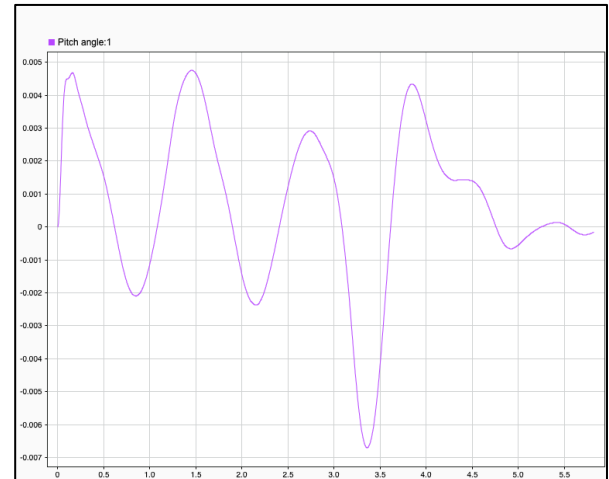


Figure 27: Simulink model Pitch angle

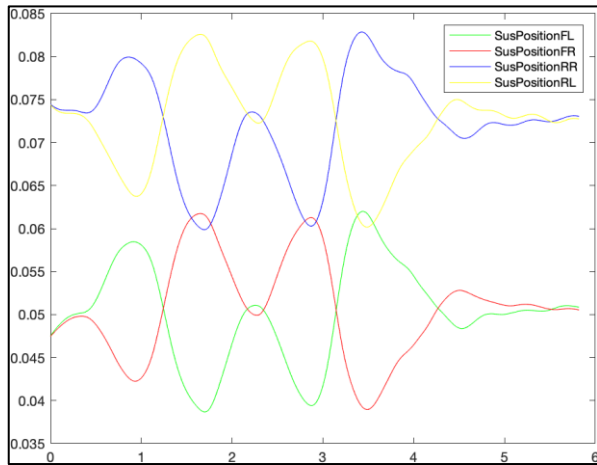


Figure 28: MoTec Suspension position

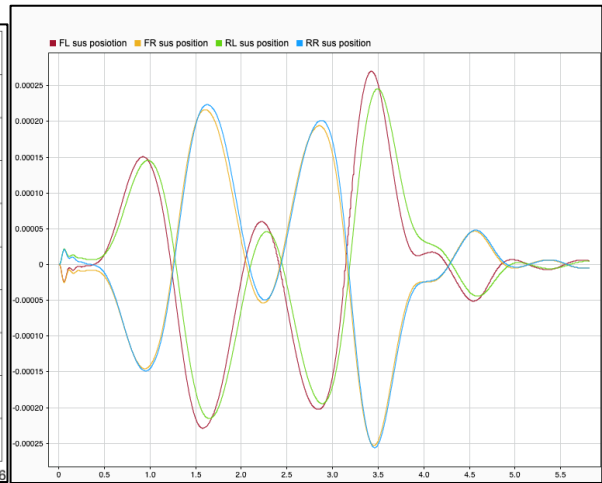


Figure 29: Simulink model suspension position

Damper velocity

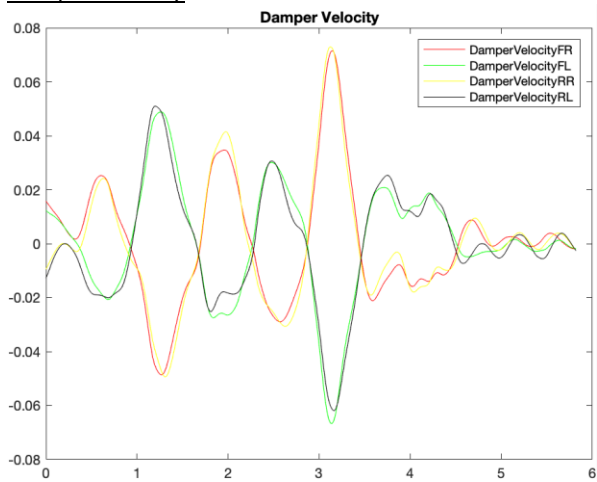


Figure 30: MoTec damper velocity

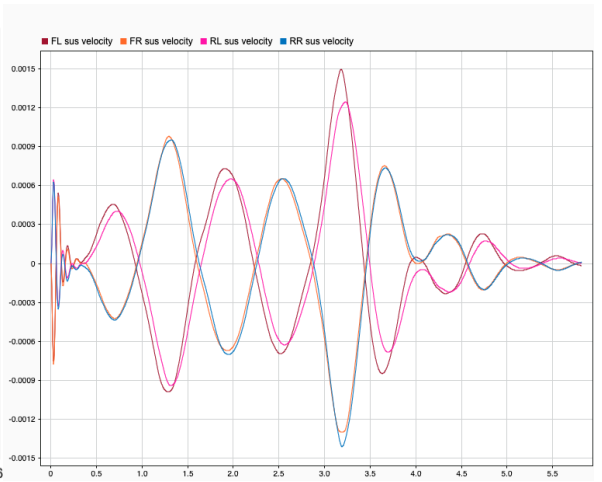


Figure 31: Simulink model damper velocity

Setup3 2Lane change driving response comparison

Roll angle

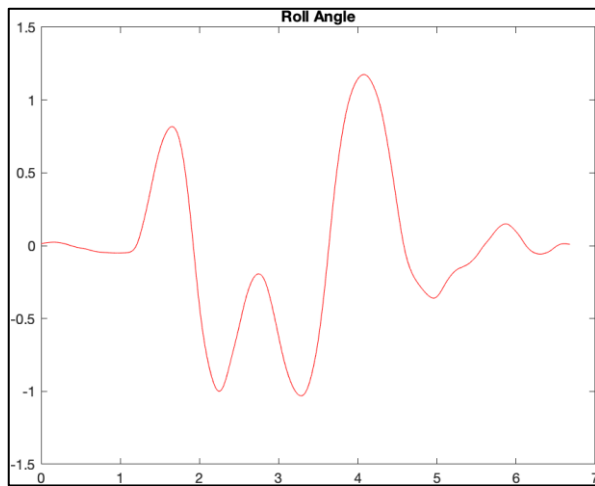


Figure 32: MoTec roll angle

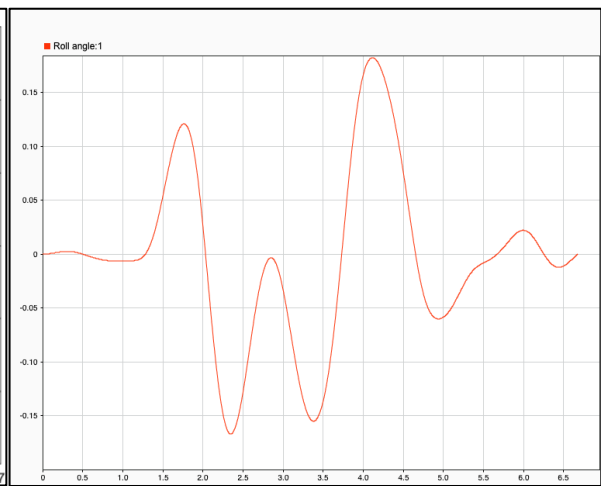


Figure 33: Simulink model roll angle

Pitch angle

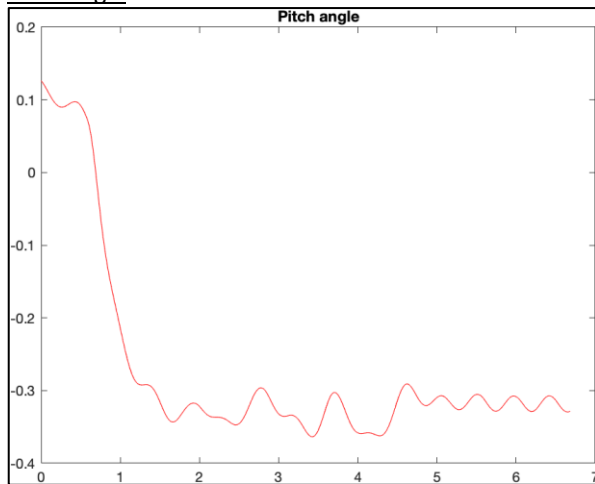


Figure 34: MoTec Pitch angle

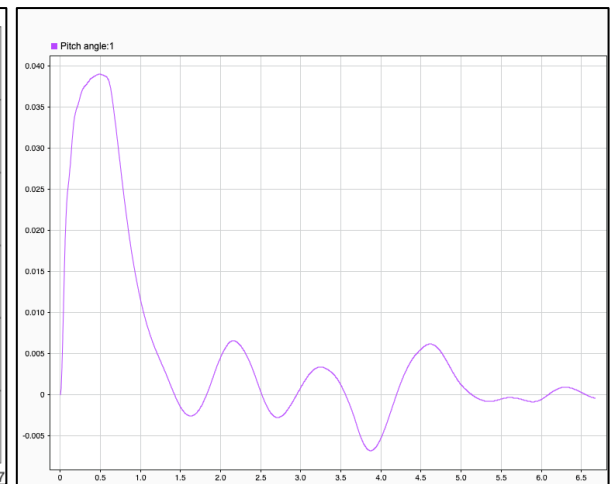


Figure 35: Simulink model Pitch angle

Suspension position

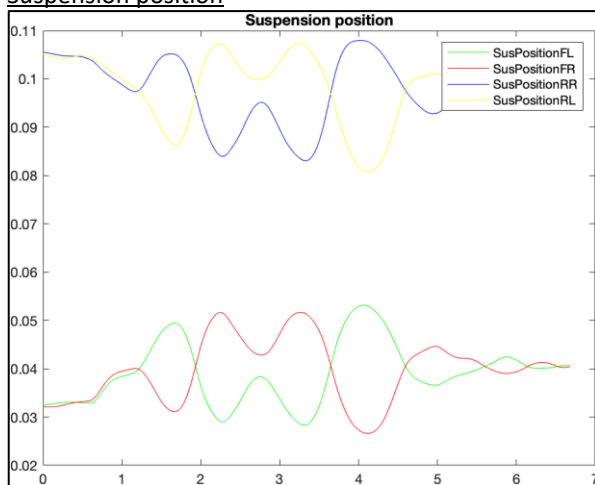


Figure 36: MoTec suspension position

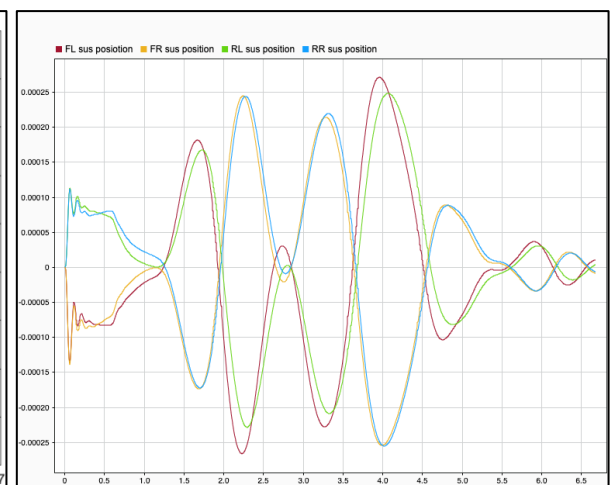


Figure 37: Simulink model suspension position

Setup1 Slalom change driving response comparison

Roll Angle

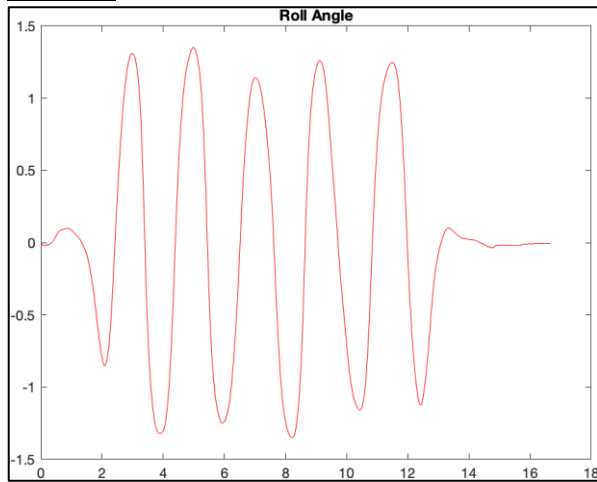


Figure 38: MoTec roll angle

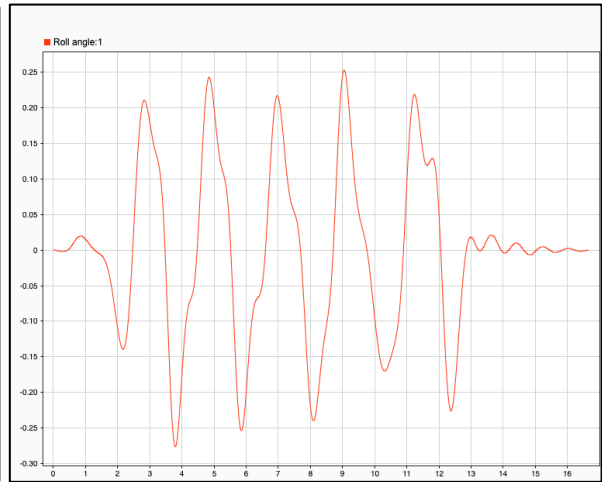


Figure 39: Simulink model roll angle

Pitch Angle

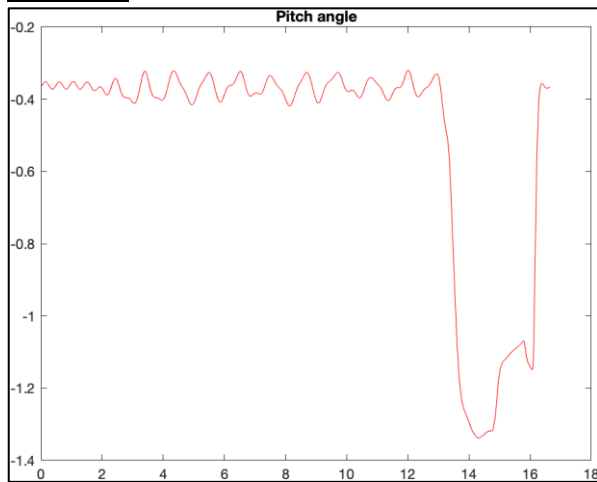


Figure 40: MoTec Pitch angle

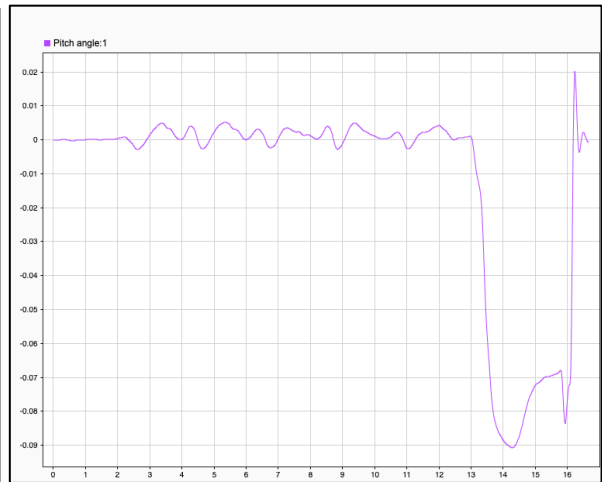


Figure 41: Simulink model Pitch angle

Setup2 Slalom change driving response comparison

Roll angle

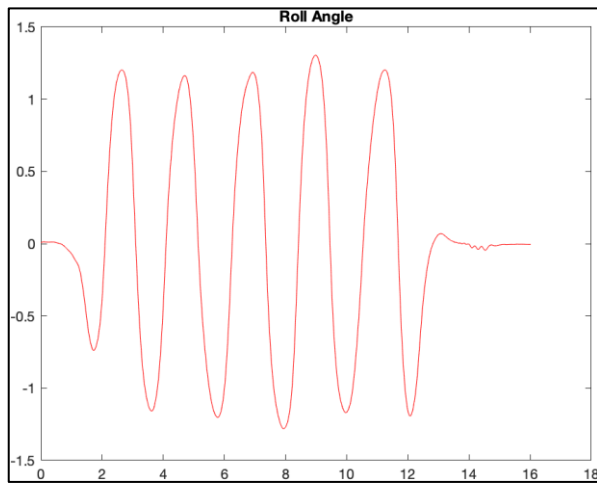


Figure 42: MoTec roll angle

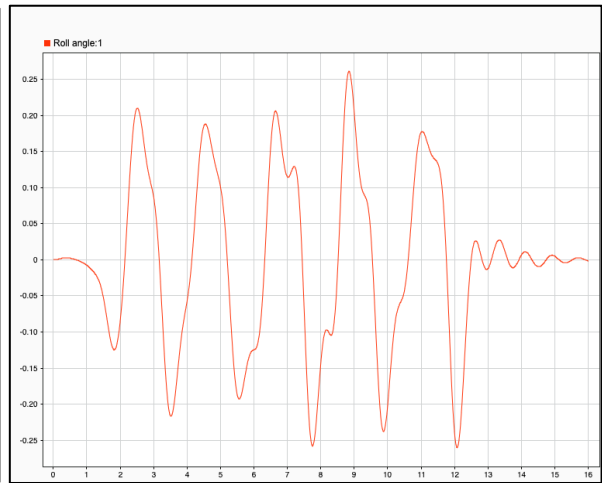


Figure 43: Simulink model roll angle

Pitch Angle

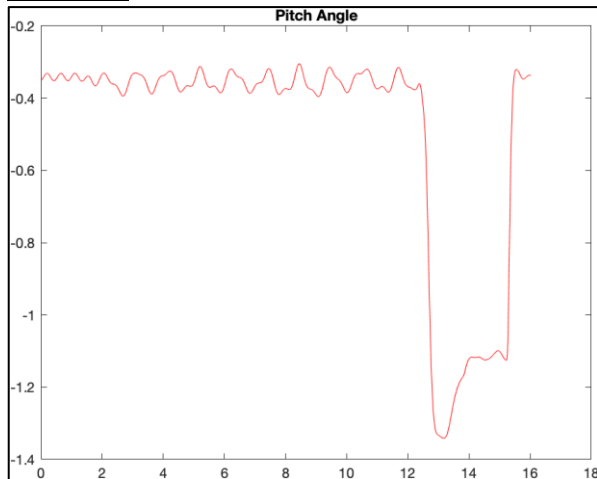


Figure 44: MoTec Pitch angle

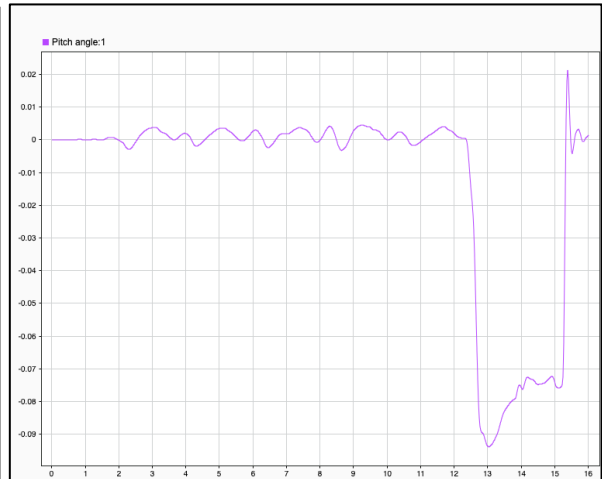


Figure 45: Simulink model Pitch angle

Setup3 Slalom change driving response comparison

Roll Angle

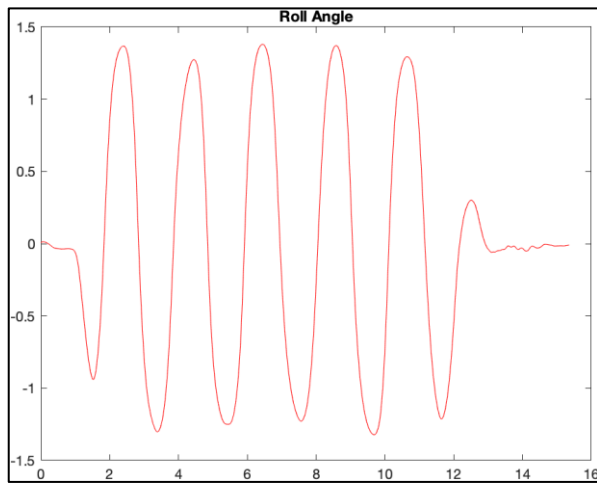


Figure 46: MoTec roll angle

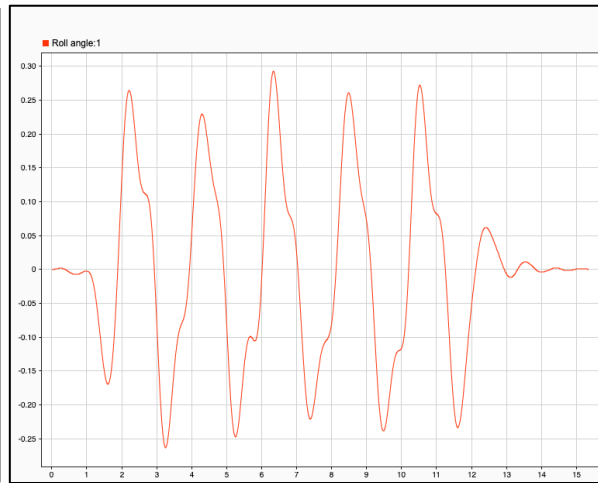


Figure 47: Simulink model roll angle

Pitch Angle

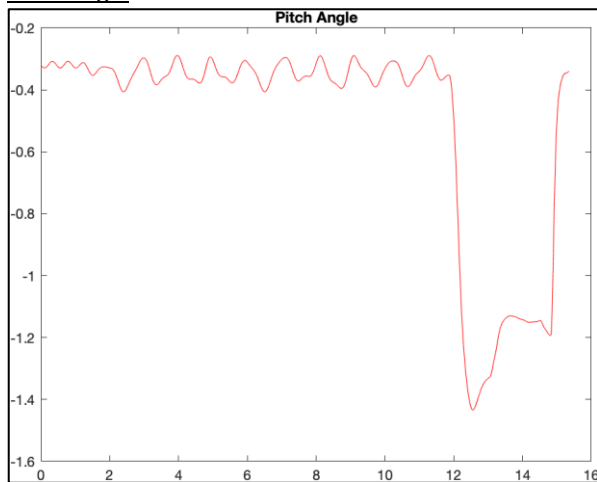


Figure 48: MoTec Pitch angle

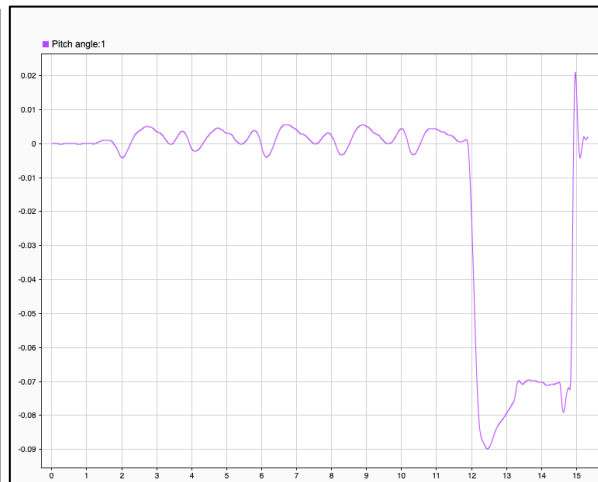


Figure 49: Simulink model Pitch angle

Comparison of vehicle response under spring rate change

In the last, using Simulink model of formula ford car, I simulate the model to investigate the response when additional parameters like spring rate are changed and effect of these change parameter in roll angle. For this simulation, I simulate model in slalom driving maneuver. In this line change I consider that the suspension is moving fast and consider fast bump and fast rebound. There is 3 sets of front and rear spring which spring rate are shown in the table 2.

Springs Investigation		Setup 1	Setup 2	Setup 3
Front Spring Rate (N/mm)	Ksf	14	18	23
Rear Spring Rate (N/mm)	Ksr	20	16	12

This model is simulated under the Motec setup 1 for lateral and longitudinal acceleration.

The impact of the spring rate change on the response of the vehicle is shown in the following graph.

Color code for the graph:

Blue- setup1

Orange- setup2

Gray- setup3

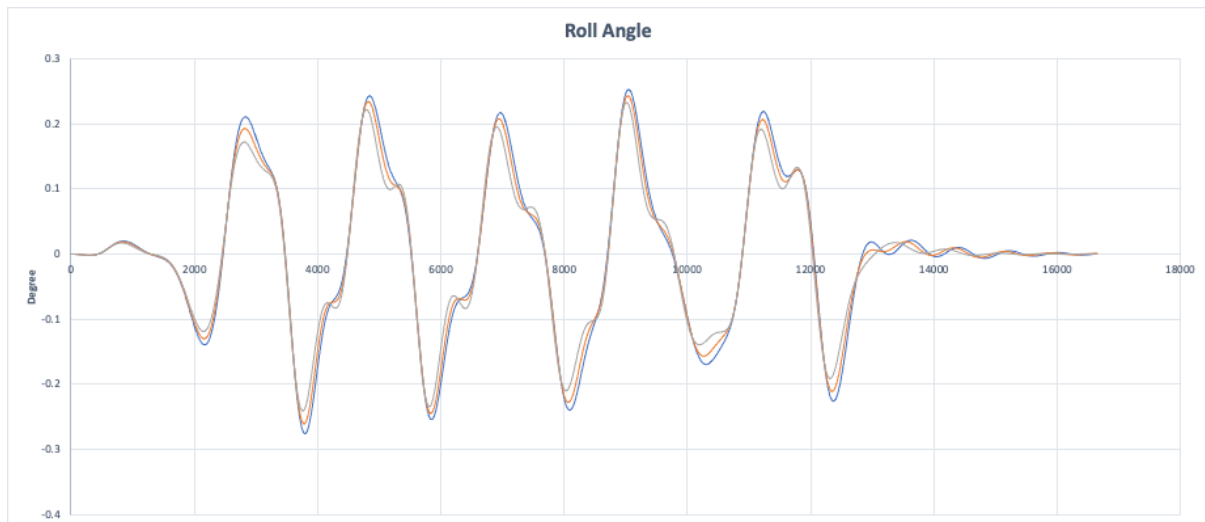


Figure 50: Impact of spring rate change on the roll response of the vehicle

As shown in the figure 50, the response of the vehicle over the changing the suspension spring rate. There is not any other change apply in this comparison. we consider setup1 as a base response and now compare other two response of the vehicle, the roll angle is decrease as compared to softer spring then stiffer spring.

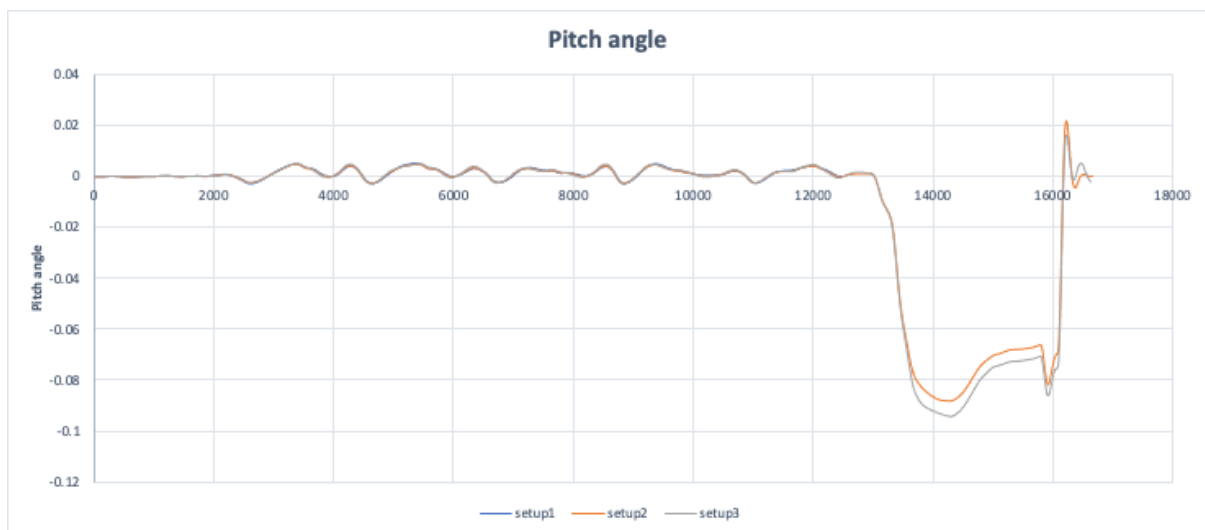
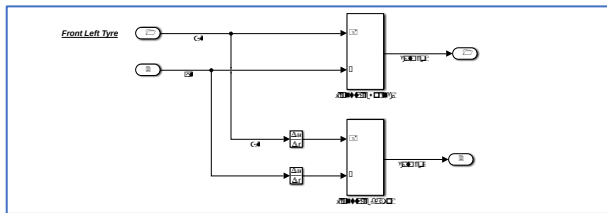


Figure 51: Impact of spring rate change on pitch angle response of the vehicle

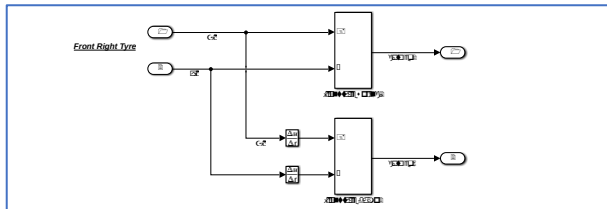
Conclusion

The spring rate in car suspension affect the roll and pitch angle. Higher spring rate spring will resist the car to roll, because high spring rate spring will resist the mass transfer during cornering or lane change. But more stiffer spring can also affect on the ride comfort. Softer spring can transfer load during cornering so increase of soft spring the roll angle will be high then in stiffer spring suspension.

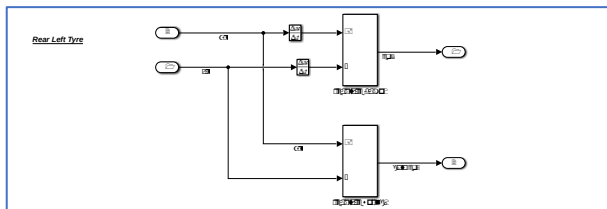
Appendix



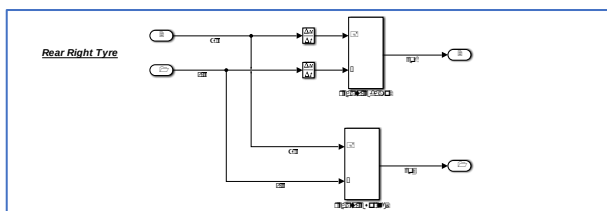
Front left tyre



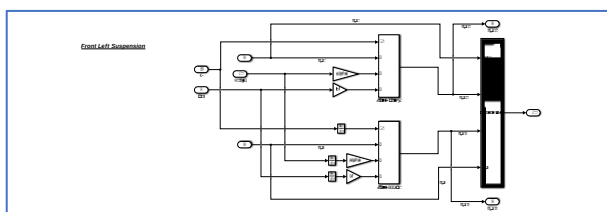
Front right tyre



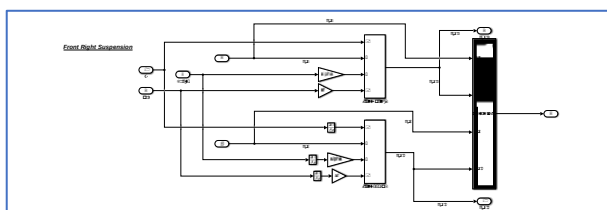
Rear left tyre



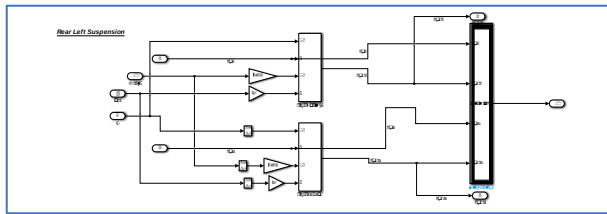
Rear right tyre



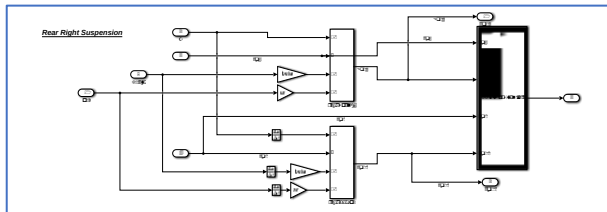
Front left suspension



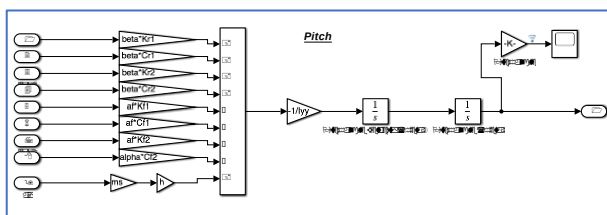
Front right suspension



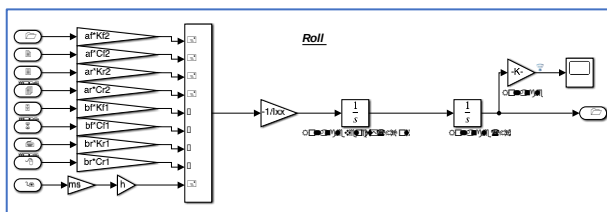
Rear left suspension



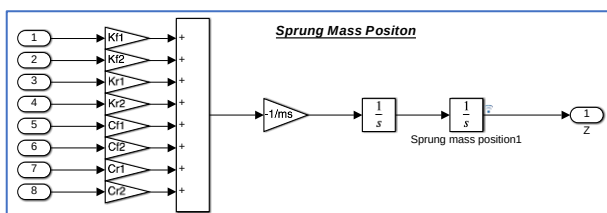
Rear right suspension



Pitch motion



Roll motion



Sprung mass position

References

1. <https://endaq.com/pages/single-degree-of-freedom-systems-basic-concepts>
2. <https://sites.google.com/site/supernumer1me421lab5/home/1-dof-results>

3. Mhanna, Mohannad & Sadek, Marwan & Shahrour, Isam. (2012). Numerical modeling of traffic-induced ground vibration. *Computers and Geotechnics*. 39. 116–123.
10.1016/j.compgeo.2011.07.005.
4. Azman, M. & King, Paul & Rahnejat, Homer. (2007). Combined bounce, pitch, and roll dynamics of vehicles negotiating single speed bump events. *Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics*. 221. 10.1243/1464419JMBD55.
5. Momiya F. 7 DOF vehicle model for understanding vehicle fluctuation during straight running. SAE Technical Paper; 2003 Nov 10.