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Optimization of Anti-Roll bar using Ansys Parametric Design Language (APDL)

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Abstract: The main goal of using anti-roll bar is to reduce the body roll. Body roll occurs when a vehicle deviates from straight-line motion. The objective of this paper is to analyze the main geometric parameter which affects rolling stiffness of Anti-Roll bar. By the optimization of geometric parameter, we can increase the rolling stiffness and reduce the mass of the bar. Changes in design of anti-roll bars are quite common at various steps of vehicle production and a design analysis must be performed for each change. To calculate rolling stiffness, mass, deflection, Von-mises stresses Ansys Parametric Design language (APDL) is used. The effects of anti-roll bar design parameters on final anti-roll bar properties are also evaluated by performing sample analyses with the FEA program developed in this paper.

Keywords— FEA, Anti Roll Bar, APDL, Design Parameters, Bushing position, Rolling stiffness, Deflection

INTRODUCTION

The anti-roll bar is a rod or tube that connects the right and left suspension members. It can be used in front suspension, rear suspension or in both suspensions, no matter the suspensions are rigid axle type or independent type. The ends of the anti-roll bar are connected to the suspension links while the center of the bar is connected to the frame of the car such that it is free to rotate. The ends of the arms are attached to the suspension as close to the wheels as possible. If the both ends of the bar move equally, the bar rotates in its bushing and provides no torsional resistance. But it resists relative movement between the bar ends, such as shown in Fig. 1. The bar's torsional stiffness-or resistance to twist-determines its ability to reduce such relative movement and it’s called as “roll stiffness”.

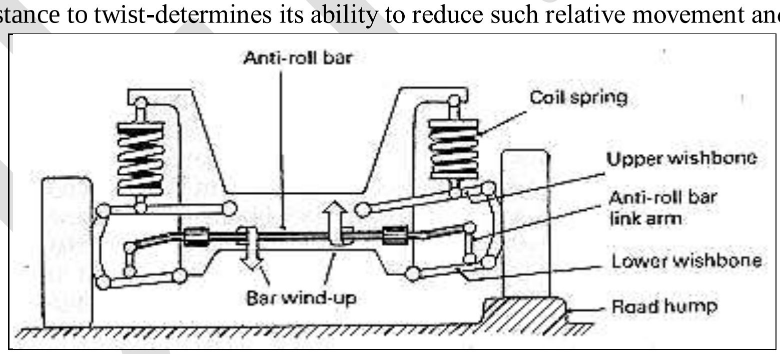

Fig.1– An anti-roll bar attached to double wishbone type suspension

Basic Properties of Anti-Roll Bars

Geometry

Packaging constraints imposed by chassis components define the path that the anti-roll bar follows across the suspension. Anti-roll bars may have irregular shapes to get around chassis components, or may be much simpler depending on the car. Two sample anti-roll bar geometries are shown in Fig.2. Anti-roll bars basically have three types of cross sections: solid circular, hollow circular and solid tapered, in recent years use of hollow anti-roll bars became more widespread due to the fact that, mass of the hollow bar is lower than the solid bar.

Fig.2 - Sample anti-roll bar geometries

Material and Processing:

Anti-roll bars are usually manufactured from SAE Class 550 and Class 700 Steels. The steels included in this class have SAE codes from G5160 to G6150 and G1065 to G1090, respectively. Operating stresses should exceed 700 MPa for the bars produced from these materials. Use of materials with high strength to density ratio, such as titanium alloys, is an increasing trend in recent years.

Connections

Anti-roll bars are connected to the other chassis components via four attachments. Two of these are the rubber bushings through which the anti-roll bar is attached to the main frame. And the other two attachments are the fixtures between the suspension members and the anti-roll bar ends, either through the use of short links or directly.

Bushings

There are two major types of anti-roll bar bushings classified according to the axial movement of the anti-roll bar in the bushing. In both types, the bar is free to rotate within the bushing. In the first bushing type, the bar is also free to move along bushing axis while the axial movement is prevented in the second type.

Fig.3 – Bushing (rubber bushings and metal mounting blocks)

The bushing material is also another important parameter. The materials of bushings are commonly rubber, nylon or polyurethane, but even metal bushings are used in some race cars [4].

The main goal of using anti-roll bar is to reduce the body roll. Body roll occurs when a vehicle deviates from straight-line motion. The line connecting the roll centers of front and rear suspensions forms the roll axis roll axis of a vehicle. Center of gravity of a vehicle is normally above this roll axis. Thus, while cornering the centrifugal force creates a roll moment about the roll axis, which is equal to the product of centrifugal force with the distance between the roll axis and the center of gravity. This moment causes the inner suspension to extend and the outer suspension to compress, thus the body roll occurs [5].

LITERATURE REVIEW :-

[1] Kelvin Hubert, Spartan chassis et.al studied and explained anti-roll bars are usually manufactured from SAE Class 550 and Class 700 Steels. The steels included in this class have SAE codes from G5160 to G6150 and G1065 to G1090, respectively. Operating stresses should exceed 700 MPa for the bars produced from these materials.

[2] Mohammad Durali and Ali Reza Kassaiezadeh studied & proposed the main goal of using anti-roll bar is to reduce the body roll. Body roll occurs when a vehicle deviates from straight-line motion. The line connecting the roll centers of front and rear suspensions forms the roll axis roll axis of a vehicle. Center of gravity of a vehicle is normally above this roll axis. Thus, while cornering the centrifugal force creates a roll moment about the roll axis, which is equal to the product of centrifugal force with the distance between the roll axis and the center of gravity.

[3] J. E. Shigley, C.R. Mischke explained that the moment causes the inner suspension to extend and the outer suspension to compress, thus the body roll occurs. Actually, body roll is an unwanted motion. First reason for this is the fact that, too much roll disturbs the driver and gives a feeling of roll-over risk, even in safe cornering. Second reason is its effect on the camber angle of the tires. The purpose of camber angle is to align the wheel load with the point of contact of the tire on the road surface. When camber angle is changed due to body roll, this alignment is lost and also the tire contact patch gets smaller.

MATHEMATICAL MODELING OF ANTI-ROLL BAR

Society of Automotive Engineers (SAE), presents general information about torsion bars and their manufacturing processing in "Spring Design Manual" . Anti-roll bars are dealt as a sub-group of torsion bars. Some useful formulas for calculating the roll stiffness of anti-roll bars and deflection at the end point of the bar under a given loading are provided in the manual. However, the formulations can only be applied to the bars with standard shapes (simple, torsion bar shaped anti-roll bars) [6].The applicable geometry is shown in Fig.4.

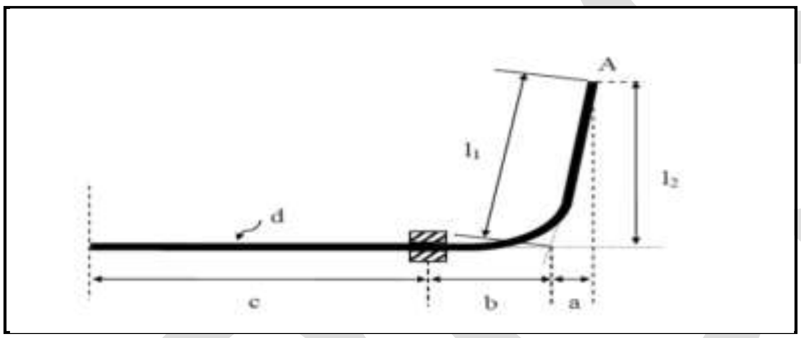

Fig.4 -Anti-roll bar geometry used in SAE Spring Design Manual

The loading is applied at point A, inward to or outward from plane of the page. The roll stiffness of such a bar can be calculated as:

L= a+b+c ……………….. (1)

(L:- half track length)

$$f_A = \frac{P[l1^3 - a^3 + \frac{L}{2}(a+b)^2 + 4l2^2(b+c)}{3EI} \quad \text{………………… (2)}$$

(f_A :- Deflection of point A)

$$KR = \frac{PL^2}{2fA} \quad \text{………………….. (3)}$$

KR: - Roll Stiffness of the bar

Max shear stress = $\frac{\mathbf{T * R}}{\mathbf{J}}$ …………………. (4)

Analysis

1. Define Element Types, Element Real Constants and Material Properties.
2. Modeling the Anti-Roll Bar
3. Applying Boundary Conditions and Loads

   The displacement constraints exist at two locations: at the bar ends and at bushing locations.

   The Ux, Uz degrees of freedom are constrained at the bar ends for spherical joints. ROTy and ROTz degrees of freedom are also constrained if pin joints are used. At the bushing locations, free ends of the springs are constrained in all Ux, Uy and Uz degrees of freedom. These elements have no rotational dof's. The other ends of the spring, attached to the beam, are constrained according to the type of the bushing. Ux dof is constrained for the second bushing type which does not allow bar movement along bushing axis. The loading for the first load step -determination of roll stiffness- is a known force, F, applied to the bar ends, in +y direction at one end and in –y direction at the other end as shown in Fig.5.

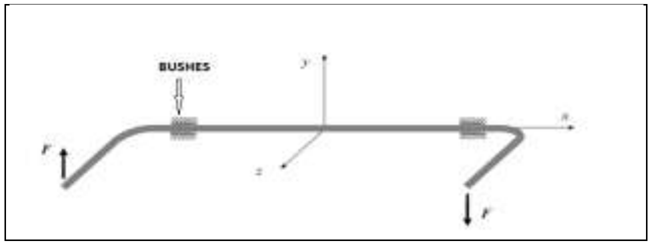

Fig.5- Load step

4. Solution & Post-processing

PROGRAM FOR STRUCTURAL ANALYSIS AND OPTIMIZATION OF ANTI-ROLL BAR

COMMAND	DISCRIPTION
Fini /clear /filname,Optimisation /title, Structural Analysis and Optimisation of Anti-Roll Bar C*** processing /prep7 *ask,Outer_dia,Outer diameter of bar,21.8 *ask,Inner_dia,Internal diameter of bar,16 *ask,Length,Total length of the bar,1100 *ask,Width,total width of the bar,230 *ask,r,fillet radius,50 *ask,Bush_Pos,Postision of Bush,390 *ask,Bush_L,Length of Bush,40 *ask,Load,VerticalLoad on bar,1000	• *Ask - Enter Input Value
k,1,0,0,0 k,2,(Length/2-Outer_dia/2),0,0 k,3,(Length/2-Outer_dia/2),0,Width k,4,-(Length/2-Outer_dia/2),0,0 k,5,-(Length/2-Outer_dia/2),0,Width k,6,(Bush_Pos-Bush_L/2),0,0 k,7,(Bush_Pos+Bush_L/2),0,0 k,8,-(Bush_Pos-Bush_L/2),0,0 k,9,-(Bush_Pos+Bush_L/2),0,0	**K**, *NPT*, *X*, *Y*, *Z* **Defines a line between two keypoints.**
l,1,6 l,6,7 l,7,2 l,2,3 l,1,8 l,8,9 l,9,4 l,4,5	**L**,P1,P2 **Defines a line between two keypoints.**
/pnum,line,1 /pnum,kp,1	**PNUM**, Label, KEY **Controls entity numbering/coloring on plots.**
Lplot	**LPLOT**, NL1, NL2, NINC **Displays the selected lines.**
lfilt,3,4,r lfilt,7,8,r	**LFILLT**, NL1, NL2, RAD, PCENT **Generates a fillet line between two intersecting lines**

k,14,5,0,0 k,15,0,0,-5	
circle,1,Outer_dia/2,14,15	**CIRCLE**, PCENT, RAD, PAXIS, PZERO, ARC **Generates circular arc lines.**
al,11,12,13,14 circle,1,Inner_dia/2,14,15 al,15,16,17,18	
/pnum,area,1	
Aplot	**APLOT**, NA1, NA2, NINC, DEGEN, SCALE **Displays the selected areas**
asba,1,2,,,2	**ASBA**, NA1, NA2, SEPO, KEEP1, KEEP2 **Subtracts areas from areas.**
lsel,s,,,11,18,1 Lplot	**LSEL**, Type, Item, Comp, VMIN, VMAX, VINC, KSWP **Selects a subset of lines.**
lesize,all,,,5	**LESIZE**, NL1, SIZE, ANGSIZ, NDIV, SPACE, **Specifies the divisions on unmeshed lines.**

Results obtained from APDL

- Max. Equivalent Stress = 332.307 MPa
- Max. Principal Stress = 351.409 MPa
- Roll Stiffness = 408.62 Nm/deg
- Deflection = 25.86
- Mass = 1.85

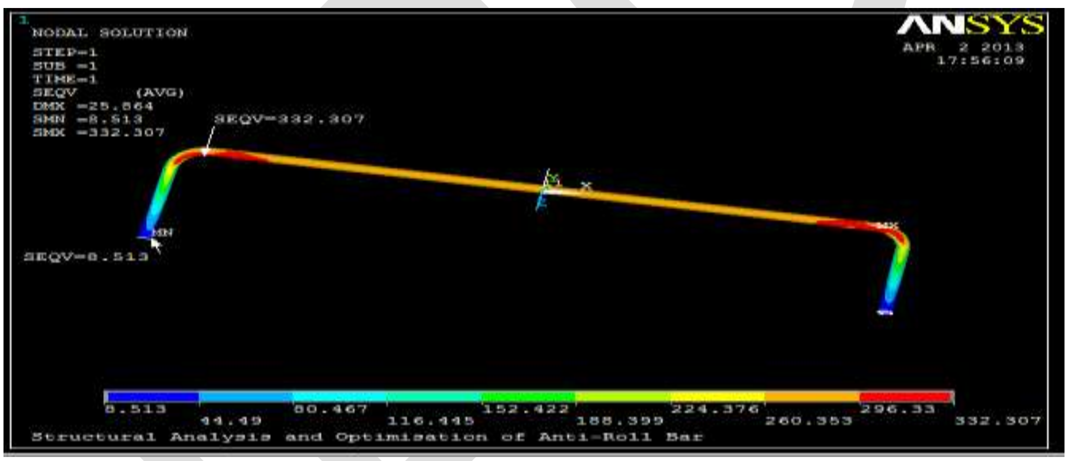

Fig.6- Equivalent Von Mises Stress Distribution on the Bar

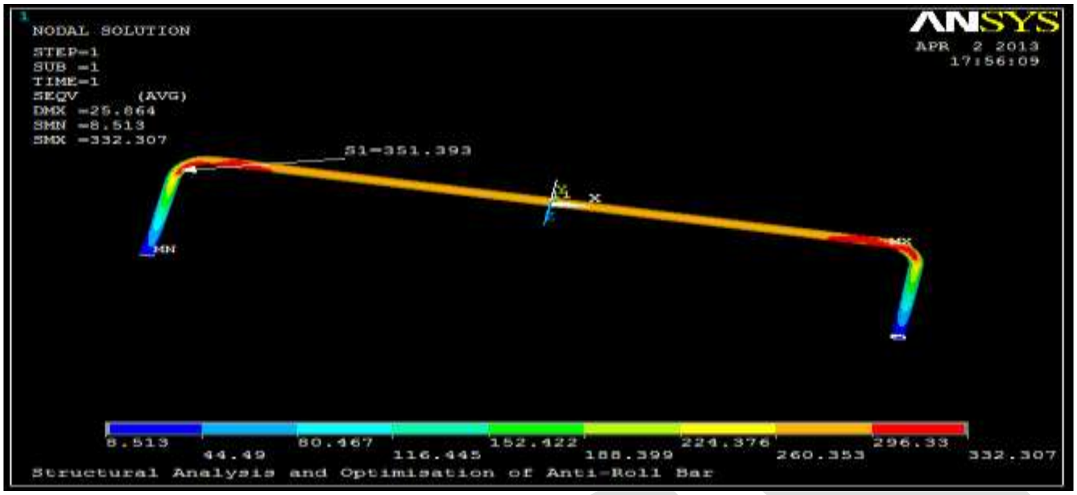

Fig.7- Principal Stress Distribution on the Bar

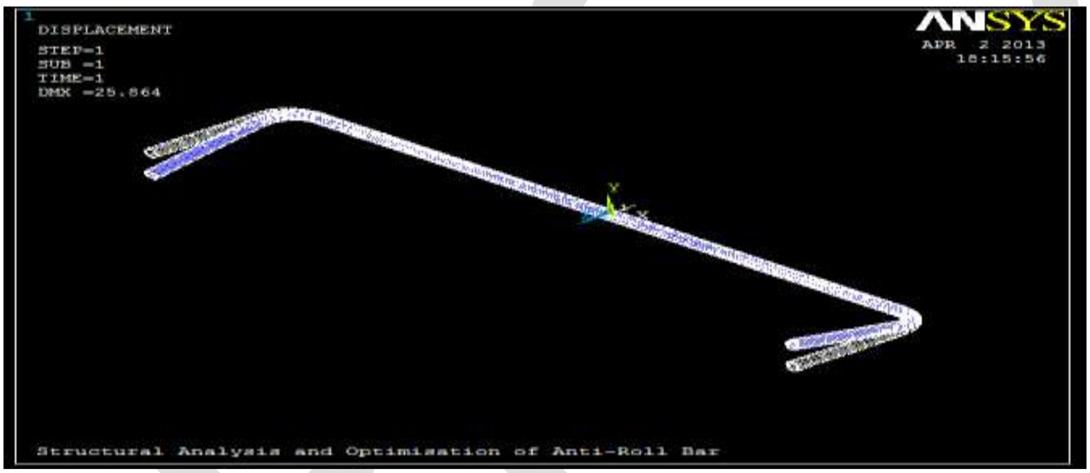

Fig.8- Deflection of Bar

SAMPLE HAND CALCULATIONS OF ANTI-ROLL BAR

Sample calculations are done considering all the input parameters given below. Design parameters are assigned as follows:

- Cross-section type = Hollow
- Outer radius = 10.9 mm
- Inner radius = 8 mm
- Bushing type = 1 (x movement free)
- Bushing position = ± 400 mm
- Bushing length = 40 mm
- Bushing Stiffness = 1500 N/mm
- End connection type = 1 (spherical joint)
- Bar material = SAE 5160
- E = 206000 N/mm^2, υ= 0.27, Syt= 1200 MPa, Sut = 1400 MPa, ρ = 7800 kg/m^3

The automated design software gives the end deflection of the anti-roll bar under a load of 1000 N as:
Deflection (f_A) = **25.86 mm**

Rolling Stiffness (KR) $= \frac{PL}{tan^{-1}\left(\frac{fA}{\frac{L}{2}}\right)}$ = **408.62** N.m/**deg**

According to the SAE formulations, roll stiffness can be calculated as:-

$$f_A = \frac{1000\left[(250^3)-(90^3)+550*(160^2)+4*(230^2)*460\right]*64}{3*206000*\prod*(21.8\text{^}4)-(16\text{^}4)} = \mathbf{25.96\ mm}$$

$$\mathbf{KR} = \frac{1000*(1100^2)}{(2*25.96)} = \mathbf{23.296\ N.m/rad = 406.59\ N.m/deg}$$

There is 0.5 % difference between mathematical and simulation results

OPTIMIZATION OF ANTI-ROLL BAR

The main goal of using Anti-roll bar is to reduce the body roll, for that purpose we need to increase roll stiffness of Anti-roll bar and also we need to reduce the weight of the Anti-roll bar. For improving Anti-roll bar performance optimization is necessary [7]. Optimization is done by trial and error method through 20 results as follows.

Table 1: Results obtained for Optimized Anti-Roll Bar

O.D (mm)	I.D (mm)	Bushing Position (mm)	Deflection (mm)	Von Mises Stress Max (N/Sq.mm)	Max Principal Stress (N/Sq.mm)	Rolling Stiffness (N.m/deg)	Mass Of The Bar (Kg)
21.8	16	300	27.814	389.636	351.417	379.96	1.85
21.8	16	350	26.562	345.562	351.409	397.84	1.85
21.8	16	390	25.964	332.270	351.404	406.98	1.85
21.8	16	400	25.864	332.307	351.393	408.56	1.86
21.8	**16**	**420**	**25.694**	**344.59**	**351.558**	**411.26**	**1.85**
20	16	300	47.575	609.889	588.642	222.50	1.217
20	16	350	45.43	540.162	588.578	232.96	1.217
20	16	390	44.414	525.67	588.63	238.26	1.217
20	16	420	43.945	536.015	588.60	240.79	1.217
20	16.5	300	52.58	671.775	682.968	201.43	1.080
20	**16.5**	**350**	**50.23**	**607.366**	**682.79**	**210.80**	**1.080**
20	16.5	390	49.104	609.43	682.77	215.60	1.080
20	16.5	420	48.58	610.079	682.98	217.92	1.080
21	16.5	300	37.26	501.794	480.978	283.83	1.426
21	17	350	38.77	489.016	546.56	272.80	1.285

21	17	390	37.90	485.247	546.58	279.04	1.285
21	16.5	420	34.426	445.63	481.09	307.12	1.426
20	15	350	38.996	465.494	469.837	271.23	1.18
20	15	400	37.968	439.45	469.776	278.55	1.18
20	17	390	55.55	761.535	819.592	190.72	0.75

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CONCLUSION

- Vehicle's performances are strongly affected by tuned Anti roll bar by changing the parameters of the bar.
- The time required for analysis of Anti-roll bar using APDL (Ansys Parametric Design Language) is very short and can be repeated simply after changing any of the input parameters which provides an easy way to find an optimum solution for anti-roll bar design.
- The most obvious effect of using hollow section is the reduction in mass of the bar.
- Locating the bushings closer to the centre of the bar increases the stresses at the bushing locations which results in roll stiffness of the bar decreases and the max Von mises stresses increases.
- By increasing the bushing stiffness of Anti-roll bar, increases Anti- roll stiffness, also increasing the stresses induce in the bar.

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ANSYS Help for Version 12.0