OPEN ACCESS

International
Journal
Of Modern Engineering Research (IJMER)

Finite Element Analysis of Anti-Roll Bar to Optimize the Stiffness of the Anti-Roll Bar and the Body Roll

Bankar Harshal[1], Kharade Rushikesh[2], P. Baskar[3]
[1, 2](M.Tech Automotive Students VIT University, Vellore, Tamil Nadu)
[3] (Professor SMBS VIT University, Vellore, Tamil Nadu)

Abstract: *The objective of this paper is to analyze the main geometric parameters which affecting the stiffness of anti-roll bar. Further these parameters are also affecting the body roll angle. By the optimization of these geometric parameters we can able to increase the stiffness of bar and which will help to reduce the body roll angle. To calculate the stiffness of anti-roll bar Finite Element software ANSYS is used. The deflection for the change in internal angle, arm length, moment of inertia, distance between bushes found by static analysis. To calculate the body roll angle equation used from the literature survey, however they haven't taken all the suspension characteristics in the calculation of moment caused by the suspended and non-suspended masses. The equilibrium condition is considered between the moments of the force acting on the suspended and non-suspended masses and moments of reaction of the springs and anti-roll bar used in suspensions. The comparison of different anti-roll bar is based on the basis of stiffness per weight. The anti-roll bar which having more ratio of stiffness per weight can be used in the vehicle. As it will improve the stiffness of bar with small increase in weight, which will result in the improving roll stability of the vehicle.*
Keywords: *moments, non suspended, stability, stiffness, suspended*

I. Introduction

Automobile industry focus on producing element which give handling and performance of the vehicle better than today's vehicle but such element should not produce the extra cost and also it should be improve the comfort level of the vehicle that is to the passenger.

Anti roll bar is one of the inventions in the automobile industry which is also called as sway bar or stabilizer bar. Structure of such anti roll bar are U shaped bar which connect two wheel that is left and right wheel and bar is fixed to the chassis of the vehicle by bush.Anti roll bar may be solid or hollow tube. The main function of anti roll bar is reducing body roll motion when the vehicle is at the cornering condition. Body roll condion occurs due to the load transfer and changes takes place in the camber of vehicle which directly affect the steering behavior of the vehicle and vehicle loses its stability therefore to eliminate the roll effect in case of under steer and over steer anti roll bar is used. Anti roll bar give comfort in driving condion and safety in case of such roll situation.

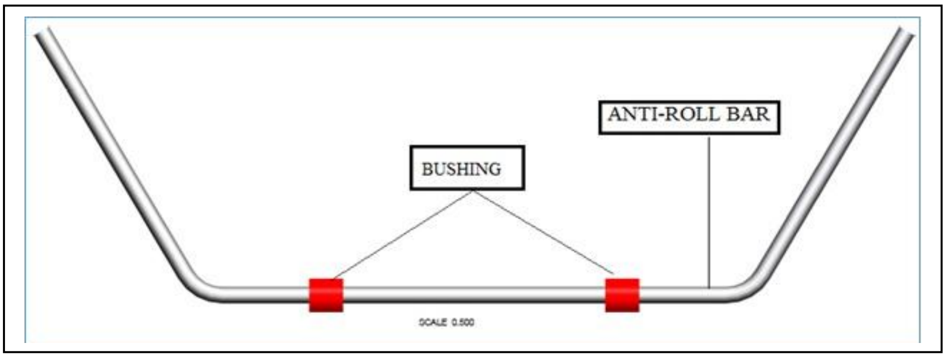


Figure-1 anti-roll bar with bus

When both the suspension affected simultaneously then the effect of the anti roll bar is eliminated. when one of the wheel moves opposite of the other then anti roll bar acts like the torsion spring and it will provide the torque such that it will oppose the motion of the vehicle so that tilting motion of the vehicle will balanced that is neutralize the vehicle[1].Another important advantage of the anti roll bar is that it allow to use the less stiff spring therefore it can absorb the uneven road shocks which can give ultimately comfort to the passengers[2].In case of anti roll bar, SAE (society of automotive engineers) will provide the information of process of manufacturing of anti roll bar, equation to find out the stiffness of anti roll bar where load is applied at the end of the bar but only simple shapes of anti roll bar is used to apply these equation[3].To develop anti roll bar different design technique should be used

The effect of anti-roll bar when used at the front suspension is studied by[4].In this paper vehicle used for study is bus and considered effect without or with anti-roll bar and finally given equation which gives information of roll angle and rigidity of anti-roll bar.

The effect of different variable which affect the roll motion of the body and stiffness of anti-roll bar is important factor to study. When the anti-roll bar is used in the vehicle then reduces the body roll by 48.4 % which gives stability to the vehicle [5].In this paper author also used finite element method to find the stiffness of the anti-roll bar and also show that lowest stiffness to the weight ratio is achieved by using the shortest length of side length of the bar.

The main objective of this paper is to analyze the anti-roll bar for stiffness and body roll of the vehicle by varying the various geometric condion of the anti-roll bar. In this paper we also find the variation which will give the lowest value of stiffness to the weight ratio.

II. METHODOLOGY

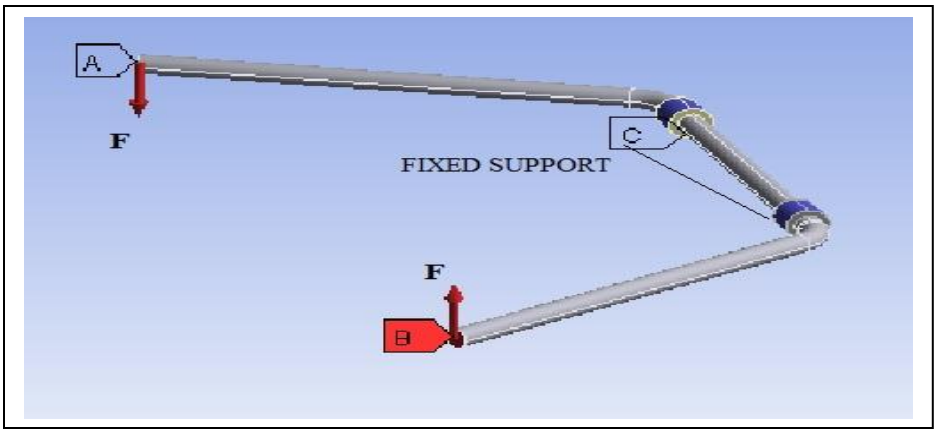
To achieve target of the paper we have done different geometrical variation in anti-roll bar that is we have variation in length of bar, variation in the distance of bush, variation in the angle between two arms of bar, varying the moments of inertia of the bar and by using the different cross section at the end of ant-roll bar. For analysis we have used simple geometry of anti-roll bar and to calculate the stiffness of the anti-roll bar we have used the finite element method that is by using software. In this paper we consider the use of anti-roll bar at the rear suspension of the vehicle.

1. Stiffness of Bar

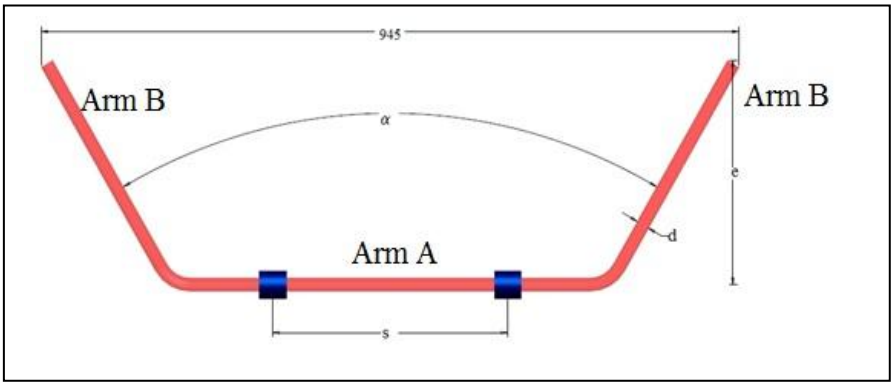
Stiffness of the anti-roll is given by the equation,

$$k_e = F/\Delta x \tag{1}$$

To calculate the roll stiffness divide the applied force by summation of absolute value of the deflection at the bar end which is find in the CAD software that is we have used the ansys workbench 14 for simulation and analysis of anti-roll bar. We have considered only static analysis means we have eliminated the effect of the dynamic condion. We have applied load at the end of ant-roll bar which is of 1KN which is of linear type and bush is fixed. In this paper material considered is steel for anti-roll bar. Properties of steel used are density, modulus of elasticity and Poisson’s ratio. Load applied at one end is 1 KN in one direction but for other end we have applied the same load in opposite direction.



In this paper we have considered the geometrical variation of anti-roll bar and considered the response in terms of deformation by applying the load which having magnitude of 1KN after applying this load deflection is found in software. We have considered the absolute value of deflection only. The main purpose to find the deflection of the anti roll bar is found out the rigidity of the anti roll bar. In this analysis we have draw simple model of anti-roll bar which mean that obtained result will not be result in the reality that is we can get the variation in the stiffness of bar and roll motion of the vehicle.



For analyze anti-roll bar we have the modification in the variation of ‘e’, variation between angle of two arms that is angle alpha, varying the moment of inertia of bar A and B by considering the different inner and outer diameter, varying the distance between arms that is ‘S’ and by considering the rectangular section at the one end of anti roll bar. In this work we have not changed the distance between the two arm ends.

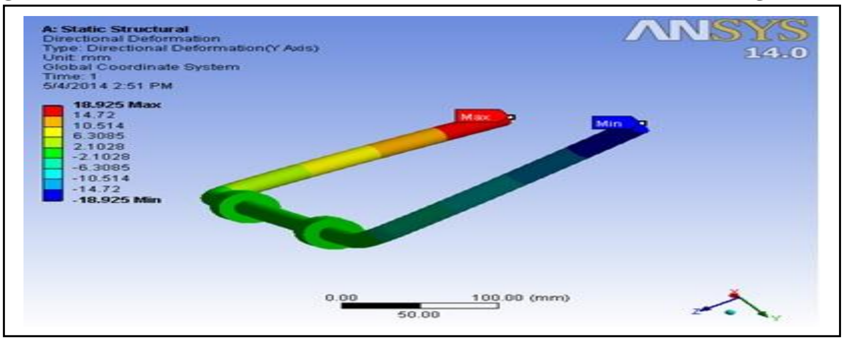
1.1 Variation in the distance between horizontal arm and end of bar that is ‘e’ in the vertical direction.
In first part of the analysis we have changed the distance ‘e’ which is will show in the TABLE

Table-1

Anti-roll bar	1	2	3	4	5
e (mm)	220	270	320	370	420

In first case we have considered the following parameters
1. α (alpha) = 60^0.
2. S = 350 mm.
3. A and B: solid geometry having an outer diameter of 20 mm, A_o = 314.2 mm^2, I = 7854 mm^4 e, J =15708 mm^4.
4. Different weights of the anti-roll bar

Whereas I is moment of inertia, A_o is the outer diameter of the bar and J is the polar moment of inertia. The profile of the anti roll bar, moment of inertia, area of cross section is same in case of the analysis of first case. We can get the weight increase of the anti-roll bar which is due to the increase in the length of bar.



Above Fig shows that when first condion of the ‘e’ variation where we have applied 1 KN load and maximum deflection is noted. Maximum deflection observed is 18.925 mm on one side of bar. Total deflection will be summation of the both side deflection where absolute value should be taken.

1.2 Variation in the moments of inertia of A and B
Second analysis mainly to observe the change in the stiffness of bar but weight of anti-roll bar is not considered that we have not changed the cross section of the bar. Polar moment of inertia, moment of inertia of A and B changed.

Table-2

Anti-roll bar	6	7	8	9	10
$Ø_o$ (mm) of arm A & B	21	22	23	24	25
$Ø_i$ (mm) of arm A & B	6.4	9.2	11.4	13.3	15
I (mm^4) of arm A & B	9464	11147	12907	14750	16689
J (mm^4) of arm A & B	18928	22294	25815	29500	33379

In second case we have considered the following parameters
1 .e=420 mm
2. $\alpha = 60^0$
3. S=350 mm
4. A and B having area equal to 314.2 mm^2
5. Weight 40.14 N

1.3 Variation of the distance between the bush 'S'
In this section we have done variation in the distanced between the bush which is fixed to the chassis and give support to the anti-roll bar. The following TABLE gives information about this variation.

Table-3

Anti-roll bar	11	12	13	14	15
s (mm)	100	150	200	250	300

For third case we have considered the following parameter
1. e =420 mm
2. $\alpha = 60^0$
3. A and B section having outer diameter equal to 20 mm and area is 314.2 $mm^2$and moment of inertia I =7854 mm^4, polar moment of inertia equal to J=15708 mm^4.
5. Weight is equal to 40.14 N

1.4. Variation in angle between the two arms 'α'
In case of forth study we have done variation in angle between A and B.

Table-4

Anti-roll bar	16	17	18	19	20
α (^{0})	0	12	24	36	48

For forth we have considered the following parameter
1. e =420 mm
2. S =350 mm
3. A and B section having outer diameter equal to 20 mm and area is 314.2 $mm^2$and moment of inertia I =7854 mm^4, polar moment of inertia equal to J=15708 mm^4.
4. Different weights.

1.5. Variation in the profile of 'B' with changes in moment of inertia.
In this section we have changed the cross section of the B section from tubular section to the rectangular section this gives rigidity to the part.

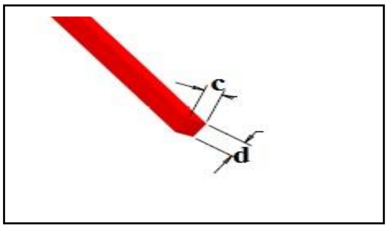


Figure-5 Changes made in profile B

Above Fig shows that rectangular cross section of profile B with variation made in the dimension of C and D but we have taken the same profile of A.
Following TABLE shows variation done on the cross section of profile B.

Table-5

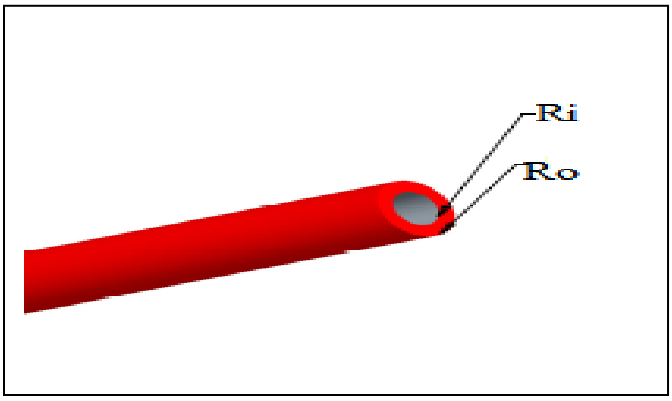
Anti-roll bar	21	22	23	24	25
Height *c* (mm)	20	22.5	25	27.5	30
Width *d* (mm)	20	17.8	16	14.5	13.3
***I* (Stretch B) (mm4)**	13333	16896	20833	25129	29925

For fifth study we have considered the following parameters
1. e =420 mm
2. α =60
3. S =350 mm
4. Profile B with area equal to 400 mm^2
5. Profile A having an outer diameter of 20 mm, Ao = 314.2 mm2, I = 7854 mm4 e, J =15708 mm4.
6. Weight 46.19 N

1.6 Variation in the profile of A without changes in the moment of inertia of section.
In this section we done variation in tubular section of the A but we have maintained the moment of inertia.
Following TABLE gives information about variation in the profile A.

Table-6

Anti-roll bar	26	27	28	29	30
$Ø_o$ (mm) of arm A	21	22	23	24	25
$Ø_i$ (mm) of arm A	13.6	16.5	18.6	20.4	21.9
A (mm^2) of arm A	285.6	363	427.8	489.6	547.5



For sixth we have considered the following parameters where moment of inertia is maintained constant
1. e =420 mm
2. α =60^0
3. S=350 mm
4. Profile A having tubular section with the I= 7854 mm^4 and J=15708 mm^4
5. Profile B having outer diameter is equal 25 mm and 19174.8 mm^4
6. Different weight is considered

2. Roll Angle Calculation

In this section roll angle calculation is done based on method given by [6].In this method roll angle is actually tire grip coefficient between the wheel and road track. In this section we have taken μ_s is equal to the 0.7.

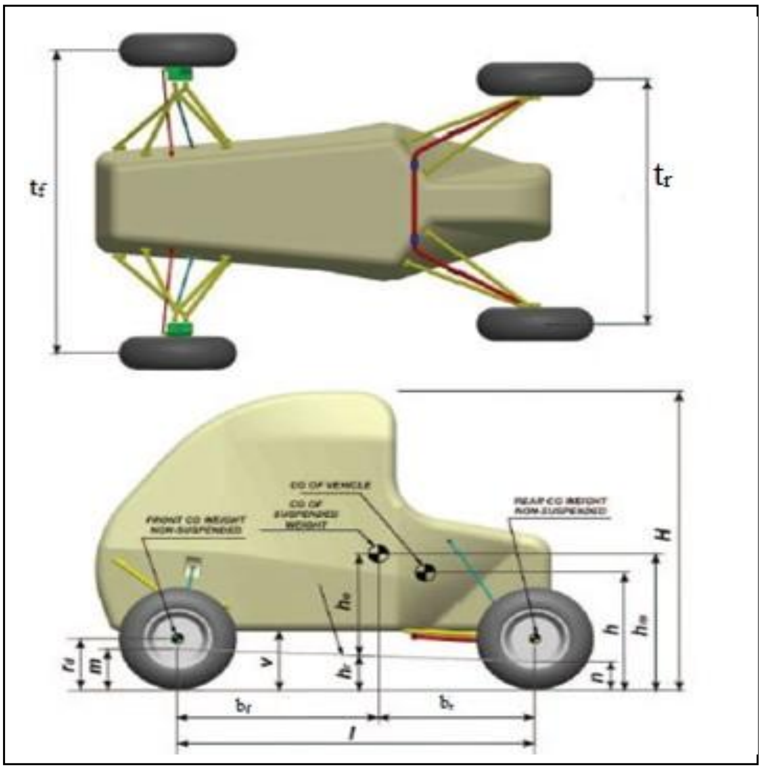


Figure-7 gives dimension of the vehicle which is to be considered in this paper [5].

Following TABLE gives information about dimension of the vehicle [5], Table-7 [5]

v	275 mm	Free vain
H	1400 mm	Total height
l	1450 mm	Distance between axles
h	550 mm	Height of center of gravity (CG)
r_d	240 mm	Dynamic radius of the tire
t_f	1420 mm	Front axle width
t_r	1150 mm	Rear axle width

In cornering situation the distribution of the weight is important. Following TABLE gives weight distribution Table-8 [5]

G	2452.5 N	Total vehicle weight with driver
		% Of the weight on the front axle
G_f	1103.625 N	Weight on front axle
Wn_f	274.68 N	Weight of the non-suspended masses on front axle
W_f	828.945 N	Weight of the suspended masses on the front axle
G_r	1348.875 N	Weight on the rear axle
Wn_r	274.68 N	Weight of the non-suspended masses on rear axle
W_r	1074.195 N	Weight of suspended masses on rear axle
W	1903.14 N	Weight of suspended masses
R_{0f}	1103.625 N	Reaction of the front wheels on the ground when the vehicle is stationary
R_{0r}	1348.875 N	Reaction of the rear wheels on the ground when the vehicle is stationary

Calculation of the b_f and b_r is done by following formulas,

$$b_f = \frac{R_{or} - W_{nr}}{W} l \tag{2}$$

$$b_r = \frac{R_{of} - W_{nf}}{W} l \tag{3}$$

Location of the center of gravity of the mass which is to be suspended mass gives the tilt of the vehicle which is due to moment produced at the axis of the vehicle. To calculate this it is considered that it will act the location of center of the wheel that is W_{nr} and W_{nf}.[6].Following formulas gives height of center of gravity of the suspended mass

$$h_m = \frac{\lfloor Gh - (W_{nf} + W_{nr}) r_d \rfloor}{W} \tag{4}$$

Following TABLE gives information of center of gravity of the suspended mass based calculation done by using the equation (4).

Table-9 [5]

b_f	818.4 mm	distance of the CG of the suspended masses until the front axle
b_r	631.6 mm	distance of the CG of the suspended masses until the rear axle
h_m	639.5 mm	height of the CG of the suspended masses

Wheel will rotate at the instantaneous center of rotation which is known as reaction point and that of suspension system considered as roll center [6]. Value of that corresponding to front and rear we have taken from the paper [5].The Following TABLE gives information about front and rear point.

Table-10 [5]

m	192.1 mm	Height of the front roll center
P_f	-731.9 mm	Height of the front reaction point
n	133.7 mm	Height of the rear roll center
P_r	275.0 mm	Height of the rear reaction point

The distance between the c.g. of the suspended mass and roll axis of the vehicle is given by following equation (5) [6].

$$h_o = h_m - \frac{(nb_f + mb_r)}{l} \tag{5}$$

The calculation of the h_o is given in the following TABLE

Table-11[5]

h_o	480.3 mm	Distance of the center of gravity of the suspended mass to the roll axis
h_r	159.1 mm	Height of the roll axis to the ground

While taking the turn momentum is produced due to the centrifugal force of the suspended mass also load transfer takes place from inner wheel to the outer wheel which gives the inclination of the vehicle [6]. When extreme condion of the transverse arm is considered then spring constant is given by function of the stiffness of the spring and u position which is attached to the arm length of v. the following equation (6) gives relation between them[6]

$$K = k\left(\frac{u}{v}\right)^2 \tag{6}$$

The value of the stiffness at the front suspension is given by Following TABLE

Table-12 [5]

k_f	15 KN/m	Front spring stiffness
u_f	250 mm	Spring fixing position on the front arm
v_f	361 mm	Front arm length
K_f	7.2 KN/m	Spring constant in the extreme of the front arm

The value of the stiffness at the rear suspension is given by following TABLE.

Table-13 [5]

k_r	10 KN/m	rear spring stiffness
u_r	485 mm	Spring fixing position on the rear arm
v_r	596 mm	rear arm length
K_r	6.6 KN/m	Spring constant in the extreme of the rear arm

The spring constant effectively of the anti roll bar when the transverse arm is at the extreme condion is given by following equation [6]

$$K_E = k_e \left(\frac{u}{v}\right)^2 \tag{7}$$

We have calculated only K_{EI} and K_{EII} because we have considered the anti-roll bar is only at the rear suspension. Value of the u and v gives the spring constant in the extreme condition of the rear arm.
To calculate the roll angle we have considered the moment between the moment of the stabilizer as well as moment of spring which is balanced by force of the suspended mass and unsuspended mass [6].
Momentum caused due to the suspended mass is given by following equation [6]

$$M_{RO} = \mu_s W h_O \tag{8}$$

Momentum caused due to the non suspended mass of the front axle is given by following equation [6]

$$M_{R1} = \mu_s W_{nf} r_d \left(1 - \frac{m}{P_f}\right) \tag{9}$$

Momentum caused due to the non suspended mass of the rear axle is given by following equation [6] ,

$$M_{R2} = \mu_s W_{nr} r_d \left(1 - \frac{m}{P_r}\right) \tag{10}$$

Roll angle of the vehicle body is given by following equation [6]

$$\varphi = \frac{M_{RO} + M_{R1} + M_{R2}}{\left(\frac{t_f^2}{2}\right) K_f + \left(\frac{t_r^2}{2}\right) K_r + \left(\frac{t_f^2}{2}\right) K_{Ef} + \left(\frac{t_r^2}{2}\right) K_{Er}} \tag{11}$$

'f' is the lateral displacement at the top most point of the vehicle when the vehicle occurs with the body roll.

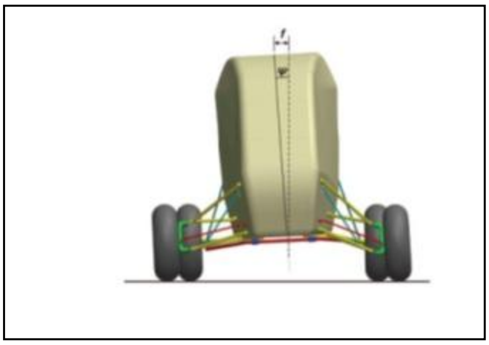


Figure-8 lateral displacement at top most point of the vehicle [5]

III. Result

Roll angle of the vehicle is calculated without considering the anti-roll bar. For calculation purpose we have used above procedure. Spring constant of the bar is zero in both the case that is in front and rear suspension. To find the roll momentum the weight of the bar is to be added to the weight of non suspended mass but this effect is very small to the roll of the vehicle.The main aim is to get the higher stiffness to the weight ratio to decrease the rolling effect of the body. Anti roll bar should be strong to the bending as well as twisting moment to reduce roll and it have less weight so that non suspended weight is less that why vehicle will follow the contour of the road due to its less inertia.

Following tables as well as graph shows result when the different geometrical variation done on the anti-roll bar, depending on that variation stiffness to the weight ratio is important to find the best suitable anti-roll bar.

3.1 Variation in the arm length 'e'

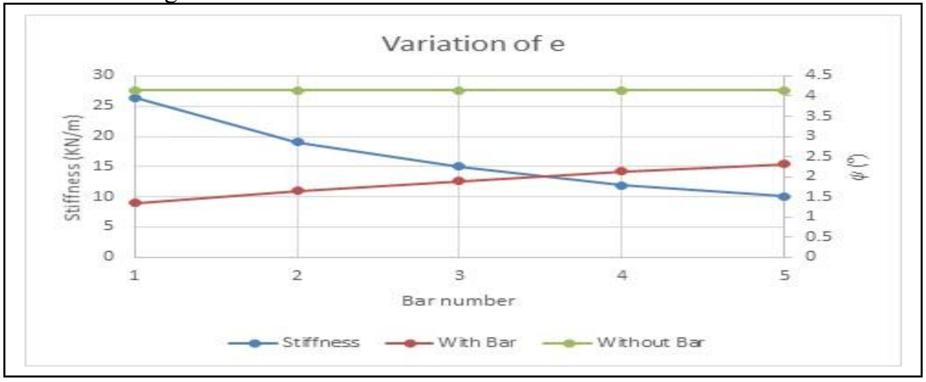


Figure-9 Effect of varying the length of arm 'e'

Following TABLE shows the effect of varying the arm length

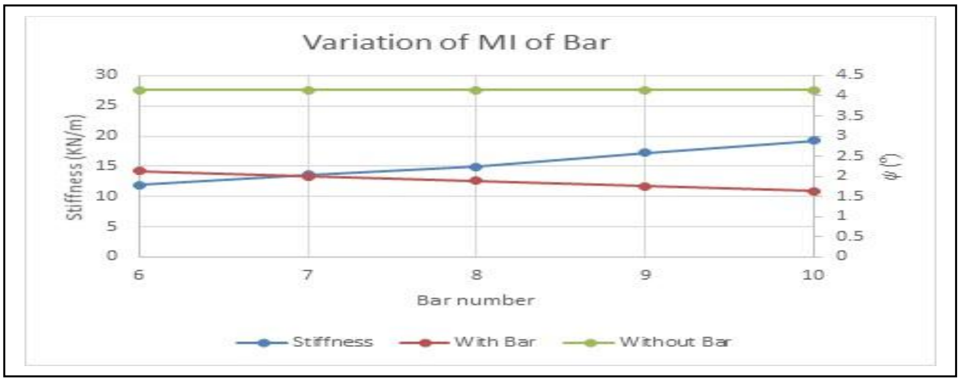
Table-14

Anti-roll bar	1	2	3	4	5
e (mm)	220	270	320	370	420
Weight (N)	33.42	35.95	37.35	38.74	40.14
K_{er} (KN/m)	26.42	19.07	14.99	11.92	10.02
Stiffness/Weight (1/m)	0.79	0.53	0.40	0.31	0.25
Ψ (°)	1.34	1.65	1.89	2.13	2.31
f (mm)	28.98	35.70	40.97	46.12	49.97

Spring constant of the anti-roll bar decrease when the length of bar 'e' is increased. It is observed that when the factor of rigidly is declined then roll angle of the vehicle is increased. The Table-14 shows that stiffness to weight ratio is decreased so bar becomes flexible which will to reduce the effect of reducing roll of the body. Out of Five bars first anti-roll bar shows best result which shows the minimum roll angle.

The best performance on the stiffness/weight ratio for the five configurations analyzed was achieved by the first bar, which provided the greatest reduction of the roll angle and displacement of 67.68% when compared to the vehicle without the anti-roll bar

3.2 *Variation* in the moment of inertia of A and B



Following TABLE shows the Effect of varying the moment of inertia of A and B on the stiffness to the weight ratio.

Table-15

Anti-roll bar	6	7	8	9	10
$Ø_o$ (mm) of arm A & B	21	22	23	24	25
$Ø_i$ (mm) of arm A & B	6.4	9.2	11.4	13.3	15
I (mm^4) of arm A & B	9464	11147	12907	14750	16689
J (mm^4) of arm A & B	18928	22294	25815	25900	33379
Weight (N)	40.14	40.14	40.14	40.14	40.14
K_{er}(KN/m)	11.87	13.59	14.91	17.24	19.22
Stiffness/Weight (1/m)	0.30	0.34	0.37	0.43	0.48
Ψ (°)	2.13	1.99	1.90	1.75	1.64
f (mm)	46.20	43.17	41.10	37.90	35.54

Variation in the stiffness of the anti-roll bar takes place due to the changes takes place in the momentum of the inertia. Flexural strength resistance is more if the momentum of inertia the bar is more and bar will have maximum torsion resistance when polar moment of inertia have maximum value. If the momentum of inertia and polar moment of inertia is more the bar will be have more rigidity.

Stiffness to weight ratio will be more in last bar which is the best result among the five bar .Roll angle and upper travel will be reduced in this anti-roll bar when we compared with anti-roll bar. The roll angle and the upper travel for this case were reduced by 60.37%, compared with the values of the vehicle without the anti-roll bar.

3.3. Variation of the distance between the bush 'S'

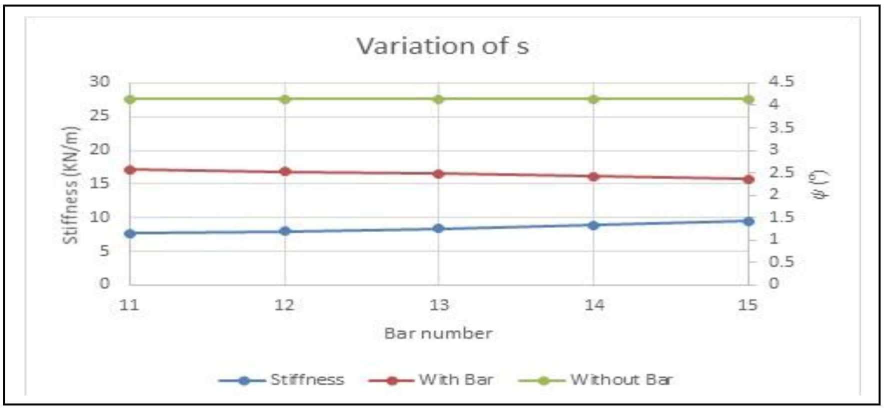


Figure-10 Effect of varying the distance between the bush

Following TABLE shows effect when distance between the bushes is changed.

Table-16

Anti-roll bar	11	12	13	14	15
s (mm)	100	150	200	250	300
Weight (N)	40.14	40.14	40.14	40.14	40.14
K_{eii} (KN/m)	7.64	8.00	8.42	8.92	9.51
Stiffness/Weight (1/m)	0.19	0.20	0.21	0.22	0.24
Ψ (°)	2.58	2.53	2.48	2.43	2.36
f (mm)	55.85	54.87	53.79	52.53	51.12

When bushing is far then the rigidity of the bar will increase as well as stiffness also increased .Rigidity to the assembly is given if bushing are away from each other but it will possible if extra support should not required for that arrangement and this will not add extra weight to the vehicle arrangement.

The best stiffness to weight ratio is given by last bar .Roll and 'f' that is upper travel is reduced by 43%, when compared with the values of the vehicle without the anti-roll bar.

3.4. Variation in angle between the two arms 'α'

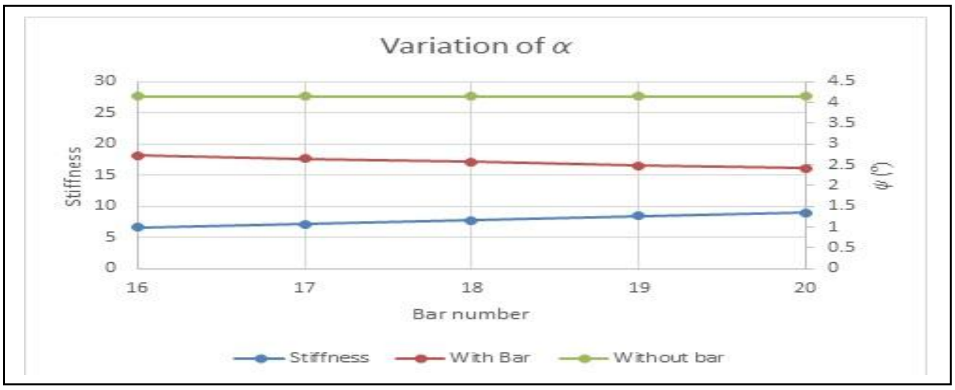


Figure-11 Effect of varying the angle between the arms

Following TABLE shows the effect of changing the angle between arms

Table-17

Anti-roll bar	16	17	18	19	20
α (^{0})	0	12	24	36	48
Weight (N)	47.95	46.16	44.50	42.96	41.51
K_{ell}(KN/m)	6.60	7.13	7.71	8.49	9.01
Stiffness/Weight (1/m)	0.14	0.15	0.17	0.20	0.22
Ψ (^{o})	2.72	2.65	2.57	2.47	2.41
f (mm)	58.88	57.31	55.65	53.60	52.31

From above figure it is observed that if angle between the arms is more the rigidity will be more. If angle small the length of the bar that subjected to the bending is greater than other bars, If the angle is increase then weight of the bar is reduced hence we got ratio of stiffness to the weight is increased and rigidity will improved Stiffness to the weight ratio is more in last result of bar 20. Roll angle and length of upper travel is reduced Over 41.67% when compared to the values of the vehicle without the anti-roll bar.

3.5. Changes in momentum of inertia of profile B of rectangular cross section.

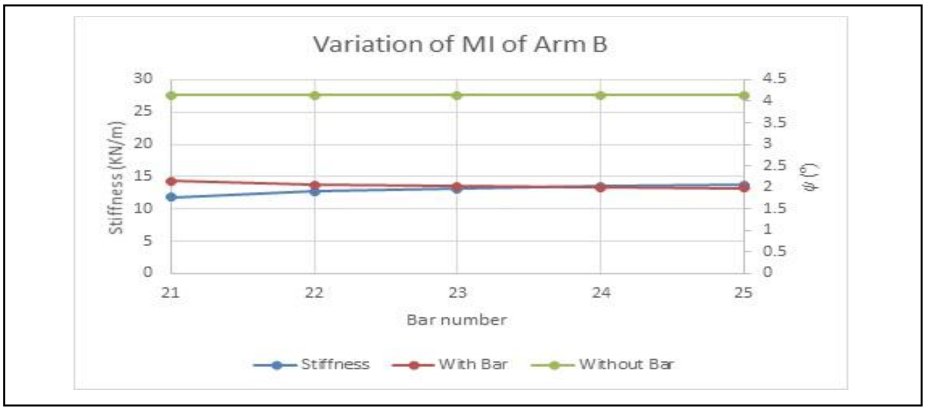


Figure-12 Shows the Effect of changing the momentum of inertia of profile B

Following TABLE shows changes in momentum of inertia of the cross section B.

Table-18

Anti-roll bar	21	22	23	24	25
Height c (mm)	20	22.5	25	27.5	30
Width d (mm)	20	17.8	16	14.5	13.3
I (Stretch B) (mm4)	13333.33	16896.09	20833.33	25129.56	29925
Weight (N)	46.22	46.26	46.22	46.13	46.15
K_{all}(KN/m)	11.78	12.69	13.06	13.50	13.74
Stiffness/Weight (1/m)	0.25	0.27	0.28	0.29	0.30
Ψ (°)	2.14	2.06	2.03	2.00	1.98
f (mm)	46.37	44.70	44.06	43.32	42.92

In this section another way to improve the stiffness is increasing momentum of inertia by varying the width and height of the profile B that is rectangular section. Weight is approximately constant in all bars and area also constant. Best result is for bar five in terms of stiffness to the weight ratio. The roll angle of body and 'f' that is upper travel is reduced by to 52.13% when compared with the values of the vehicle without the anti-roll bar.

3.6. Variation in the profile of A without changes in the momentum of inertia of section.

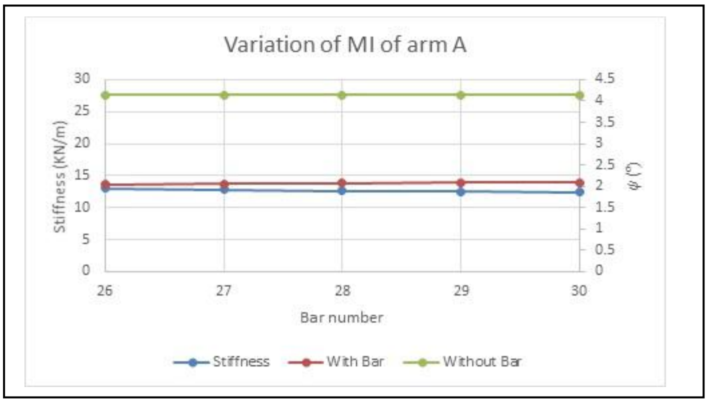


Figure-12 shows the effect of variation in profile A

Following TABLE shows changes in momentum of inertia of A profile.

Table-19

Anti-roll bar	26	27	28	29	30
$Ø_a$ (mm) of arm A	21	22	23	24	25
$Ø_i$ (mm) of arm A	13.6	16.5	18.6	20.4	21.9
A (mm^2) of arm A	285.6	363	427.8	489.6	547.5
Weight (N)	48.89	48.59	47.68	46.89	45.72
K_{all}(KN/m)	12.96	12.77	12.61	12.46	12.38
Stiffness/Weight (1/m)	0.27	0.26	0.26	0.27	0.27
Ψ (°)	2.04	2.06	2.07	2.08	2.09
f (mm)	44.23	44.57	44.85	45.12	45.26

In sixth analysis we can get that if profile is changed and polar moment of inertia is constant then stiffness does not changes which shown by practically also. By variation in cross section and loss in weight we can get same rigidity. After analysis last bar give good result in terms of stiffness to the weight ratio. Roll angle and 'f' upper travel length reduced by more than 49.53%, when compared to the values of the vehicle without the anti-roll bar.

IV. Conclusion

We have got satisfactory result of anti-roll bar in terms of reducing the roll of the body that is roll angle of the vehicle by using the simple geometry of anti-roll bar. we can reduce body roll up to 67.68 % than vehicle without anti-roll bar so finally we can conclude that if implementation of the our modified bar is used then finally stability of the vehicle will improved.

Best stiffness to weight which is having the minimum length of the arm 'e'. Rigidity of the anti-roll is important criteria which is achieved if the less portion of bar is subjected to bending and in terms of flexibility of bar we can conclude that bush should be away from with one another.

We have changed profile of bar, momentum of inertia of bar increased then we got that stiffness of bar is improved but without increment in weight this conclusion give idea about we can achieve our target but with reduced cost of the vehicle. Stiffness to weight ratio give conclusion about the rigidity of the bar ,in this paper we have done modification in anti-roll bar to analyze performance in terms of this ratio

In this project we have found the stiffness of bar by using the software of finite element method and for roll angle calculation we have used equation proposed by literature review author. Future work of this project is that stress analysis of the anti-roll bar which will show which areas of bar subjected to more stress and depending on that use of anti-roll bar.

REFERENCES

[1] Bernd Heibing, Metin Ersoy Chassis Handbook (Wiesbaden, Germany, 2011)
[2] Thomas D. Gillespie Fundamentals of Vehicle Dynamics (Society of Automotive Engineers, Warrendale, USA, 1992)
[3] SAE Spring Committee Spring Design Manual (Society of Automotive Engineers, 1996)
[4] Uhlmann, C., Análise da influência da barra estabilizadora no ângulo de rolagem da suspensão dianteira de ônibus Universidade Federal do Rio Grande do Sul, Porto Alegre, Brazil, 2009.
[5] Sérgio Yuri Ribeiro, Márcio Eduardo Silveira Application of Finite Element Method in the Study of Variables that Influence the Stiffness of the Anti-Roll Bar and the Body Roll Society of Automotive Engineers(SAE) journal 2013-36-0643
[6] Leal, L., Rosa, E.,Nicolazzi, L.,Uma Introdução à Modelagem Quase-Estática de Veículos Automotores de Rodas Internal Publication, Universidade Federal de Santa Catarina, Florianópolis, Brazil, 2008.
[7] Ian Czaja, Mohammad Hijawi , Automotive Stabilizer Bar System Design and Reliability Society of Automotive Engineers(SAE) journal 2004-01-1550
[8] Çalişkan, K., Automated design analysis of anti-roll bars Ankara, Turkey, 2003.
[9] Reimpell, J., and Stoll, H., The Automotive Chassis: Engineering Principles (Society of Automotive Engineers, Warrendale, USA, 1996.