ANALYSIS AND CONTROL OF SUSPENSION SYSTEM IN GROUND VEHICLES

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ME 780 Project: Design of Mechatronics System

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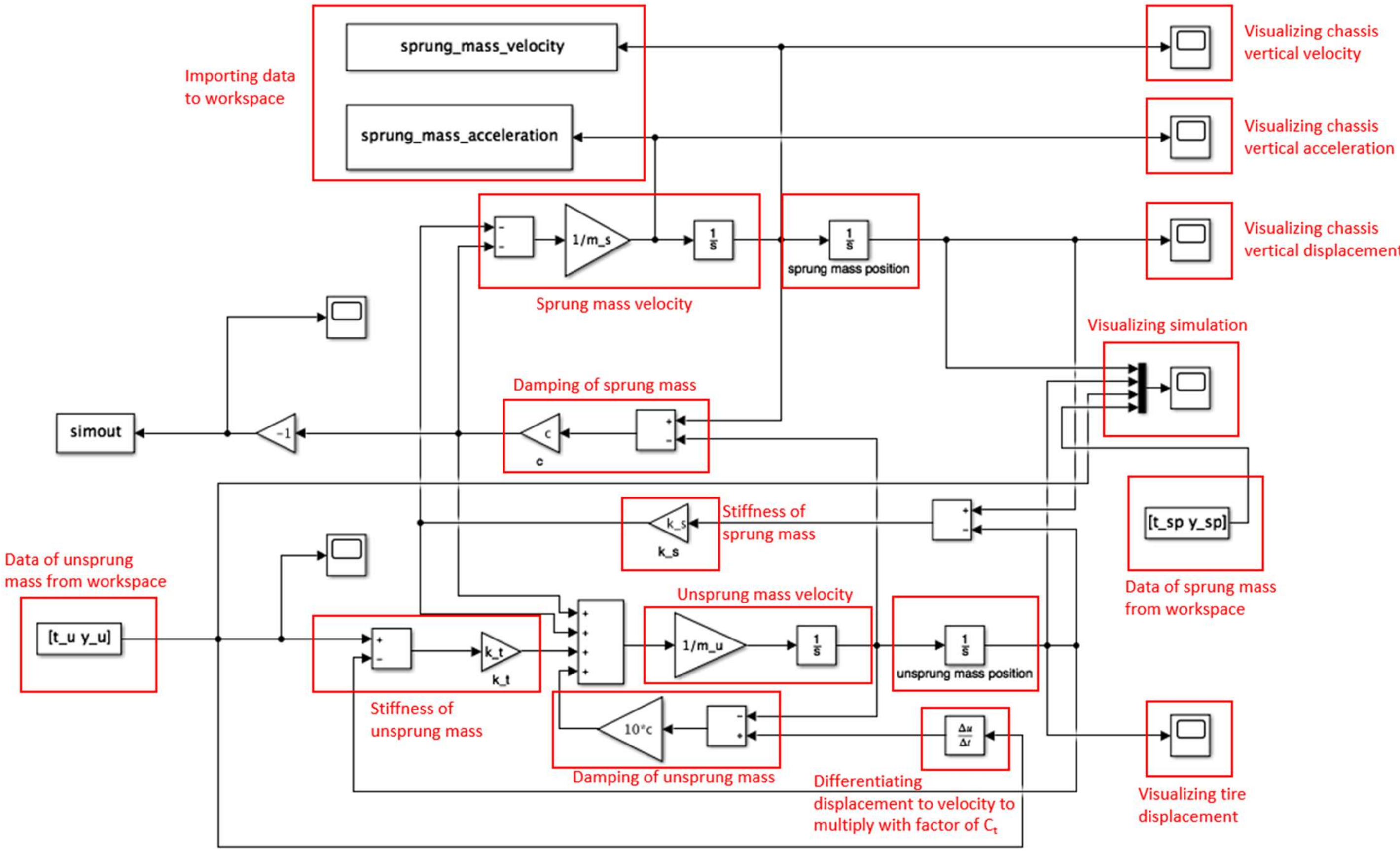
Simulink Model



Simulink Model is enclosed in submission along with the response paper.

Enclosed files in the submission:

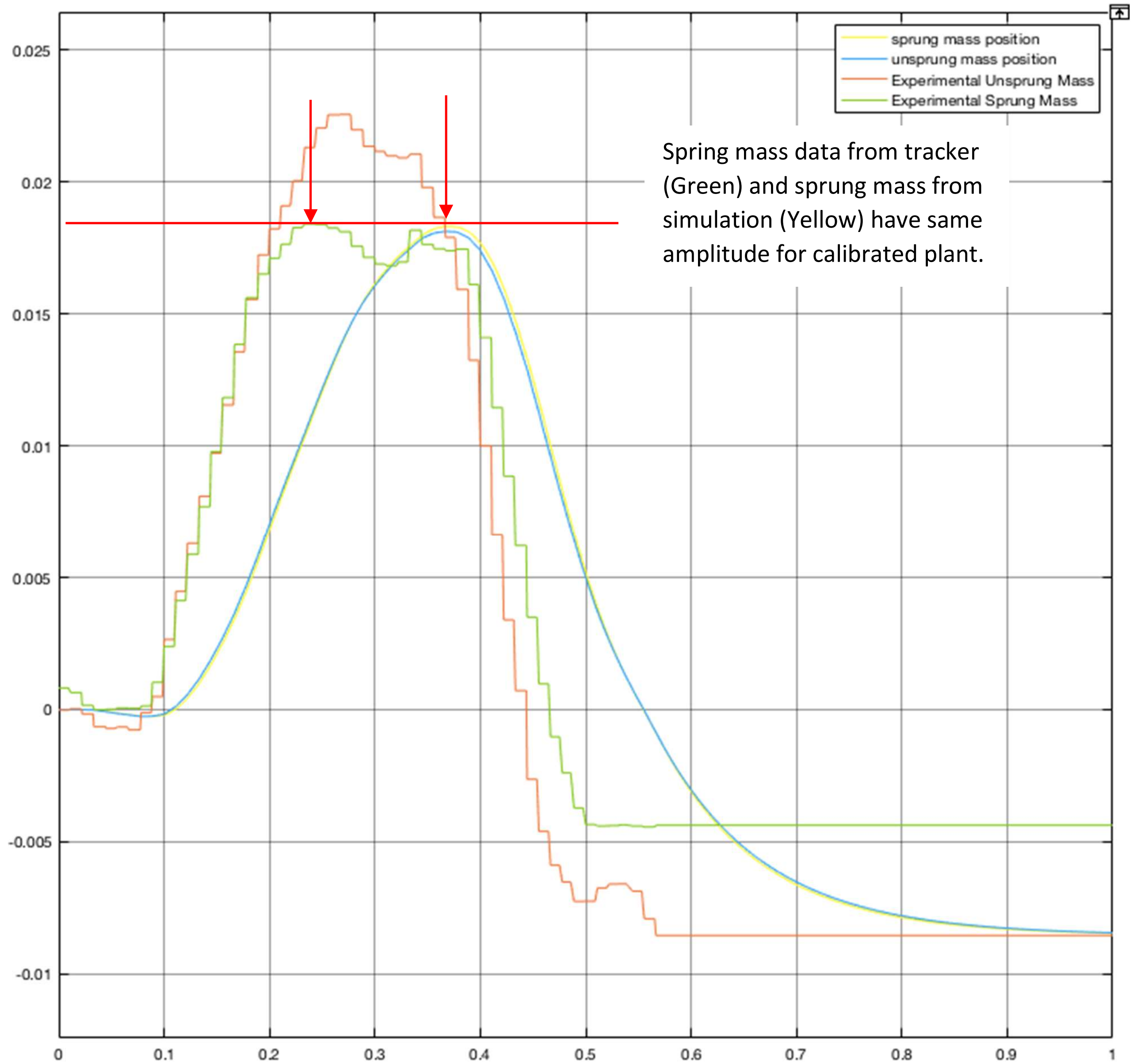
Parent File:	DA3run.m (MATLAB script)
Child File:	gen_modified_haversine.m (MATLAB function)
	DA3sim.slx (SIMULINK model)
Excel Data File:	sprung_slow.xlsx, unsprung_slow.xlsx (for v=1m/s)

NOTE: KEEP ALL THE FILES IN A SAME FOLDER WHILE SIMULATING

The experimental values for k (N/m) and c (Ns/m) as well as the calibrated (installed) constants

- Values of k_s and c_s were change from the initial guess (experimental values) such that the sprung mass amplitude from simulation run matches to that from tracker data.
- Then, the effect of inclined spring-mass of suspension was compensated to find the calibrated values.

Below figure shows that the amplitudes are matching.



Parameter	Experimental Values	Calibrated Values	% Difference
k_s [N/m]	586.39	637.03	8.63 %
k_t [N/m]	10*K_s = 5863.9	6370.3	8.63 %
c_s[Ns/m]	95	65.18	31.38 %
c_t [Ns/m]	10*C_s = 950	651.8	31.38 %

Calibration of system parameters

The values based on the free-body diagram and values obtained experimentally differed. This was due to the fact, that the values obtained from free-body diagram and experimentally had different geometry.

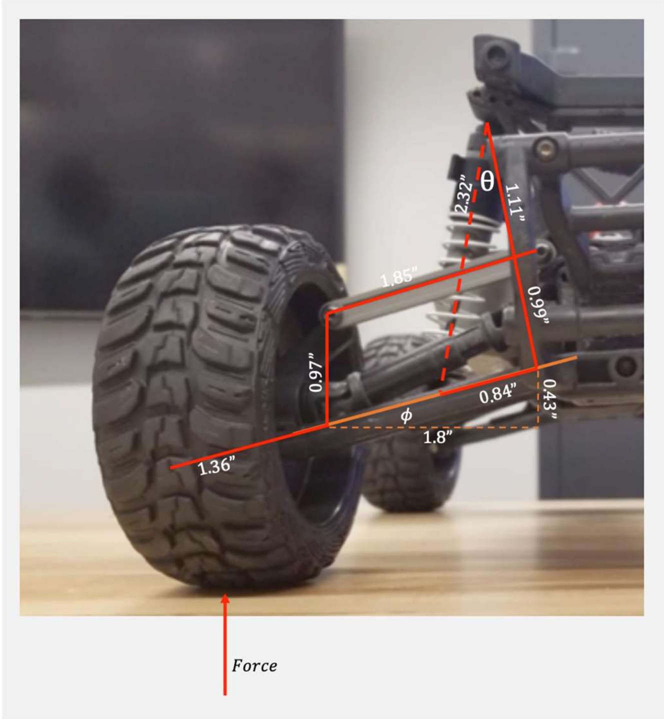


Image: Damper setup for calculation. In the left, free-body diagram setup for the RC car. In the right, shown is the experimental model setup for getting the damping coefficient.

Hence, there is a visible difference in the model setup between the FBD and experiment. The angle θ is 23°, which mean the experimental value K, needs to be divided by $\cos(\theta)$. This will help in minimizing the error from the value obtained in FBD. In addition, the controlling arm is attached to the wheel at an angle ϕ. This means the force acting in the both ends of the damper is not the same direction.

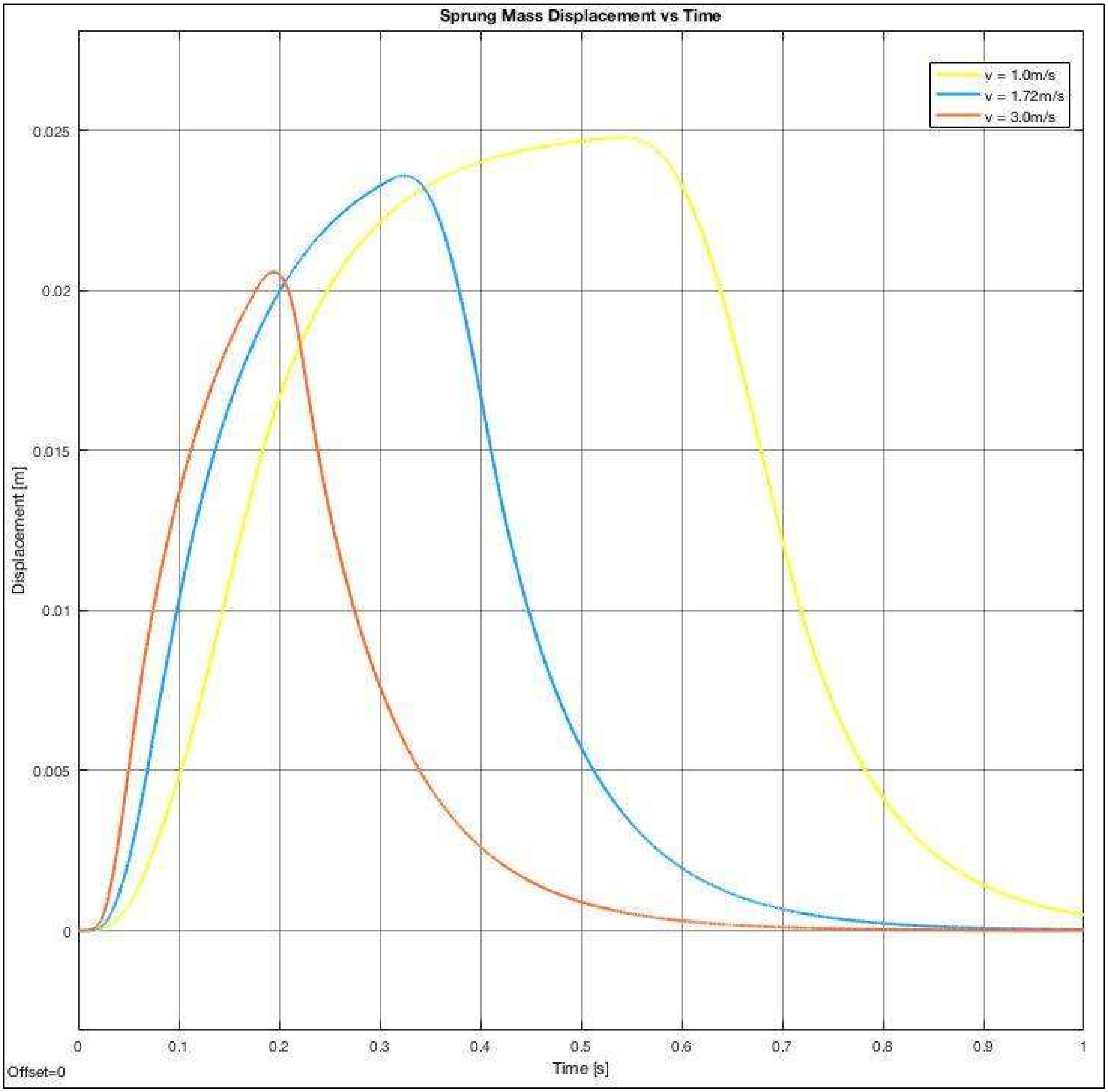
In addition, the damper and spring linkages are attached to the controller arms of the suspension assembly. Hence, as discussed in class, if the dampers are pre-compressed, they will yield a different K value, than that of the damper that is able to full release. In this case, the FBD has a fixed end that do not allow the dampers to fully expand, whereas, the experimental method allow for that freedom. This creates the difference in values between the FBD and experimental calculation.

Moreover, there are non-linearity's present in the chassis, tire and controller arms, where they absorb some force from the road profile. However, in the model this was neglected. As well as, the camber angle of the wheel, and castor angle were also neglected.

Lastly, in the experimental method it was assumed that the loading was at the damper mechanism. However, in the FBD it is not the case. The loading actually occurs on the wheel. With this assumption, there is a difference in the K values between the two methods.

Sprung mass response for the 'prediction case'

Road profile: Haversine up (25 mm amplitude) → flat (116 mm) → haversine down for the following velocities:
Vslow = 1.0 m/sec; Vmed = 1.72 m/sec; Vfast = 3.0 m/sec

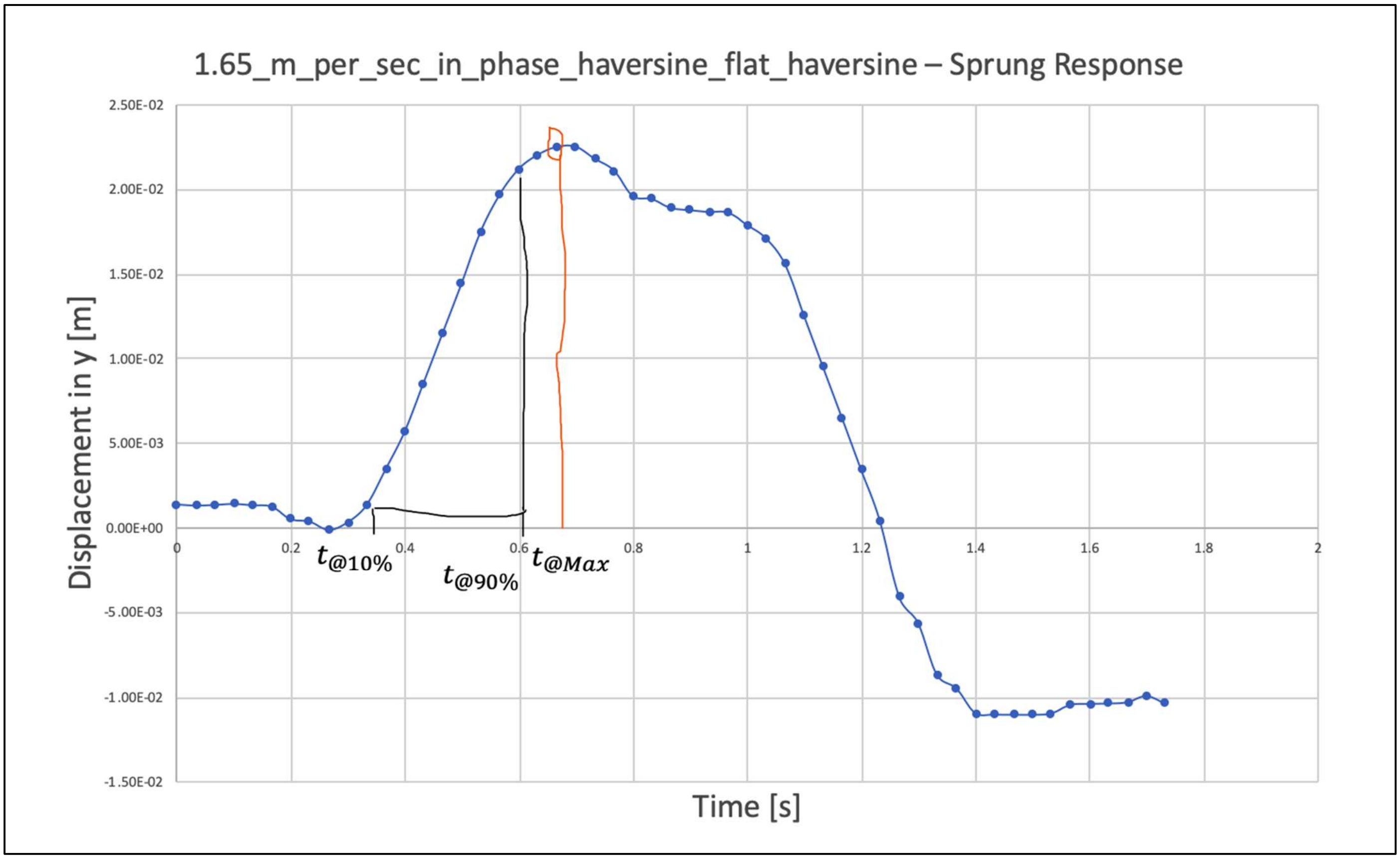


Model response of the sprung mass vs time, with varying velocities.

Q3. Using your ¼ car model, calculate the following metrics about the predicted response and compare them to the provided experimental "prediction case" video data responses. ***Please note that for the experimental cases, some alignment (time/displacement shift) of the response curves may be necessary to obtain accurate calculated results.***

Simulation Case	Peak Amplitude x_s [mm]	Overshoot [%]	Rise time [ms] $\Delta t_{10\% \rightarrow 90\%, x_{ss}}$	Peak Time [ms]	Peak Sprung Mass Acceleration $\ddot{x}_s$ [m/sec^2]
V_{slow} = 1.00 m/sec	24.8	0	322.6	528	2.89

V_{med} = 1.72 m/sec	23.6	0	173.4	317	4.55
V_{fast} = 3.00 m/sec	20.6	0	117	191	7.55
Experimental Case	Peak Amplitude x_s [mm]	Overshoot [%]	Rise time [ms] $\Delta t_{10\% \rightarrow 90\%, x_{ss}}$	Peak Time [ms]	Peak Sprung Mass Acceleration $\ddot{x}_s$ [m/sec^2]
1.32 m/sec	17.97	0	234	733	0.588
1.65 m/sec	22.50	0	267	700	0.766
1.78 m/sec	24.63	0	267	667	1.083
2.03 m/sec	25.10	0	233	600	1.471
2.62 m/sec	27.52	4.92	234	500	1.155



Displacement v/s time plot showing the calculation for experimental case

Comparison between predicted sprung mass responses to the experimental test cases.

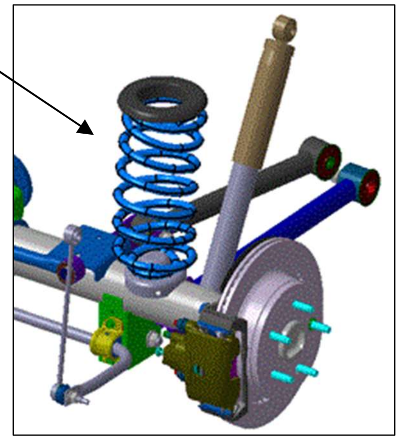
The spring mass response did not behave completely similar to the experimental test case due to following reasons:

1. The mass concentration of the RC car is offset from the spring-damper assembly.
   The SIMULINK model is made by assuming that the spring mass is right above the spring-damper assembly. However, RC car's centre of mass is eccentric from the spring-damper which produces reaction torque along with forces.

2. The SIMULINK model is not accurate model of the plant as it does not consider Motion Ration (MR):

The SIMULINK model is comparatively more relatable to the '**strut style suspension**'. In this type of suspension, the geometry is simple and there are no links to leverage force. The spring-damper is connected directly to control arm.

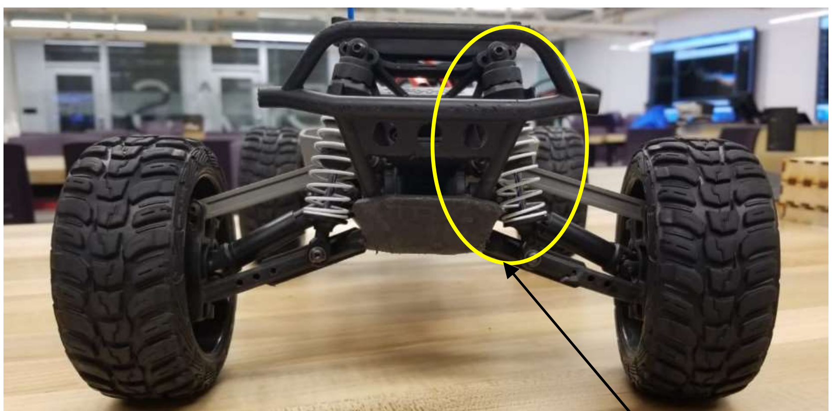
Motion ratio for strut type suspension is generally from 0.92 to 0.95. For an ideal case, it can be considered as unity. Hence the model used for simulation is more relatable to strut type suspension

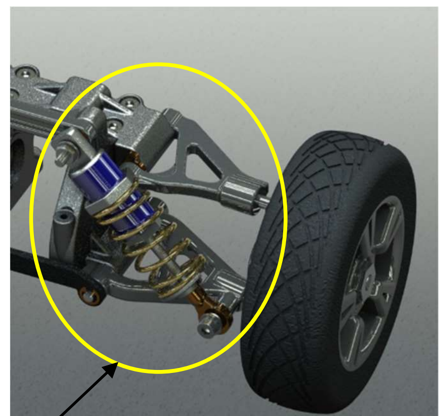


$$\boxed{WR} = MR^2 \cdot \boxed{K}$$

MR = 0.92 – 0.95 for strut type suspension ~ 1.0 for ideal case
Effective sprung mass stiffness = 1.0 x spring's stiffness.

However, RC car has **'double wish bone' suspension,** where motion ratio can generally vary from 0.6-0.8. Hence, the effective spring mass stiffness is 0.35-0.65 times the actual stiffness of the spring.





Double wish bone suspension

MR = 0.6-0.8 for double wish bone suspension
Effective sprung mass stiffness = (0.35-0.65) x spring's stiffness.

For commercial cars, in general, the front wheel spring stiffness is kept more than that of rear. For example in BMW M240, front wheel stiffness if 8000 N/m while of rear is 6000 N/m.

But in some cars, for instance in BMW E36, the rear wheel spring stiffness is 12,000 N/m. The effect of over-stiffness is diminished by motion ratio of 0.7. Hence, the effective spring mass stiffness experienced by tyres is 6000 N/m.

There are certain advantages in vehicle cornering performance and handling of using stiffer rear springs with high MR.

3. Limitation of the data extraction software:

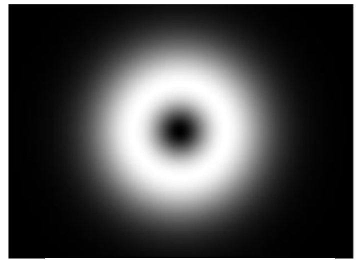
   There is always a probability that the data is not extracted accurately due to:

a) Human intervention in data extraction to define coordinate axes and other parameters (for tracker software).

b) The software is inconsistent in tracking the exact pixel:

Both data extraction methods (MATLAB script and tracker software) uses 'gaussian filter' in image processing. The led light sampled is not perfectly uniformly distributed light gradient. Hence, there are chances that after using gaussian filter, there are chances of picking wrong pixel.

LED sample from the video
(not uniformly distributed)



Perfectly uniformly
distributed light sample.