OXFORD BROOKES UNIVERSITY

ADAVANCED VEHICLE DYNAMICS PORTFOLIO

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MODULE

P04731: ADVANCED VEHICLE DYNAMICS

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Contents

“Throttle, Green, Amber. Change. Brake, turn the wheel, point it at a corner, accelerate. Simple. The challenge is doing it faster than everybody else without losing control.”

ADRIAN NEWEY

1. INTRODUCTION

Enzo Ferrari once said, “Aerodynamics are for people who can’t build Engines”. Fast forward to now, and I cannot believe how wrong was the great Enzo. Aerodynamics is very essential, in fact one of the most important aspects of the current Formula one car or for that matter any racing and road car. Vehicle Dynamics is not only a crucial point but also the starting point for the design of a car.

In this document we are going to discuss the dynamic behaviour of a sample car using advanced tools like ADAMs and basic tools like Excel. In order to do that five important areas of vehicle dynamics would be thoroughly analysed with complex models and comparisons.

2. STRAIGHT LINE ACCELERATION

One of the primary and important aim of this portfolio was to determine a realistic straight-line performance of the car. This is done by considering various factors like aerodynamics, weight transfer, longitudinal force and its two regimes, etcetera.

An excel spreadsheet depicting all these and laying out a final time for 0-100 is dished out.

RPM	TORQUE Nm
6000	240
7000	260
8000	280
9000	310
###	340
11000	350
###	355
###	355
###	353
###	348
###	342
###	330
###	315
###	315
###	310

Aero @100mph	1.7
Mu	1.2
Torq Loss Nm	15
wheel rad m	0.35
aver rad m	0.29
car mass(N)	7358
Cd	1.1
p (kg/m^3)	1.29
Frontal area(m^	1.1
Wheel Mass (kg	12
CG (ht)	0.3
Wheelbase	3
Inertia of rot (I)	3
wt distribution	0.5

gear	ratio	final drive	tot
1	2.85	5.76	16.42
2	2.21	5.76	12.73
3	1.85	5.76	10.66
4	1.41	5.76	8.122
5	1.33	5.76	7.661
6	1.11	5.76	6.394

mph	vel m/s	whee V	whee rpm	Engine rpm 1st	2nd	3rd	4th	5th	6th	rpm used
0	0	0	0	0	0	0	0	0	0	7500
10	4	12.8	122	2001	1553.4	1306	990.9	934.8	780	10000
20	9	25.5	244	4003	3099.6	2611	1982	1870	1560	10000
30	13	38.3	366	6004	4649.4	3917	2973	2804	2339	10005
40	18	51.1	488	8005	6199.2	5223	3964	3739	3119	10090
50	22	63.9	610	10007	7749	6529	4954	4674	3899	11064
60	27	76.6	732	12008	9298.8	7834	5945	5609	4679	13870
70	31	89.4	854	14009	10849	9140	6936	6543	5458	15654
80	36	102	976	16010	12398	10446	7927	7478	6238	17098
90	40	115	1098	18012	13948	11752	8918	8413	7018	19748
100	45	128	1220	20013	15498	13057	9909	9348	7798	16270
110	49	140	1342	22014	17048	14363	10900	10282	8578	18023
120	54	153	1464	24016	18598	15669	11891	11217	9357	19156
130	58	166	1586	26017	20147	16975	12882	12152	10137	17456
140	63	179	1708	28018	21697	18280	13872	13087	10917	19899
150	67	192	1830	30020	23247	19586	14863	14021	11697	16540
160	72	204	1952	32021	24797	20892	15854	14956	12476	18099
170	76	217	2075	34022	26346	22197	16845	15891	13256	19543
180	80	230	2197	36024	27896	23503	17836	16826	14036	17946
190	85	243	2319	38025	29446	24809	18827	17760	14816	19365
200	89	255	2441	40026	30996	26115	19818	18695	15596	17057

mph	Engine Torque	gr	Wheel Torque	Long F a Wheels frm torq	Long F rear due to Mu	Aero Load	Actu Long	Load Rear	acce
0	260	1	4268.2	14717.79	6180.64	0	6181	5151	8.2
10	340	1	5581.4	19246.34	6724.06	123	6724	5603	9
20	340	1	5581.4	19246.34	6724.06	493	6724	5603	9
30	340	1	5581.4	19246.34	6724.06	1110	6724	5603	9
40	340	1	5581.4	19246.34	6724.06	1973	6724	5603	9
50	350	1	5745.6	19812.41	6791.99	3083	6792	5660	9.1
60	355	1	5827.7	20095.45	6825.95	4439	6826	5688	9.1
70	348	1	5712.8	19699.2	6778.4	6042	6778	5649	9
80	330	1	5417.3	18680.28	6656.13	7891	6656	5547	8.9
90	315	1	5171	17831.17	6554.24	9988	6554	5462	8.7
100	342	2	5614.3	19359.56	6737.65	###	6738	5615	9
110	315	2	5171	17831.17	6554.24	###	6554	5462	8.7
120	315	2	5171	17831.17	6554.24	###	6554	5462	8.7
130	330	3	5417.3	18680.28	6656.13	###	6656	5547	8.9
140	315	3	5171	17831.17	6554.24	###	6554	5462	8.7
150	342	4	5614.3	19359.56	6737.65	###	6738	5615	9
160	315	4	5171	17831.17	6554.24	###	6554	5462	8.7
170	315	4	5171	17831.17	6554.24	###	6554	5462	8.7
180	330	5	5417.3	18680.28	6656.13	###	6656	5547	8.9
190	315	5	5171	17831.17	6554.24	###	6554	5462	8.7
200	330	6	5417.3	18680.28	6656.13	###	6656	5547	8.9

Drag	Weight Transfer	Load Front(	Load Rear(N)	Actua .ong. F	acce	Tim	tot time
0	1471.779	2207	5150.52931	6181	8.24	0	0
15.6	1924.634	1754.1	5603.384483	6724	8.97	0.5	0
62.4	1924.634	1754.1	5603.384483	6724	8.97	1	1
140	1924.634	1754.1	5603.384483	6724	8.97	1.5	3
250	1924.634	1754.1	5603.384483	6724	8.97	2	3
390	1981.241	1697.5	5659.991379	6792	9.06	2.5	7
561	2009.545	1669.2	5688.294828	6826	9.1	2.9	10
764	1969.92	1708.8	5648.67	6778	9.04	3.5	14
998	1868.028	1810.7	5546.777586	6656	8.87	4	18
1263	1783.117	1895.6	5461.867241	6554	8.74	4.6	22
1560	1935.956	1742.8	5614.705862	6738	8.98	5	27
1887	1783.117	1895.6	5461.867241	6554	8.74	5.6	33
2246	1783.117	1895.6	5461.867241	6554	8.74	6.1	39
2636	1868.028	1810.7	5546.777586	6656	8.87	6.5	46

Table 1: Straight Line Acceleration Spreadsheet

2.1 GRIP LIMITED FORCE ACTING LONGITUDINALLY

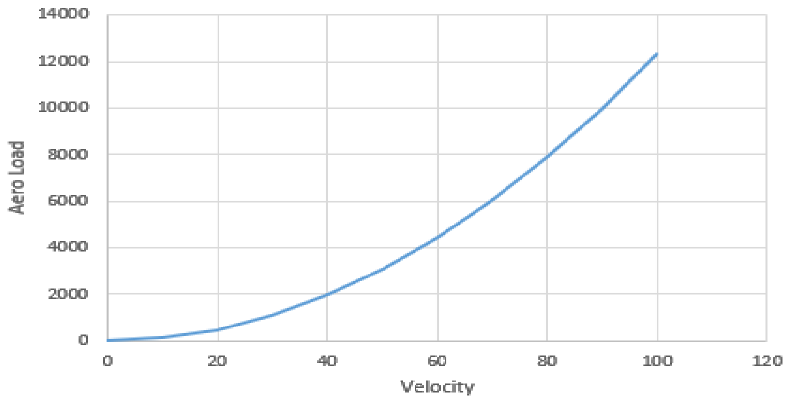
The two main regimes of Longitudinal Force are torque limit and grip limit. In the basic version it is assumed that the tyres would always grip the road, however low gear vehicles are very much capable of providing much more torque than the tyres can transfer to the road. This would result in wheel spinning and much less tractive force being generated. Therefore, it is desired to have mu value as large as possible.

2.2 LOAD DUE TO AERODYNAMIC FORCES

The vehicle is supposed to be slower in the straight-line acceleration run, with the front wing and rear wing in position, as they aid in increasing the cornering performance of the car rather than straight line performance.

2.3 DRAG

Drag force acting in the car has many sources. Rolling resistance of tyres, friction acting on the rotating components and aerodynamic drag.



Aero Load (Drag and Force) vs Velocity

Aerodynamic Drag is the largest contributor to the total drag force. In formula 1 it is so high that the car can generate a decelerating force of around 1G even without the brakes. Drag force is related to the frontal area of the car, velocity, density of air and drag coefficient.

$$F_D = \frac{1}{2} \rho A C_D V^2$$

The force required to roll the tyres along the road is called the Rolling Resistance. The rubber carcass deforms every time it comes in the contact patch and a large amount of rolling resistance is observed from the flexing of the rubber as it passes through the deformed shape near contact patch.

2.4 WEIGHT TRANSFER

When a vehicle accelerates or brakes, there is pitching and diving. This results in transfer of weight from front to rear providing more grip at the rear tyres when accelerating in a straight line. This can be a good thing if the vehicle has rear wheel drive.

The weight transfer is calculated using the formula;

$$Wt = h.m.\frac{g}{l}$$

This shows how the weight transfer depends on height of CG, mass, and wheelbase. Higher the CG, more would be the weight transfer.

2.5 CONCLUSION

The main objective of this section of the portfolio was to analyse a realistic straight-line behaviour of a vehicle. Several important parameters that would affect the acceleration of a car in straight line has been added on to the basic spreadsheet.

PARAMETERS	0-60 MPH TIME (SEC)
BASIC WITH WEIGHT TRANSFER	2.34
AERODYNAMIC FORCES	2.42
AERODYNAMIC DRAG	2.68
ROLLING RESISTANCE	2.77

A comparison of the effectiveness of the add-on parameters have been draw, and the findings have been listed down in the table below.

3. SUSPENSION PERFORMANCE

The main aim of this section of the portfolio is to explain the understanding of one's knowledge on suspension by explaining the table below and justifying the explanation using the model made in ADAMs.
To perform analysis of the suspension, a SDoF model and a 2DoF model has been made on the multi simulation software tool ADAMs. The 2DoF is represented below.

3.1 SUSPENSION MODELLING

A simple 2DoF model of the suspension was made in ADAMs Multi simulation software. The model is shown below. The top block is representative for the body of the car weighing 906kg, the block in the middle acts as the upright of the car which weighs another 90kg and the lowest block represents the tyres, the most important aspect of a car.

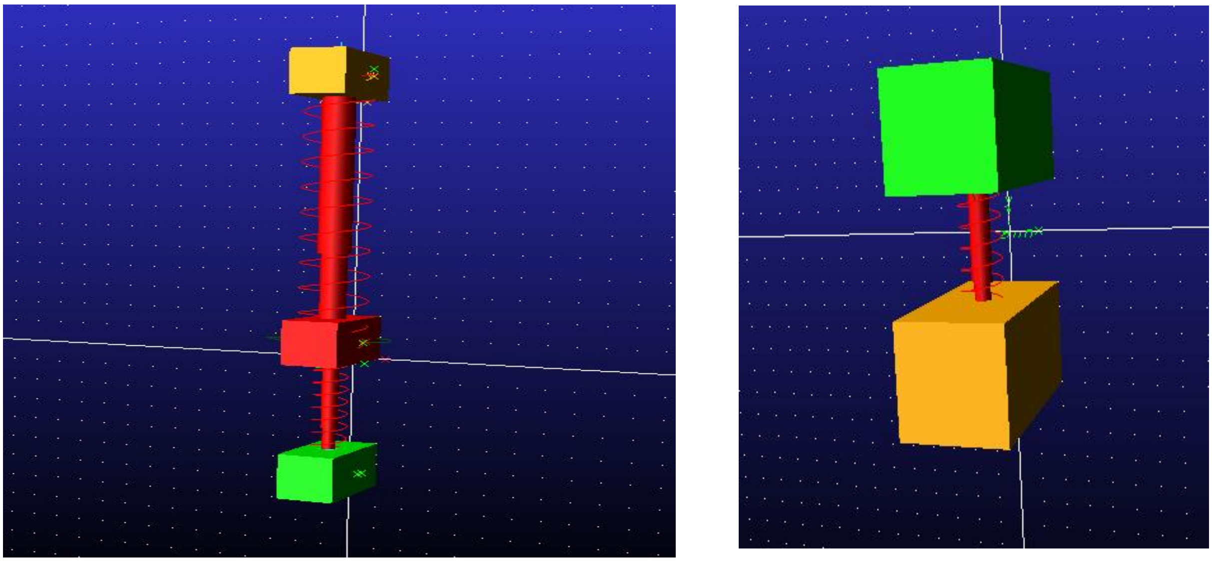


Figure 1: 2DoF Representation of Suspension Model in ADAMs; SDoF Model

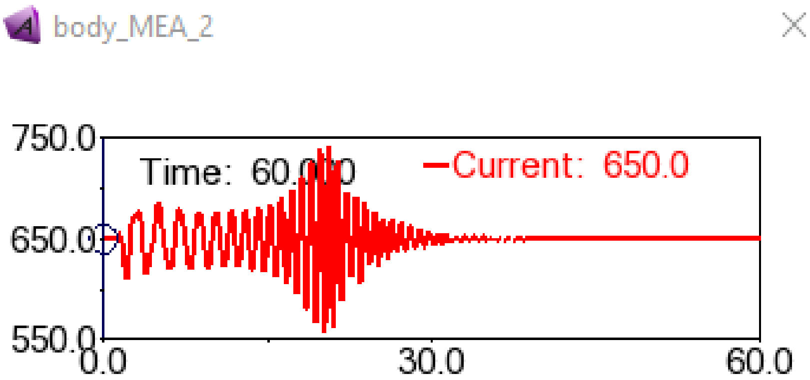


Figure 2: body vertical motion

The above shown model 2DoF model was simulated with swept sin wave input for deciding an optimal spring and damper value. This value apparently happens to be optimised for a certain condition and would therefore change from vehicle to vehicle.

3.2 SUSPENSION IMPROVEMENTS

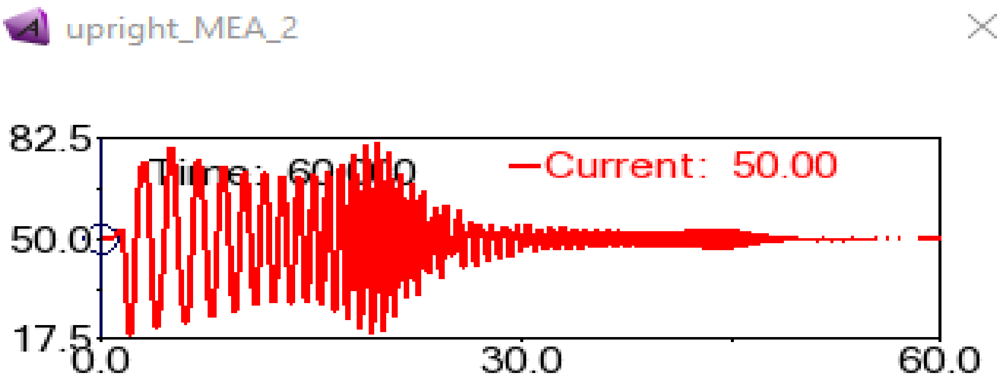


Figure 3: upright vertical motion

For the system a vertical tyre stiffness of 1920N/mm was used with the damping of the tyre being 0. In the main spring-damper system, two design variables were used at two

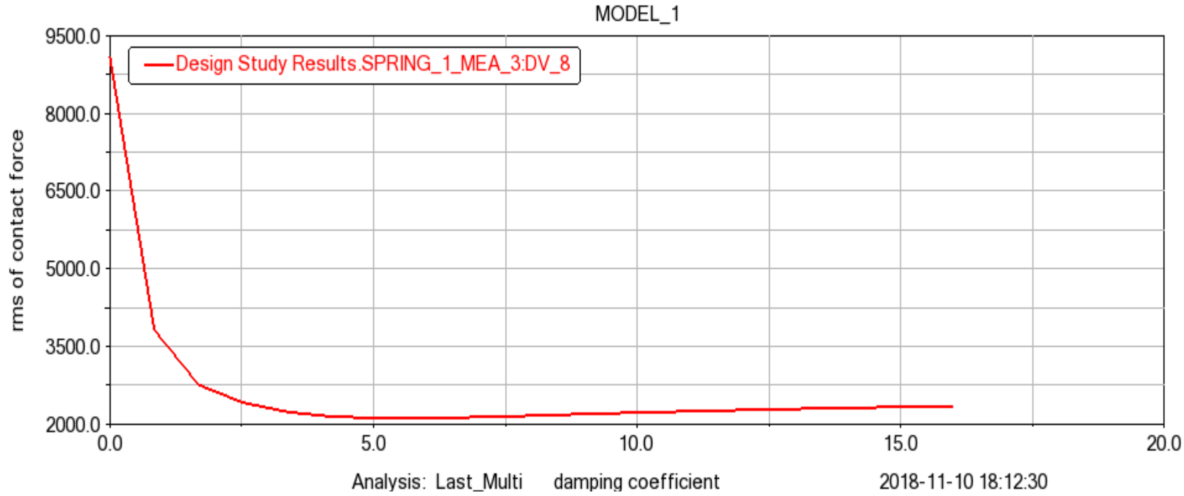


Figure 4: Suspension Optimisation

different times, one to vary the spring coefficient and the other one to vary the damping coefficient. The damping coefficient of the spring damper system can be assessed with the above graph. An optimum damping coefficient value which is 3.81 has been observed for a range of frequency from 0.4Hz to 25Hz. The optimisation process is carried out with the design variable created for varying the frequency in the above-mentioned range.

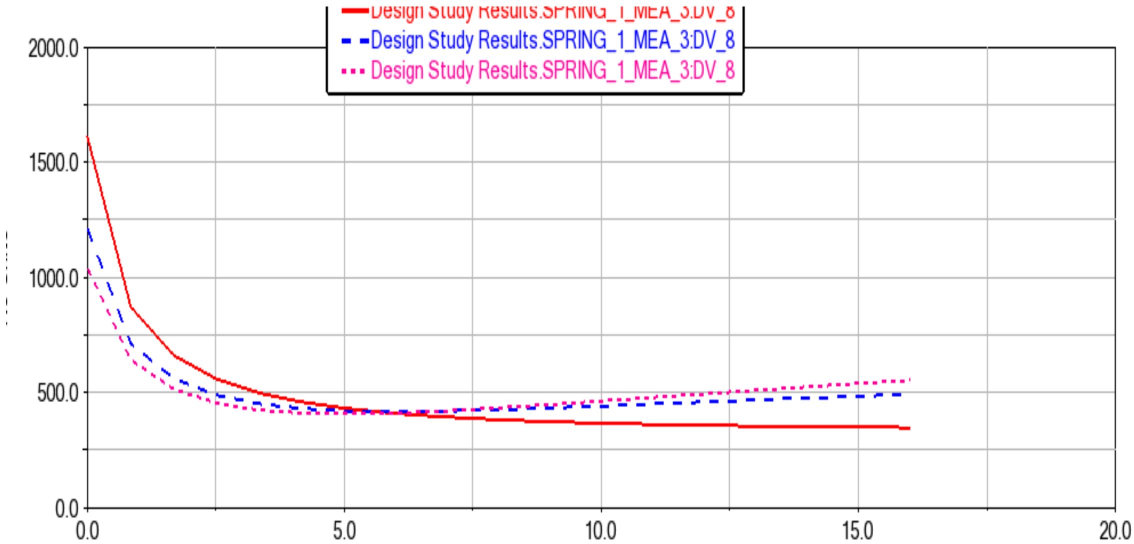


Figure 5: Suspension Optimisation for different road inputs

The graph above shows the variation of damping coefficient when varied with road profile input. The first curve shows the damping value varied over the same frequency range as before but with road profile input of 2x natural frequency,4x natural frequency and 6 times. The optimum value of damping obtained is 15.98 N-sec/mm.

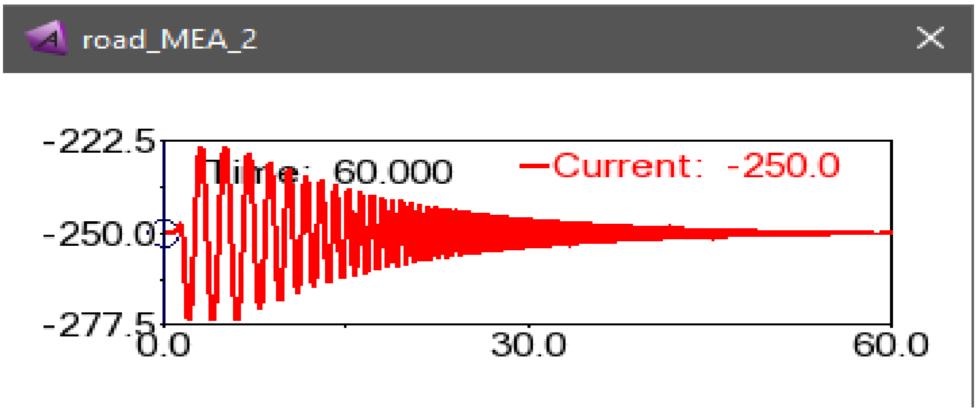


Figure 6: Road Displacement

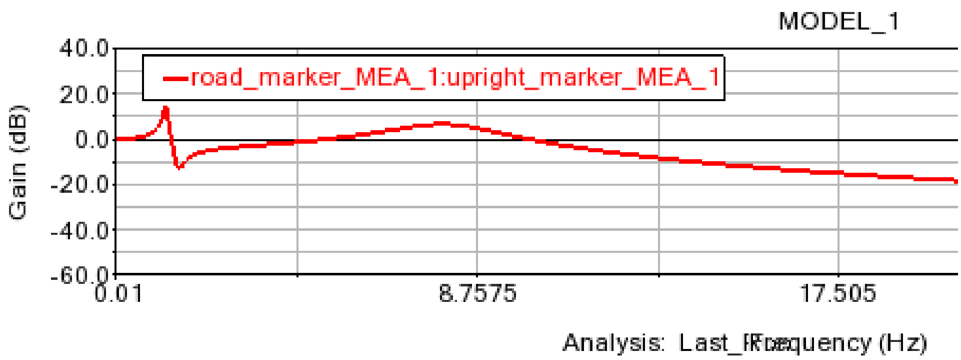
The real road input would help in providing a better and more accurate values. This has been done with the help of spline function. An excel sheet was prepared to just gather all the relevant information and was later imported in to ADAMs and was made a spline function.

3.3 UPRIGHT MOTION

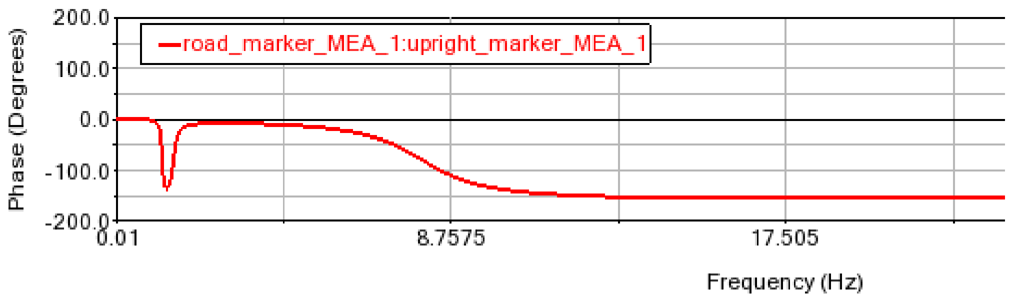
Further improvements can be made by adding in the real upright motion. The motion of the uptight in the above shown 2DoF model is considered to be translational in Y-Axis. However, a more real-life scenario would be when the upright motion is more along a curve with a centre somewhere. This could be an easy improvement as it would not only help in achieving aa more realistic motion but also help in optimising the suspension better.

3.4 BODE PLOT

Bode plot is a graph which gives frequency response of a system. Sine waves are used to represent the frequency response and a bunch of mathematical calculation called Fourier Theory is involved.

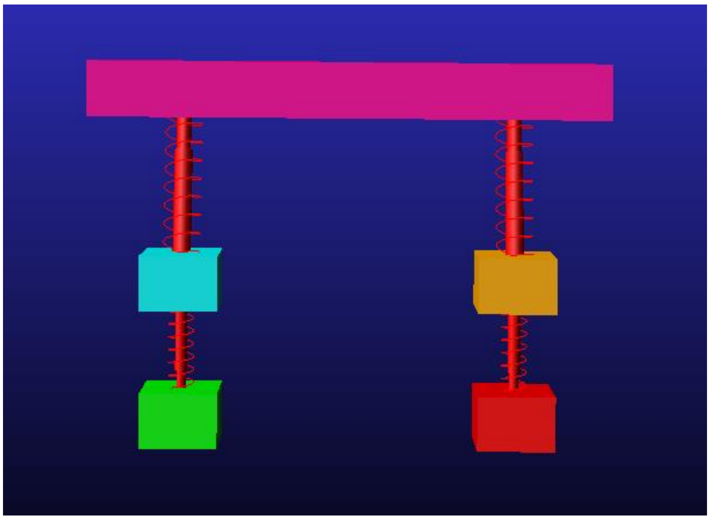


Bode plot is used to study the gain and phase of the 2DoF suspension system. The graphs are obtained in the multi simulation software ADAMs and are given below.



3.5 4DoF MODEL

A 4DoF model of the suspension can also be made in the ADAMs. This would be a heavy improvement from the SDoF and 2DoF models. The 4DoF model can be very helpful in understanding the yaw, roll and warp conditions and how the suspension model copes up with them.



4DoF Model made in ADAMs

The study could be very complex as well depending on the values sorted after and even more complex models can be made. the complexity of the model made can be improved and increased drastically depending on the accuracy sorted after.

3.6 FINDINGS

The suspension model was optimised and checked for two scenarios. Both of them being the difference in road input. The first was harmonic road profile input and it was found that the idea damping ratio for the rms of contact patch force is 0.3 and for the rms of body accel is 0.33.

The second input was pot hole step input. Both the cases discussed above were analysed again and it was found that the ideal damping ratio for rms of contact patch force is 0.17 and the rms of body acceleration is 0.41.

4. BICYCLE MODEL

The part of the portfolio is very important. The bicycle model can help a vehicle dynamist in understanding the steering response of the car.

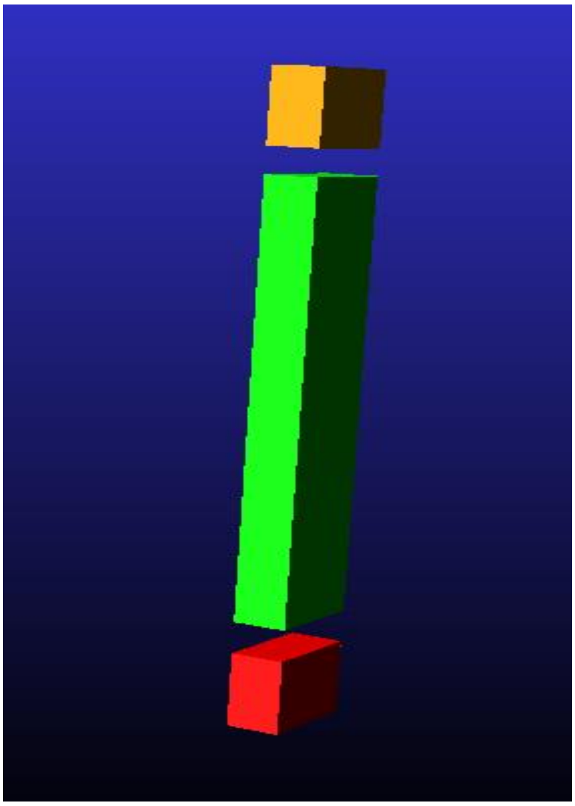


Figure 7: Bicycle Model

The model consists of three blocks representing front wheel, rear wheel and the body. Bicycle model is made with basic tools and with step steer input and run at first with no modifications. Once the model is running perfectly, the changes are made to the model with respect to the car which helps in obtaining the CoG location, etc.

The yaw velocity of the model was measured at the same speed mentioned above and the finding is shown below in graphical manner.

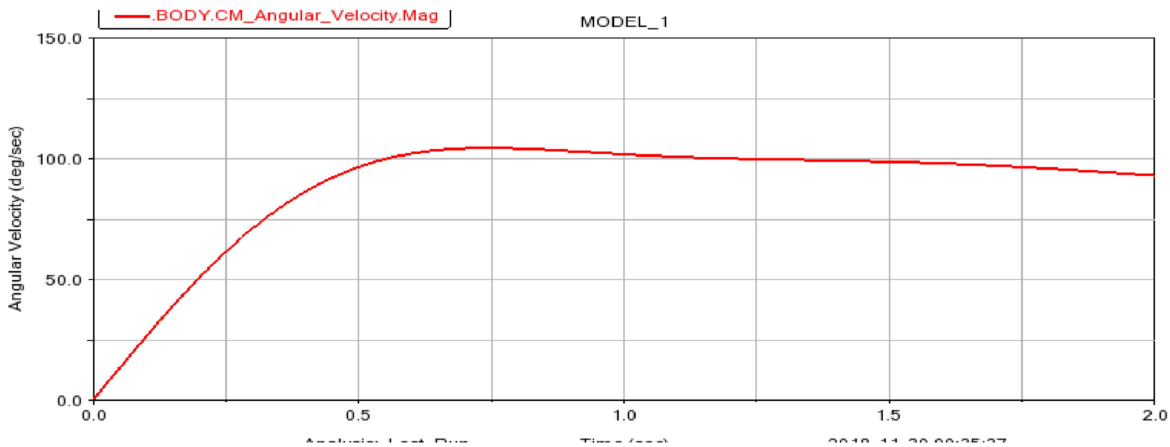


Figure 8: Yaw Velocity with respect to time

The model is run for 5 seconds and at a constant speed of 80mph. however this result is far from reality as will be seen in the next section.

4.1 IMPROVEMENTS

The bicycle model can be improved in a number of ways. In this portfolio the improvement comes along as the introduction of tyres. In the basic analysis forces in the tyres are assumed to be linear.

In order to improve this to attain more realistic results, Pacejka's tyre model is introduced in the bicycle model.

$$Fx = F.D.Sin(C.\arctan[B.k - E(B.k - \arctan[B.k])])$$

Pacejka's tyre formula is defined with the magic equation given above. The force values are calculated with the small car sample values used for this portfolio. The values of B, C, D & E are found to be changing with the atmospheric conditions and hence are taken from the table shown in the book written by Pacejka himself titled as fundamentals of tyre.

A syntax in ADAMs software was written defining the force on the front tyre. Since the constants in the tyre model varied with track conditions, a dry atmospheric condition was assumed.

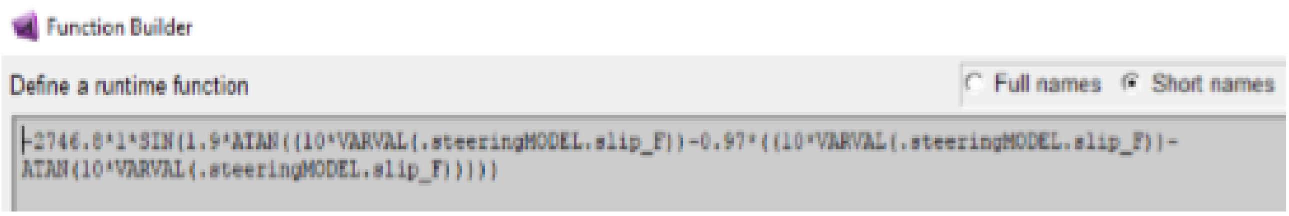


Figure 9: Tyre Model syntax in ADAMS

The bicycle model was run for 10 seconds with step steer input given after 2 seconds. The yaw velocity of the bicycle model with Pacejka's tyre formula was observed and the finding is shown blow in graphical manner.

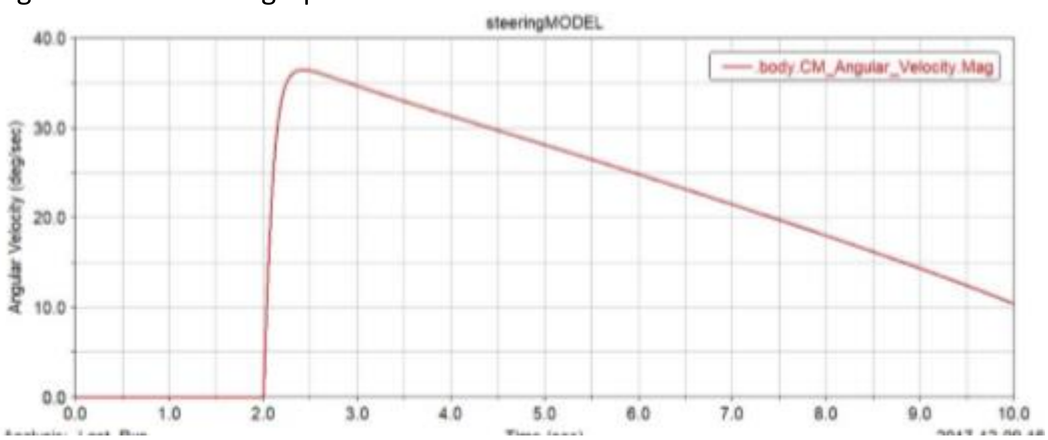


Figure 10: Yaw Velocity with Tyre model

The yaw velocity obtained with the tyre input changes drastically for the same car running at same velocity but longer. While the simulation here is run for 10 seconds the simulation in the model without the tyre model was run for just five seconds.

5. DERIVATIVES ANALYSIS

It is essential to know how to optimise the step steer of the car during cornering. In this section the step-steer response of the sample formula car is analysed. Microsoft Excel is the preferred choice of weapon in this case. A spreadsheet each for Underdamped, critically damped and Overdamped steering responses have been made.

This spreadsheet combined with the findings on bicycle model developed in ADAMs paints a very clear picture in developing the knowledge of the vehicle dynamist. The main aim of this is to obtain yaw rate for each condition and compare the that with the results obtained from the bicycle model developed in ADAMs.

5.1 UNDERDAMPED CASE

The spreadsheet does not consider tyre model and for comparing the basic bicycle model without the tyre model is used.

Input Parameters		
Mass	Kg	950
Polar Mon	Kg.m^2	1200
Weight F/Weight R		65/35
CoG-front	m	1.9
Wheelbas	m	2.7
Cornering	N/deg	-1066
Cornering	N/deg	-900

Derived Parameters		
CoG-rear a	m	0.8
Cf in N/rad		-61077.3
Cr in N/rad		-51566.2

Derivatives		
Yβ		-112643
Yr	/V	-74793.9
Yδ		61077.29
Nβ		40102.46
Nr	/V	-218106
Nδ		53748.02

Input Parameters for Underdamped Response		
Speed	mph	80
delta	rad	0.4

Derived Parameters for Underdamped Response		
Understeer Grad	deg/G	2.518126468
rdot t=0	rad/sec^2	-17.91600531
r t=inf		1.696992699
Speed	m/s	35.76
k		59002.10429
c		10078.09333
Wn	rad/sec	7.012019698
Nat Freq	Hert	1.115612334
Ccrit		16828.84727
Zeta		0.598858209
Wd		5.615611538
Damped Nat Freq		0.893443796
C2		250315.3506
tan phy		-0.883194235
phy		-0.723452179
X		2.563525609

step=0.02			
t	r/ro		
	sin term	exp term	total
0	-0.66198	1	0
0.02	-0.5738	0.919446	0.344529
0.04	-0.4784	0.845381	0.660233
0.06	-0.37696	0.777282	0.945863
0.08	-0.27078	0.714669	1.200904
0.1	-0.16118	0.657099	1.425479
0.12	-0.04956	0.604167	1.620237
0.14	0.062692	0.555499	1.786269
0.16	0.174153	0.510751	1.925015
0.18	0.283419	0.469608	2.038188
0.2	0.389114	0.431779	2.127695
0.22	0.489906	0.396998	2.195577
0.24	0.584525	0.365018	2.243952
0.26	0.671778	0.335614	2.274961
0.28	0.750567	0.308579	2.290728
0.3	0.819897	0.283722	2.293327
0.32	0.878896	0.260867	2.284744
0.34	0.926821	0.239853	2.266866
0.36	0.963067	0.220532	2.241451
0.38	0.987177	0.202767	2.210125
0.4	0.998848	0.186433	2.174368

Ini Grad	-17.2265

Figure 11: underdamped steering response analysis

The input parameters and therefore final yaw rate is calculated for a time period of 2 seconds with 0.02 second increment. the final yaw rate calculated in the right most column with respect to time is plotted.

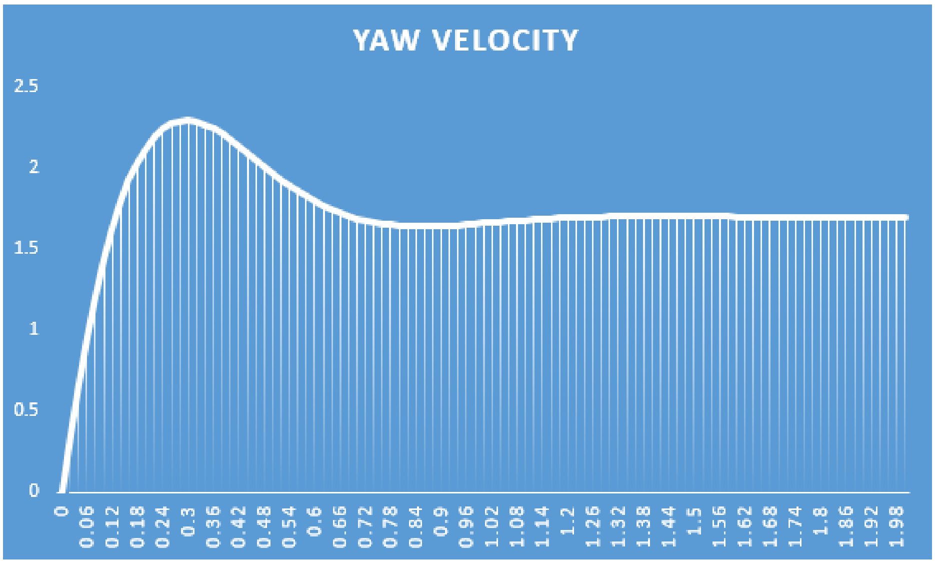


Figure 12: yaw velocity underdamped

The graph above shows the typical characteristics of an underdamped system, oscillating a lot before attaining equilibrium.

5.2 CRITICALLY DAMPED CASE

The spreadsheet is similar to the previous one in terms of input values except for the CoG location.

	A	B	C	D	E	F	G	H	I	J	K	L	M	N
1														
2		Input Parameters				Input Parameters for critically damped Response								
3		Mass	Kg	950		Speed	mph	80		t	r/ro			
4		Polar Moment of Inertia	Kg.m^2	1200		delta	rad	0.4					STEP=0.02	
5		Weight F/Weight R		65/35						0	0		INI GRAD	4.53718
6		CoG-front axle (a)	m	1.236		Derived Parameters for critically damped Response				0.02	0.090743531			
7		Wheelbase (l)	m	2.7		Understeer Gradient	deg/G	8.63456E-05		0.04	0.173461108			
8		Cornering Stiffness f	N/deg	-1066		rdot t=0	rad/sec^2	-25.16384382		0.06	0.248849226			
9		Cornering Stiffness r	N/deg	-900		r t=inf		5.297392351		0.08	0.317544817			
10						Speed	m/s	35.76		0.1	0.380130286			
11		Derived Parameters				k		18901.02405		0.12	0.437138123			
12						c		9678.849383		0.14	0.489055133			
13		CoG-rear axle (b)	m	1.464		Wn	rad/sec	3.96873448		0.16	0.536326302			
14		Cf in N/rad		-61077.291		Natural Frequency	Hert	0.631425656		0.18	0.57935835			
15		Cr in N/rad		-51566.193		Ccrit		9524.962753		0.2	0.618522977			
16						Zeta		1.01615614		0.22	0.654159839			
17		Derivatives				Wd		#NUM!		0.24	0.686579275			
18						Damped Nat Freq		#NUM!		0.26	0.716064804			
19		Yβ		-112643.48		C2		250315.3506		0.28	0.74287541			
20		Yr	/V	1.3750985		tan phy		#NUM!		0.3	0.767247636			
21		Yδ		61077.291		phy		#NUM!		0.32	0.789397501			
22		Nβ		1.3750985		X		#NUM!		0.34	0.809522254			
23		Nr	/V	-203829.15		A		-5.297392351		0.36	0.827801981			
24		Nδ		75491.531		B		4.13990014		0.38	0.844401078			
25										0.4	0.859469593			

Figure 13: critical damped steering response analysis

Yaw rate is calculated in the final column here again. However, two constants are added to the sheet for the calculation of the yaw rate in the case of critical damped. The constants A and B are basically substituted constants required for the calculation of the yaw rate velocity.

The value of the CoG location is changed to 1.236 from the front axle. Changing the location of CoG by moving it to the front and rear helps in changing the steering response of the vehicle. This can be easily done in the spreadsheet and in the bicycle ADAMs model, but could be a little trickier in real life.

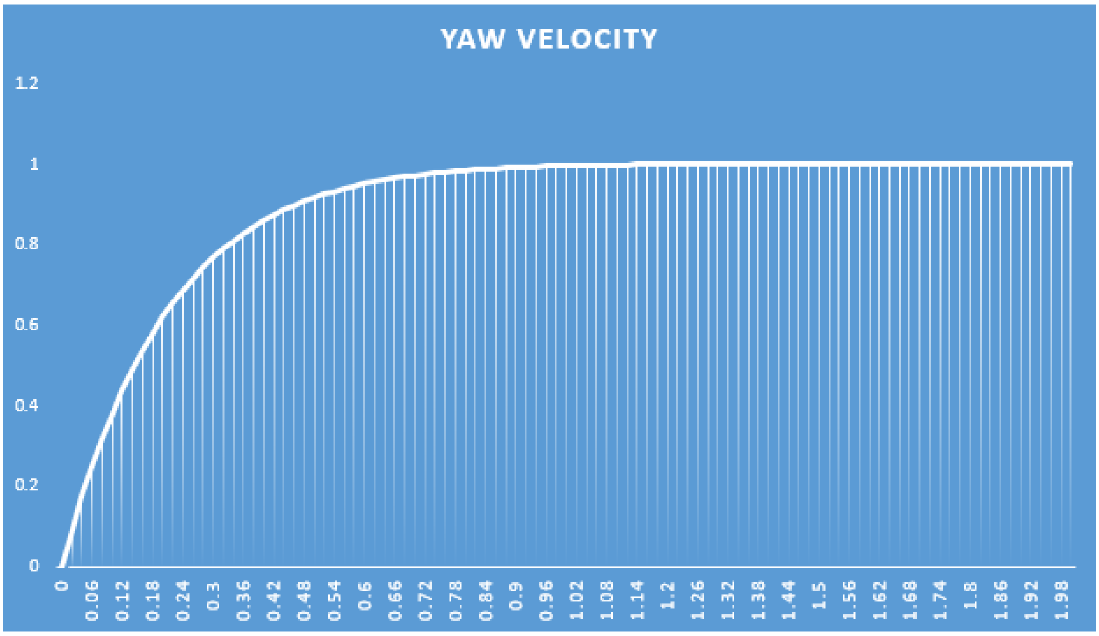


Figure 14: yaw velocity for the critically damped case.

The graph above shows the typical characteristics of a critical damped system, not oscillating at all and attaining equilibrium quickly.

5.3 OVERDAMPED CASE

Overdamped condition also has a similar approach to the previous ones seen in the previous two sections.

Input Parameters		
Mass	Kg	950
Polar Moment of Inertia	Kg.m^2	1200
Weight F/Weight R		65/35
CoG-front axle (a)	m	1.3
Wheelbase (l)	m	2.7
Cornering Stiffness f	N/deg	-1066
Cornering Stiffness r	N/deg	-900

Derived Parameters		
CoG-rear axle (b)	m	1.4
Cf in N/rad		-61077.291
Cr in N/rad		-51566.193

Derivatives		
Yβ		-112643.48
Yr	/V	-7207.8079
Yδ		61077.291
Nβ		-7207.8079
Nr	/V	-204290.36
Nδ		79400.478

Input Parameters for overdamped Response		
Speed	mph	80
delta	rad	0.4

Derived Parameters for overdamped Response		
Understeer Gradient	deg/G	-0.452595025
rdot t=0	rad/sec^2	-26.46682602
r t=inf		8.563761643
Speed	m/s	35.76
k		11691.84109
c		9691.746802
Wn	rad/sec	3.121410083
Natural Frequency	Hert	0.496616344
Ccrit		7491.384199
Zeta		1.293719097
Wd		#NUM!
Damped Nat Freq		#NUM!
C2		250315.3506
tan phy		#NUM!
phy		#NUM!
X		#NUM!
A		-5.865691668
B		-2.698069975

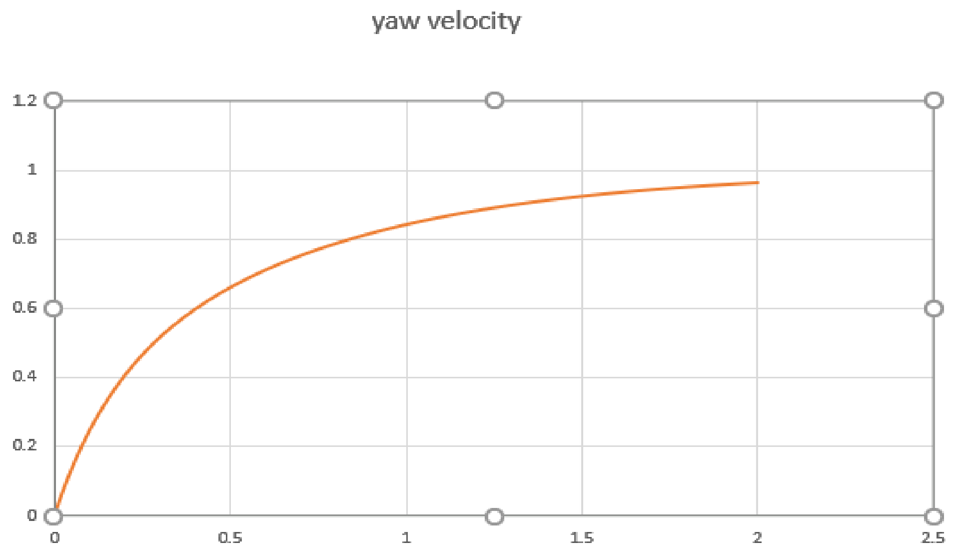
STEP=0.02

t	r/ro yaw rate	Term I	Term II
0	0	-5.86569	-2.69807
0.02	0.504298675	-5.69505	-2.36442
0.04	0.962372054	-5.52937	-2.07202
0.06	1.379467039	-5.36851	-1.81579
0.08	1.76019534	-5.21232	-1.59124
0.1	2.108611626	-5.06069	-1.39446
0.12	2.428282025	-4.91346	-1.22202
0.14	2.722344167	-4.77052	-1.0709
0.16	2.993559805	-4.63173	-0.93847
0.18	3.244360949	-4.49699	-0.82241
0.2	3.476890298	-4.36616	-0.72071
0.22	3.693036685	-4.23914	-0.63158
0.24	3.894466147	-4.11582	-0.55348
0.26	4.082649167	-3.99608	-0.48503
0.28	4.258884546	-3.87982	-0.42505
0.3	4.424320342	-3.76695	-0.37249
0.32	4.579972225	-3.65736	-0.32643
0.34	4.726739567	-3.55096	-0.28606
0.36	4.865419556	-3.44766	-0.25068
0.38	4.996719567	-3.34736	-0.21968
0.4	5.121268015	-3.24998	-0.19252

INI GRAD	25.2149
f	-6.60027
g	-1.47618

Figure 15: step steer analysis- overdamped condition

A similar approach with additional values required for the calculation of yaw velocity in case of overdamped.



Overdamped Behaviour of the system

The graph above shows the typical behaviour of an overdamped system, no oscillations, in fact creating a light negative impact and attaining equilibrium quickly.

5.4 COMPARISON BETWEEN ADAMs AND DERIVATIVES ANALYSIS

This section essentially helps the vehicle dynamist to understand the difference between simulation and derivatives way. All the three cases of underdamped, critically damped and overdamped have been compared with the results obtained from the bicycle model.

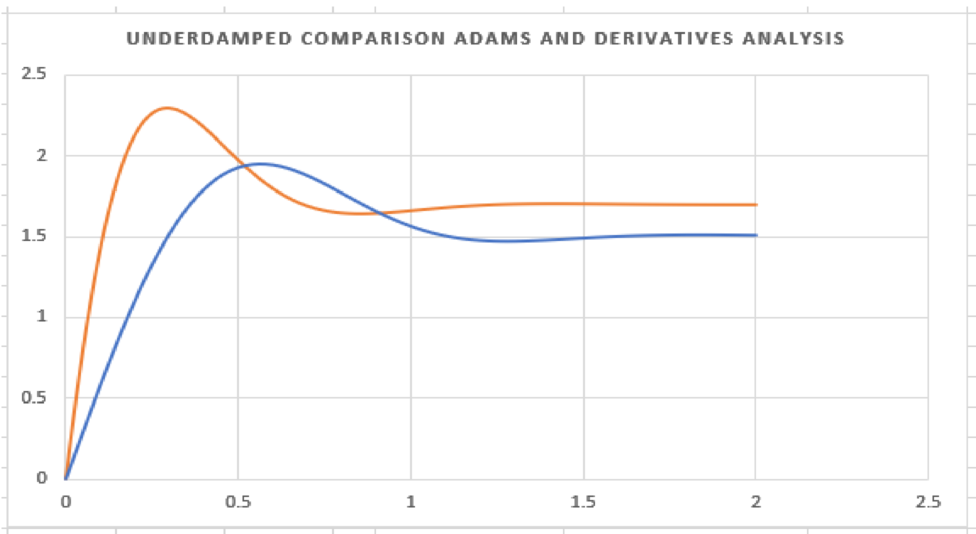
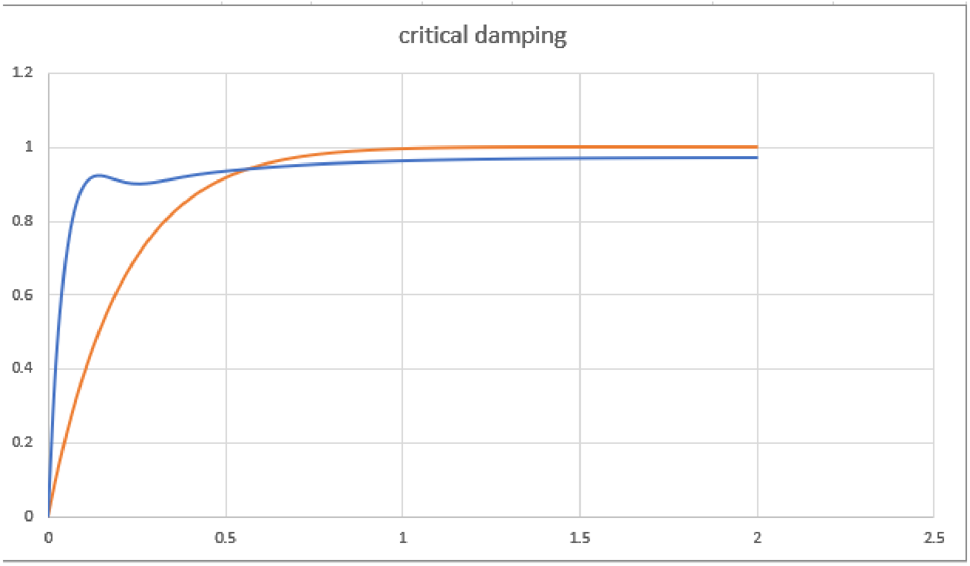
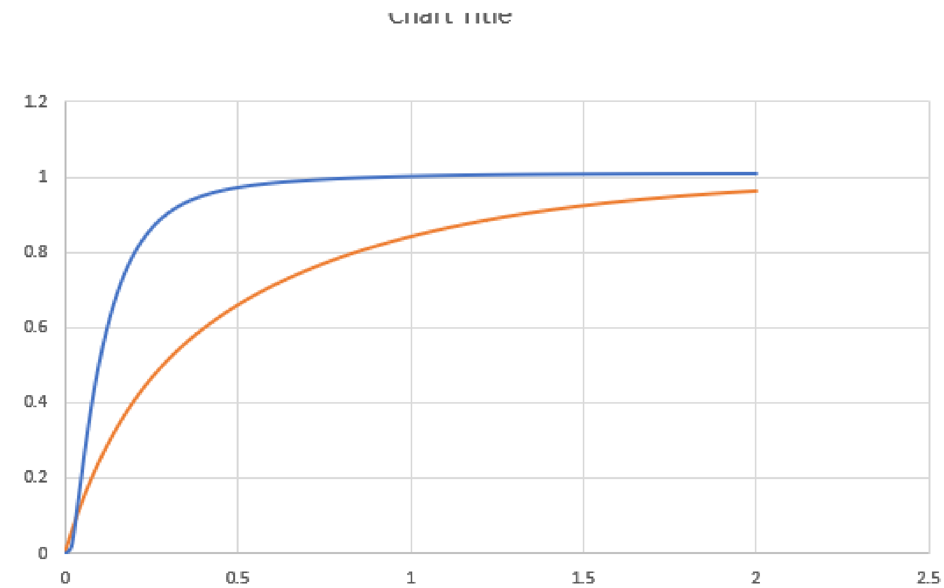


Figure 16: Underdamped case comparison



Comparison of critical damping of bicycle ADAMs model and derivatives analysis



Comparison of Overdamping of Bicycle ADAMs model and Derivatives Analysis

It is found that the differences occur because of the assumption of presence of wheel inertia even without any user input. This causes the bicycle model in ADAMs to be slightly different than the derivatives analysis model.

6. TYRE MODELS

6.1 FIALA MODEL

Available as std in ADAMS, hence very common amongst ADAMS user. Requires only 10 input parameters all of which are physical parameters. They are given in the table below.

Table 5.6 Fiala Tyre Model Input Parameters

R_1 – The unloaded tyre radius (units - length)
R_2 – The tyre carcass radius (units - length)
k_z – The tyre radial stiffness (units - force/length)
C_s – the longitudinal tyre stiffness. This is the slope at the origin of the braking force F_x when plotted against slip ratio (units - force)
$C\alpha$ – lateral tyre stiffness due to slip angle. This is the cornering stiffness or the slope at the origin of the lateral force F_y when plotted against slip angle α (units - force/radians)
$C\gamma$ – lateral tyre stiffness due to camber angle. This is the cornering stiffness or the slope at the origin of the lateral force F_y when plotted against camber angle γ (units - force/radians)
C_r – the rolling resistant moment coefficient which when multiplied by the vertical force F_z produces the rolling resistance moment M_y (units - length)
ζ – the radial damping ratio. The ratio of the tyre damping to critical damping. A value of zero indicates no damping and a value of one indicates critical damping (dimensionless)
μ_0 – the tyre to road coefficient of 'static' friction. This is the y intercept on the friction coefficient versus slip graph, effectively the peak coefficient of friction
μ_1 – the tyre to road coefficient of 'sliding' friction occurring at 100% slip with pure sliding

Figure 17: F tyre input

FIALA model ignores the coefficient of camber angle, lateral stiffness coefficient so it is limited in terms of capabilities. It estimates a parabolic normal pressure distribution on the contact patch with a rectangular shape, R_1, R_2, K_z and zeta are the four parameters used to calculate vertical load. Cy is not used at all, because of absence of Camber Angle. The longitudinal force the lateral force and the aligning moment is all controlled by Just five parameters, Cs, C_α, C_r, μ_0, μ_1.

The effective friction coefficient m is determined as a function of the comprehensive slip ratio SL_a. The comprehensive slip ratio SL_a is taken to be the resultant of a longitudinal slip coefficient S_x and a lateral slip coefficient Sa.

Despite the advantage of a simple parameter. The main limitations of the model include:

1. The model cannot represent combined cornering and braking or cornering and driving.

2. Lateral force and aligning moment resulting from camber angle are not modelled.

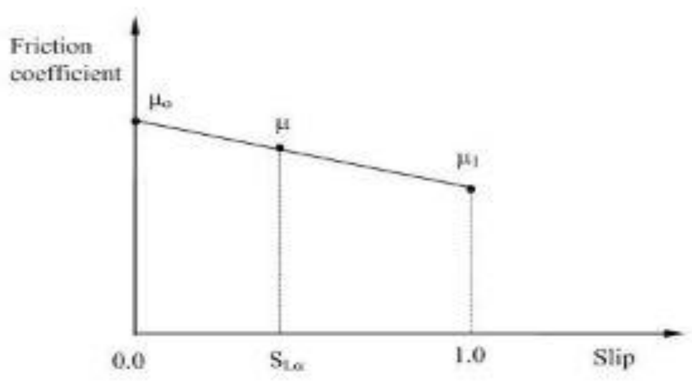


Figure 28: Linear tyre to road Friction model

3. The variation in tire stiffness at zero slip angle with tyre load is not considered.

Therefore, I conclude that FIALA model has no practical use in the ground vehicle modelling whatsoever as the calculations would, though with great efforts, more often end up being wrong.

ADVANTAGES	DISADVANTAGES
• It uses only 10 input parameters. • It is a quick way to model simple and modest models.	• Camber angle is not considered on lateral force and aligning moment. • Combined slip cannot be considered as it does not account combined cornering, acceleration and braking

6.2 RMOD-K MODEL

RMOD-K is a steady-state (combined) slip that gives a detailed finite element description of the actual tyre structure. This model allows the calculation of the tyre response over a wide range of excitation frequencies and driving manoeuvres.
Moreover, this is a model that covers at least the rigid body modes of vibration of the tyre belt. It can be used to optimise simulation performance by switching between models of different complexity with respect to the road surface Curvature.

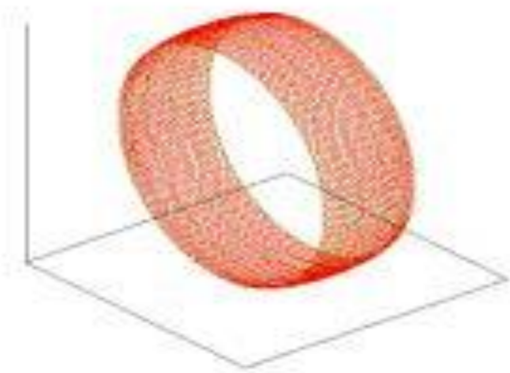


Figure 19: RMOD-K Finite element structure representation

This system of tyre models allows the calculation of tyre response over a wide range of excitation frequencies and driving manoeuvres. At low to medium frequency excitations concerning inputs from the vehicle or the road, the rigid belt model RMOD-K 7 RB covers the frequency area to 100 Hz.

The belt is modelled by one or more layers that interact with each other, and the road contact is realised through an additional sensor layer. In sensor points the normal and frictional forces are calculated.

ADVANTAGES	DISADVANATAGES
• Road unevenness is taken in mind. It can cover situations where the forward speed vanishes. • Moving road surfaces can be handled.	• High cost of the software, instrumentation, etc. • Applications are limited to smooth roads.

6.3 HARTY TYRE MODEL

Harty tyre model is more simplified than Pacejka's tyre model but has better acting features than FIALA model discussed in previous sections. Harty tyre model can be found as a default tyre model in the multi-simulation software ADAMs as TIRSUB since 1996.

It has been found that measured tyre data for both lateral and longitudinal that could be generated easily with just a single parameter 'A', also called Curvature Factor.

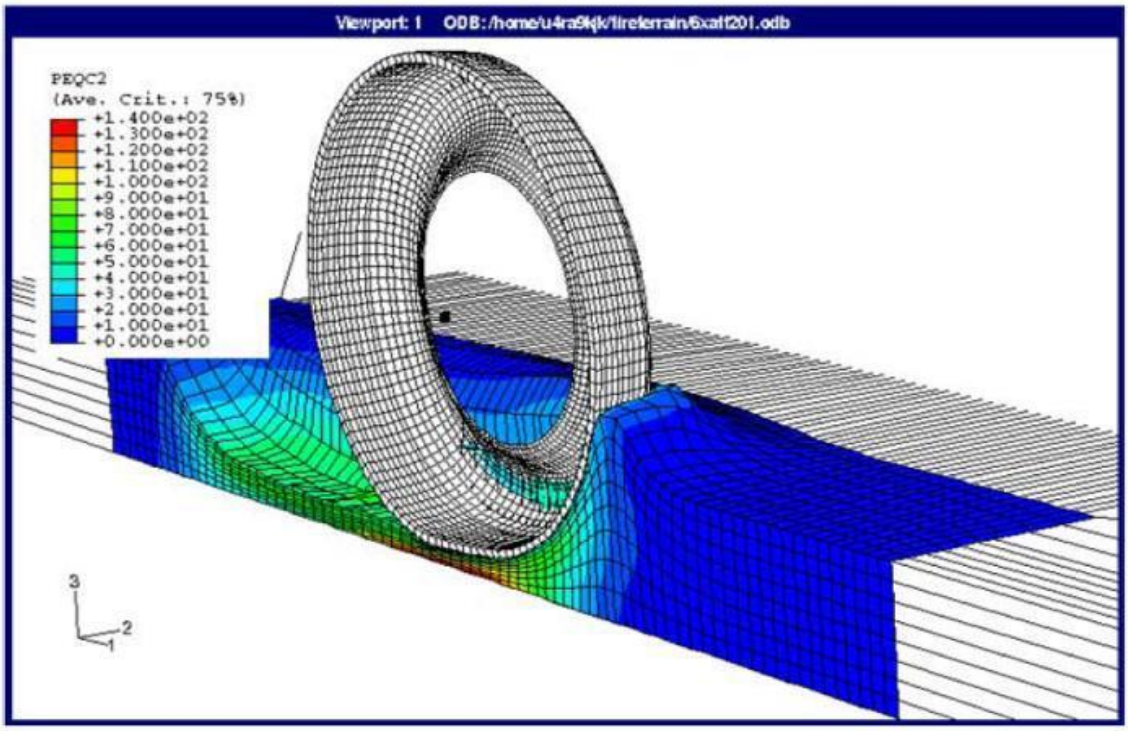


Figure 20: tyre simulation in OPTIMUM-T using HARTY tyre

The input values in case of HARTY tyre model is less complicated than the Pacejka's model. Friction coefficient treatment is retained form the FIALA model. Slip Angle and Slip Ratio is declared constant as the peak side force and the peak Longitudinal Force generated respectively do not vary much.

In multi simulation software ADAMs, the coding for HARTY model is similar to the FIALA model and half of the coding is reused including some of the input parameters.

ADVANTAGES	DISADVANATAGES
• Model simpler than Pacejka's model. • Uses common coding to FIALA but is more complex and Robust.	• High cost of the software, instrumentation, etc. • Formula based simulation can be sometimes off from reality.

7.0 REFERENCES

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