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Anti-Roll Bar Design and Material Selection for the Torsion Member for A Formula Student Vehicle

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***Abstract*- The following study will be dealing with the design procedure, constraints, and limitations that were dealt with during the design of the Anti-roll bar system of a formula student vehicle. Also, simulations with proper forces for torsion on the torsional member to select the best possible material for the torsion member was done.**

Keywords- FSAE, Anti-roll bar, Roll, Formula student

INTRODUCTION –

Anti-roll bar sometime also known as a stabilizer bar or sway bar is used for the purpose of controlling the roll motion of the vehicle. It provides better stability during corning. Roll motion referred as to the motion in the lateral direction on the Y-axis during which the vehicle rolls around the central axis of the vehicle.

Image Courtesy- RHD-JAPAN

Anti- Rollbar increases overall roll stiffness of the vehicle, thus providing a number of advantages to the vehicle and its performance.

One of the major advantages is that it provides better traction during cornering the vehicle.

METHOD USED –

The calculations were done in the following syntax -
RIDE RATES - ROLL ANALYSIS - ARB DIMENSIONS AND MATERIAL SELECTION

Ride rates were calculated by first determining the desired Natural frequencies of the front as well as rear but separately. These values were chosen using the general values which have validated after years of experimentation and observation.

Natural frequency for the front suspension = 2 Hz
Natural frequency for the rear suspension = 2.5Hz

The reason behind aiming for higher natural frequency in the rear is to compensate for the response delay, since its always the front suspension which has to act first.

Then, the ride rates were calculated for the front and rear individually using the following expression-

$$K_{RF} = \frac{4\pi^2 \omega_F^2 (W_{SF}/2)}{386.4}$$

Here, Krf= Ride Rate front wF=Natural Frequency Front Wsf= Sprung mass front

<u>Roll Analysis</u> - Here the first which needs to be calculated is the COG height of the sprung mass.
which can be calculated using the following expression -
Here, hs= sprung COG ht., h=COG ht., Wuf=Unsprung

$$h_S = \frac{W_T\,h - W_{UF}\,RL_F - W_{UR}\,RL_R}{W_S}$$

wt. Front,
RLf= Tire loaded radius(Static) front, Wur=Unsprung wt. Rear
RLr=Tire loaded radius(Static) Rear
Sprung Wt. distribution = Sprung wt front/Total sprung mass.
Rolling Moment (lever Arm) =
(Sprung mass cog ht.)-[Front Roll Centre ht+(Rear Roll Centre ht.-Front Roll Centre ht.)(1-sprung mass wt. distribution)]
Rolling moment per g lateral acceleration =

$$\frac{M_\phi}{A_y} = \frac{h_{RM} W_S}{12}$$

Here, Hrm= Rolling moment (lever arm) ----------------- Calculated above

Now, dividing this value with the desired roll gradient will give us the Roll Rate required(KΦ).

Roll gradient should be taken between 1.5 - 3deg/g for an fs vehicle.

Now calculate available Front spring roll rate by using the following expression -

$$K_{\phi SF} = \frac{K_{RF}\,T_F^2}{1375}$$

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	A	B	C	D	G	H
1						
2						
3	RIDE AND ROLL RATES					
4						ReqD I/P
5	Roll Gradient(table 16.1)	1.2				O/P
6	ϕ	0.8				
7	Motion Ratio (sheet 1)	0.7				
8	Wf(NF)	2	Wr(NF)	2.5		
9	Wsf(sprung wt.)	220.462	Wsr	343.921		
10						
11	Krf(ride rate front)	45.04904865	Krr	109.8071		
12						
13						
14	Kw(wheel rate)(sheet1)	47.1967				
15						

Table 16.1 Typical Roll Gradients, Based on Ref. 91

Very Soft—Economy and basic family transportation, both domestic and import, pre-1975.	8.5 deg/g
Soft—Basic family transportation, domestic and import, after 1975.	7.5
Semi-Soft—Contemporary middle-market sedans, domestic and import.	7.0
Semi-Firm—Imported sport sedans.	6.0
Firm—Domestic sport sedans.	5.0
Very Firm—High-performance domestic, such as Camaro Z-28 and Firebird Trans Am.	4.2
Extremely Firm—Contemporary very-high-performance sports, such as Corvette, and street cars extensively modified to increase roll stiffness.	3.0
Hard—Racing cars only.	1.5
Active Suspension—Servo-controlled roll stiffness. Roll-in, zero-roll, and roll-out all possible.	—

Here, KΦsf = Front spring roll rate

Krf = Ride rate front (calculated during ride analysis)

Tf = Track width front

Similarly, by using adequate value, the available rear spring roll rate(KΦsr) can be determined.

Adding both the front and rear roll rates will give us the total available roll rate.

So, arb must provide = Required roll rate - available roll rate

Now, to size the front and rear bars, we need to calculate this requirement based on the weight distribution of the vehicle.

Multiply this with %load transfer X 100 to get front/rear load transfer.

Front Roll Stiffness (Kϕf) =

$$\frac{FLT}{A_y} = \frac{12(K_{\phi F})\phi}{T_F} + \frac{W_{SF}Z_F}{T_F} + \frac{W_{UF}RL_F}{T_F}$$

Here, ϕ= Roll Gain (0.5 - 1 for non-aero fs vehicles)

From here, it can be calculated that the rear arb must provide

16										
17			ROLL ANALYSIS							
18										
19	Sprung mass CG ht.				Sprung mass wt dist.		rolling moment lever arm			
20										
21	Wt(total wt)	617.294			Wsf(sprung wt F)	220.462	hs	9.91634		
22	h(cg ht)	9.84252					Zf	2.16575		
23	Wuf(unsrpung mass)	26.4555			as	0.390624806	Zr	2.27992		
24	RLf(tire raduis loaded static)	9.05512			%	39.06248062				
25	Wur	26.4555					hrm	7.68102		
26	RLr	9.05512								
27	total sprung wt(Ws)	564.383								
28	hs	9.91633888								
29										
30	Rolling moment per g lat. Acc				REQd roll rate		Front /Rear Spring roll rate			
31										
32	hrm	7.6810165			Mϕ/Ay	361.2529281	Krf	45.049	Krr	109.807
33	Ws	564.383			Roll Gradient	1	Tf(t/w front)	46.4567	Tr(t/w rear)	42.1575
34	Mϕ/Ay	361.252928								
35					Kϕ	361.2529281	Kϕsf	70.7098	Kϕsr	141.931
36										
37										
38	Total roll rate by springs	212.640753			Total Load Transfer		front Reqd Stifness			
39										
40	ARB roll rate(Karb)	148.612176			Wt	617.294	FLT/Ay	127.006		
41					h	9.84252	Tf	46.4567		
42					Tavg	21.07874015	Wsf	220.462		
43							Zf	2.16575		
44					TLT/Ay	288.2396432	Wuf	26.4555		
45							RLf	9.05512		
46					Front load transfer	127.0055369				
47										
48							Kϕf	345.549		
49										
50	REQD BAR STIFFNESS REAR	-126.22698	lb-ft/deg	in N-m/deg						
51	diff	274.839158								
52	REQD BAR STIFFNESS FRONT	560.896242	lb-ft/deg	in N-m/deg	760.4731927					

Sheet1 Sheet2

Here, TLT/Ay = Total load transfer during lateral acceleration.

Tave= Average of Front and Rear track width

the difference between the total, rear, and that provided by the rear springs.

Kϕ(arb)rear = Kϕ - Kϕf - Kϕsr

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Divide this by the Linkage ratio(rear) to get the final requirement.
For the front,
Kϕ(arb)front = (Kϕf - Kϕsf)/linkage ratio(front)

RESULTS AND DISCUSSION FOR THE METHOD

The whole calculation was done by preparing a spreadsheet to facilitate the calculations. For which the results are shown as followed -

It can be seen that the required stiffness for the rear comes out to be negative, which symbolizes that the rear suspension is stiff enough and does not require an arb setup.

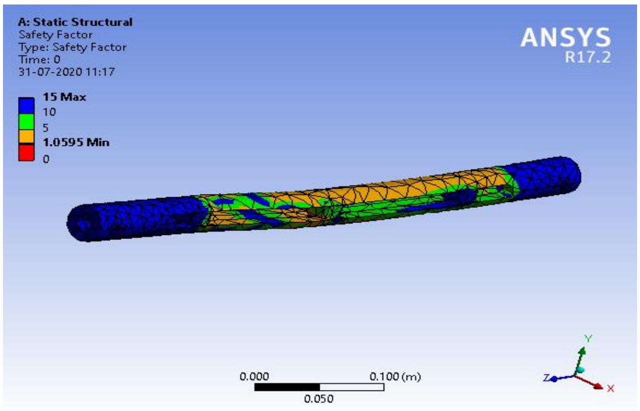
MATERIAL SELECTION

The following materials were considered-
Mild steel - AISI 1020 80 GPa shear modulus
Aluminum - AL-7075 T6 26.9 GPa shear modulus
Magnesium alloy - AZ31B 17 GPa shear modulus

Although magnesium alloy provides great weight reduction advantage it is expensive and also for the same thickness, withstands lower stress than aluminum.

After running simulations aluminum was considered as the optimal choice as it provided enough durability, is lightweight, and cheaper than magnesium alloy.

The size was determined by comparing the required area moment of inertial to the area moment of inertia of the arb system torsion bar with desired dimensions.
Required area moment of inertia = [(Required Roll stiffness X Length of tube)/Shear Modulus]
Area moment of inertia for cylindrical tube = $[\pi(D^4 - d^4)]/32$
The following simulation was done by applying 650N on both ends in torsion that sums up to 1300N in total.



Reqd bar stiffness Front	760.473193
G	2.6E+10
L	0.573
i	1.676E-08
MATERIAL	7075 T6
i	3.7276E-08
D	0.035
d	0.015

With dimensions OD= 35mm
ID= 15mm

CONCLUSION -

Anti-roll bar torsion member with outer diameter of 35mm and inner diameter 15mm was designed to provide roll stiffness of 760.4731927 Nm/mm.

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Softwares Used - CAD Modelling - Solidworks 2020
Simulation - Ansys R17.2

REFERENCES

Racecar vehicle dynamics, William F. Millikan; Douglas Millikan