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Hollow Cross-Section vs. Solid Cross-Section & Increasing the Diameter of Solid Cross-Section by using finite element Analysis of Anti-Roll Bar

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Abstract -Vehicle anti-roll bar is part of an automobile suspension system which limits body roll angle. This U-shaped metal bar connects opposite wheels together through short lever arms and is clamped to the vehicle chassis with rubber bushes. Its function is to reduce body roll while cornering, also while travelling on uneven road which enhances safety and comfort during driving. Design changes of anti-roll bars are quite common at various steps of vehicle production, and a design analysis must be performed for each change. So Finite Element Analysis (FEA) can be effectively used in design analysis of anti-roll bars. The finite element analysis is performed by ANSYS. The study includes five steps: information gathering, pre-processing, analysis, post-processing, and analyzing the FEA results to arrive at conclusions. 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 study.

Key words- Antiroll bar, roll stiffness, FEA.

I.INTRODUCTION

Anti-roll bar, also referred to as stabilizer or sway bar, is a rod or tube, usually made of steel, that connects the right and left suspension members together to resist roll or swaying of the vehicle which occurs during cornering or due to road irregularities. The bar's torsional stiffness (resistance to twist) determines its ability to reduce body roll, and is named as "Roll Stiffness". An anti-roll bar improves the handling of a vehicle by increasing stability during cornering. Most vehicles have front anti-roll bars. Anti-roll bars at both the front and the rear wheels can reduce roll further. Properly chosen (and installed), anti-roll bars will reduce body roll, which in turns leads to better handling and increased driver confidence. A spring rate increase in the front anti-roll bar will produce understeer effect while a spring rate increase in the rear bar will produce oversteer effect. Thus, anti-roll bars are also used to improve directional control and stability. One more benefit of anti-roll bar is that, it improves traction by limiting the camber angle change caused by body roll. Anti-roll bars may have irregular shapes to get around chassis components, or may be much simpler depending on the car.

II.THEORY

A. Anti-Roll Bars and Vehicle Performance

Ride comfort, handling and road holding are the three aspects that a vehicle suspension system has to provide compromise solutions. Ride comfort requires insulating the vehicle and its occupants from vibrations and shocks caused by the road surface. Handling requires providing safety in maneuvers and in ease in steering. For good road holding, the tires must be kept in contact with the road surface in order to ensure directional control and stability with adequate traction and braking capabilities. The anti- roll bar, as being a suspension component, is used to improve the vehicle performance with respect to these three aspects. 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. A typical anti-roll bar is shown in Figure 1. The ends of the anti-roll bar are connected to the suspension links while the centre 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. The bar's torsional stiffness-or resistance to twist-determines its ability to reduce such relative movement and it's called as "roll stiffness".

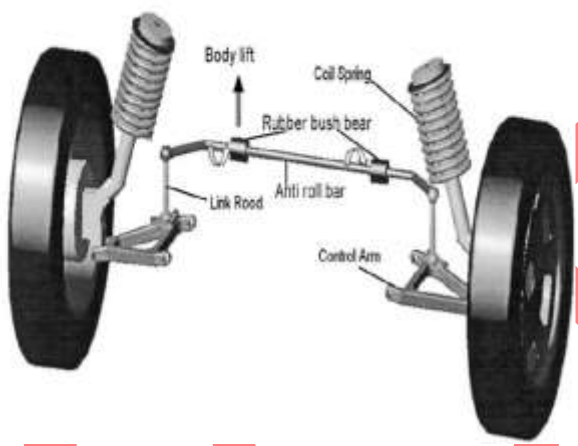


Fig.1. A typical anti-roll bar

B. Design of Anti-roll Bars

The design of an anti-roll bar actually means to obtain the required anti-roll stiffness that improves the vehicles' stability and handling performance without exceeding the mechanic limitations of the bar material. Since, it's a straightforward process to analyze the anti-roll bar, it's not possible find published studies in the literature. The standard design analyses are performed by manufacturer companies, and the results are not published. Rather, the studies focused on the bushing characteristics and fatigue life analysis of the anti-roll bars is available. Also, some design automation studies about anti-roll bars are present.

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).The applicable geometry is shown in Figure 2.

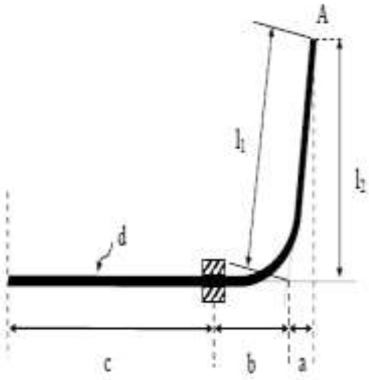


Fig.2. Anti-roll bar geometry

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:

$$L = a + b + c$$

(L = half track length)

$$f_A = \frac{P}{3EI}\left[l_1^3 - a^3 + \frac{L}{2}(a+b) + 4l_2^2(b+c)\right]$$

(f_A = Deflection of point A)

$$k_R = \frac{PL^2}{2f_A} \quad \text{Nmm/rad}$$

(k_R = Roll Stiffness of the bar)

III. ANALYSIS OF ANTI-ROLL BARS IN ANSYS

A typical ANSYS analysis has three distinct steps:

1. Build the model. 2. Apply loads and obtain the solution. , 3. Review the results.

These 3 steps are performed using pre-processing, solution and post-processing processors of the ANSYS program. Actually, the first step in an analysis is to determine which outputs are required as the result of the analysis, since the number of the necessary inputs, analysis type and result viewing methods vary according to the required outputs. After determining the objectives of the analysis, the model is created in pre-processor. The next step, which is to apply loads, can be both performed in pre-processor or the solution processor. However, if multiple loading conditions are necessary for the required outputs and if it is also necessary to review the results of these different loading conditions together, solution processor must be selected for applying loads. The last step is to review the results of the analysis using post-processor, with numerical queries, graphs or contour plots according to the required outputs.

i. Determination of Design Outputs

The basic goals of using anti-roll bars are to reduce body roll during cornering and to improve handling characteristics of the vehicle. The roll-stiffness property of the anti-roll bar is used to provide extra roll-stiffness to the front or rear suspensions. Therefore, to perform an anti-roll bar analysis basically means to determine its roll stiffness. In order to determine the roll stiffness, the deflection of the bar ends under a defined loading, in the direction of suspension motion, must be obtained. This deformation value, with some trigonometric relationships, can be then used for calculating the roll-stiffness of the bar.

ii. Determination of Design Parameters

The parameters of anti-roll bar design are bar cross- section, bushing location, stiffness of the bushing material end connection type. Bar geometry is defined by a single curved bar centerline. The cross-section types that will be considered in this study are solid circular and hollow circular, since use of tapered bars are not common. Two types of bushings will be considered in the analysis, one of which constraints the bar movement along bushing axis while the other not. Also spherical and pin joints will be used for providing bar ends' connection to the suspension members.

iii. Determination of Constraints and Loads

The anti-roll bar is connected to the other chassis components via four attachments. Two of these are the bushings through which the bar is connected to the main chassis of the vehicle, while the other two are the connections between the bar and the suspension links at bar ends. At the bushing connections, the bar is free to rotate within the bushing and its vertical and lateral movements are constrained by the bushing material in both bushing types. However, movement along bushing axis is dependent on the bushing type. This movement may be constrained or not. At the bar ends, since the bar is to travel vertically along with the suspension member, bar ends' lateral displacements are constrained. These constraints may create some erroneous results if the suspension member does not travel absolutely vertical, but this is not a common case. If end connections are provided with spherical joints, there are no rotational constraints while only the rotational degree of freedom about x-axis exists for the pin joint and the other two rotational freedoms are constrained.

When the vehicle experiences body roll, one wheel will pull one end of the stabilizer bar down while the other wheel will pull the opposite end of the stabilizer bar up. The loading of the bar is the relative displacement of the bar ends which are connected to the suspension members. Hence, the stabilizer bar will be under combined bending and torsional loading. The deflection of the bar ends is related to maximum permissible body rolls angles. For passenger cars this angle is limited around 3.5°. Assuming a track length of 1300 mm with a beam axle suspension, the maximum deflection at the bar ends will be around:

$$f_A = (1300/2)\sin(3.5) \cong 40 < 50$$

This displacement will be smaller for independent type suspensions.

A. Analysis

i. Basic Procedures- After starting the ANSYS session, a jobname and an analysis title should be defined before entering any processors.

ii. Define Element Types, Element Real Constants and Material Properties

After performing the basic analysis procedures, the user must enter PREP7 preprocessor in order to continue with the analysis. The first thing to be done in the pre- processor is to define the element types. Two different element types are required for modeling the anti-roll bar with its bushings.

iii. Modeling the Anti-Roll Bar- The geometry of any anti-roll bar, even with irregular shapes, can be defined by a single curved bar centreline and a cross-sectional area swept along this centreline. The cross-section of the bar was created in the previous sections. This cross-section will be assigned to the beam elements during meshing of the beam. However, before any other operations the bar centreline must be created. Two alternatives exist at this stage, creation of the model in ANSYS or importing the model from Proe, Catia etc.

iv. Modeling the Bushings - Modeling the bushings of the anti-roll bar requires careful attention on the structures of the bushings. Two types of bushings, in both types, bar is free to rotate within the bushing. This property is automatically accomplished by setting element as longitudinal springs. Since longitudinal spring elements have only translational degrees of freedom at their nodes, they cannot resist rotation. The basic difference between the two bushing types is the movement of the bar along bushing axis. Consider the first type, where the bar is free to move axially within the bushing. Here, the only restriction on the bar exists for its radial movement.

iv. Applying Boundary Conditions and Loads

This step can be performed in PREP7 preprocessor or SOLUTION processor. Since there are two loading conditions, one for obtaining roll stiffness and one for determining maximum stresses under maximum loading, and since it's preferred to review the results of these two loadings together, SOLUTION processor must be selected for applying 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. 3.

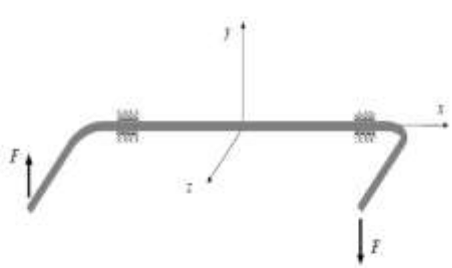


Fig.3. Load Step1

For the second load step the force loads of the first load step are removed and displacement loads, representing maximum suspension deflection, are applied to the bar ends again in opposite directions.

vi. Solution- The first step of solution is to choose the analysis type based on the loading conditions and the required outputs. For the first two loading cases given in the previous section, analysis type is static, since the loading is steady.

vii. Post-processing the Results- POST1 post-processor of ANSYS is used for reviewing the analysis results. POST1 has many capabilities, ranging from simple graphics displays and tabular listings to more complex data manipulations such as load case combinations. The first step in POST1 is to read data from the results file into the database. When each load step is solved in the SOLUTION processor, the results of that load step are written to a results file. This results file must be read into database for post-processing.

IV. SAMPLE ANALYSES AND DISCUSSION OF THE RESULTS

A. Sample Analysis with a Typical Anti-roll Bar

The first sample analysis is performed with a solid round anti-roll bar. Moment-free bushings and spherical joints at bar ends are used for connections. The results obtained in this analysis will be used for demonstrating the program outputs. Also, the same anti-roll bar geometry will be used with different connection types and in other sample analysis in order to discuss the effects of the design parameters on anti-roll bar performance. The units used in all graphs, plots and tables are in terms of N, mm and MPa.

Inputs: The preview of the geometry to be analyzed. This geometry is created in ANSYS, and the keypoint coordinates and fillet radii.

After generating the model, other design parameters are assigned as follows:

Cross-section type = Solid round cross-section, Section radius = 10 mm, Bushing type = 1 (x movement free), Bushing locations = ± 390 mm, Bushing length = 40 mm, Bushing Stiffness = 1500 N/mm, End connection type = 1 (spherical joint), Bar material = SAE 5160, E=206 GPa, υ = 0.27, Sy = 1180 MPa, Sut = 1400 MPa, ρ = 7800 kg/m^3,Number of elements = 100, Loading = ± 50 mm on both sides

Results: The results obtained from the analysis of this bar are:

Roll Stiffness = 410.2 Nm/deg

Total Length = 1394 mm

Mass = 3.416 kg

Max. Prin. Stress = 578.9 MPa Max. Eqv. Stress = 652.1 MPa Max. Prin. Strain = 0.304 % Max. Eqv. Strain = 0.372 %

The variation of equivalent and principal stresses along bar length are plotted in fig.4. and fig.5.

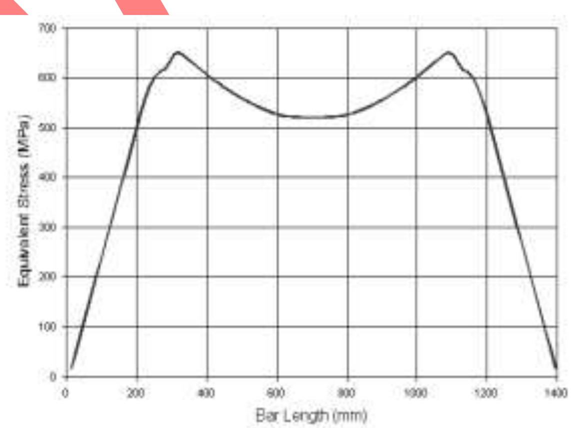


Fig. 4- Variation of Equivalent Stress along bar length

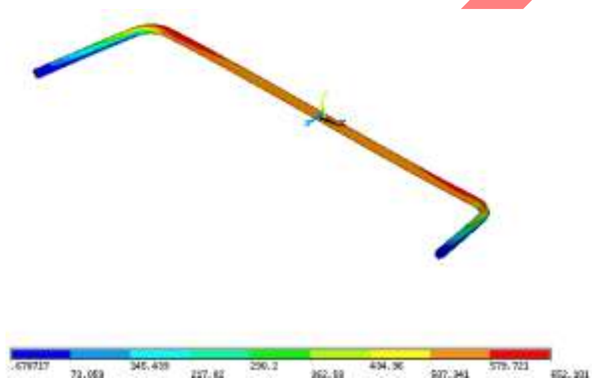


Fig.5- Equivalent Stress Distribution on the Bar

B. Effects of Bar Cross-Section Type and Dimensions:

In this analysis the effect of the bar cross-section on the bar properties is presented. The primary parameter to be considered for an anti-roll bar is the roll stiffness. Therefore, the anti-roll bar stiffness obtained in part (A), is obtained with a hollow bar and the other analysis results are compared.

Case1 - Hollow Cross-Section vs. Solid Cross-Section

Inputs: All inputs are same as part (A) except cross-section properties. After some iteration the following section dimensions are determined in order to obtain specified roll stiffness:

Cross-section type = Hollow cross-section

Section outer radius = 10.9 mm, Section inner radius = 8 mm

Results:

Following results are obtained from the analysis of the hollow bar as Mass = 1.872 kg, Max. Eqv. Stress = 683 MPa As seen from Fig.4. variation of equivalent stress along bar length is same as (A), except the values of the peaks. Same situation is valid for principal stress variation along bar length. The contour plot of equivalent stress shown in Fig.6 is also similar to part (A).

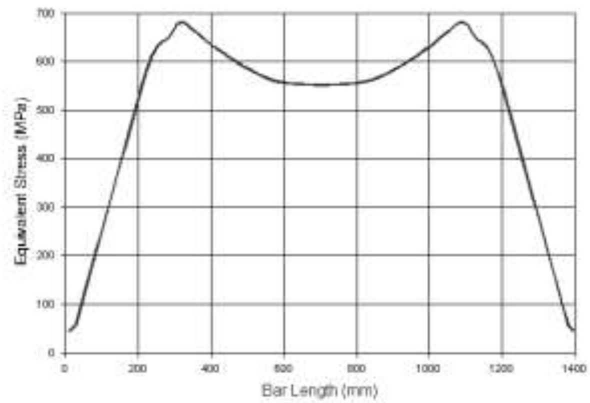


Fig. 6.Variation of Equivalent stress along bar length (Hollow cross- section)

Case2 – Increasing the Diameter of Solid Cross-Section

Inputs: All inputs are same as part (A) except cross-section dimension is Section radius = 12 mm

Results:

Following results are obtained from the analysis of the hollow bar:

Roll Stiffness = 805 Nm/deg, Mass = 4.919 kg

Max. Prin. Stress = 657 MPa

Max. Eqv. Stress = 741 MPa

The variations of the stresses on the bar are same as part (A) except the peak values.

C. Changing Bushing Locations

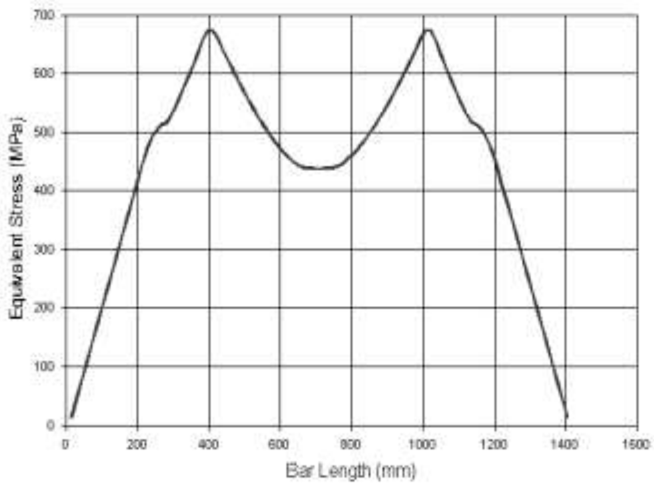
Anti-roll bars may have various shapes since their geometry depends on the availability of space in the chassis. Also, bushings can be located at different positions. Normally bushings are fitted near bend portions as close as possible. The effect of bushing locations will be now analyzed by using bushings closer to centre compared to part (A).

Inputs:

All inputs are same as part(A) except bushing locations. Bushing locations = ± 300 mm on both sides

Results:

Max. Prin. Stress = 622.6 MPa Max. Eqv. Stress = 678.0 MPa Roll Stiffness = 342.8 Nm/deg



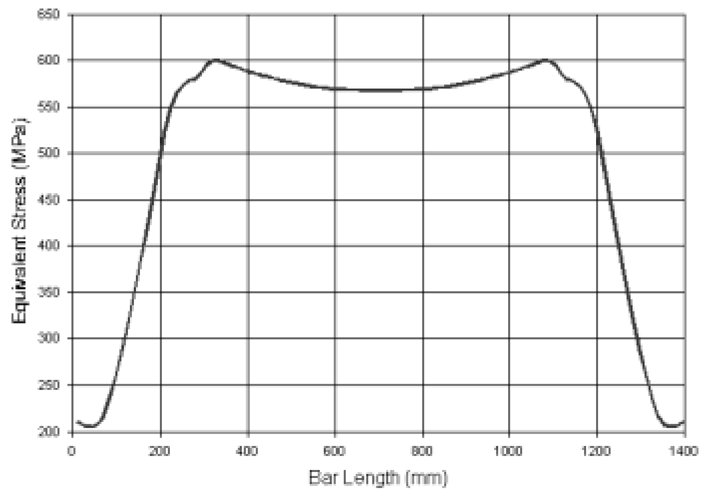


Fig. 7.Variation of Equivalent stress along bar length
(Bushing locations = ± 300 mm)

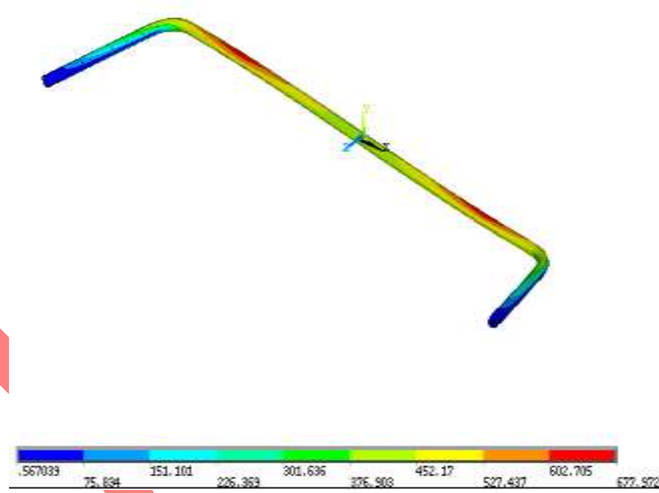


Fig.8.Equivalent Stress Distribution on the Bar (Bushing locations = ±300 mm)

D. Effect of end Connection Type:

There are two joint types used for connecting anti-roll bar to the suspension members; spherical joints and pin joints. In part(A) spherical joints were used for the providing end connections. Now, pin joints will be used and analysis results will be compared with part (A).

Inputs:

End Connection Type = pin joint

Results:

Roll Stiffness = 449.4 Nm/deg Max. Prin. Stress = 528.0 MPa Max. Eqv. Stress = 600.8 MPa

The equivalent stress distribution on the bar is given in **Fig. 9**

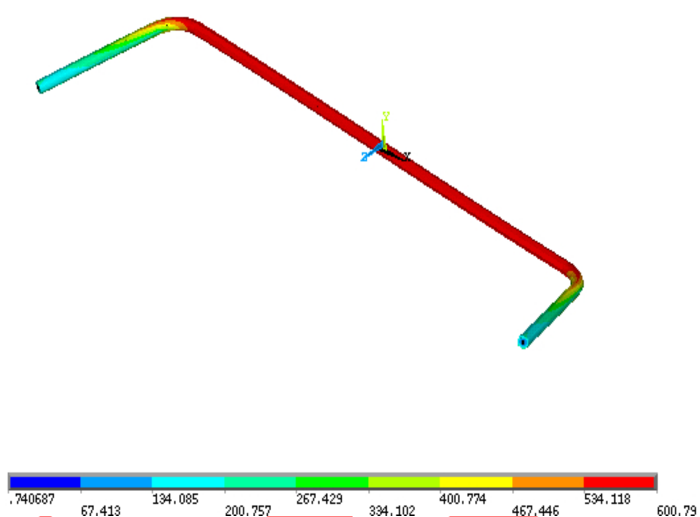


Fig. 9. Variation of Equivalent stress along bar length (end pin joint)

V.CONCLUSION

Following conclusions are derived about anti-roll bar design parameters from FEA: -

1. Increasing the cross-sectional diameter of an anti- roll bar will increase its roll stiffness. But larger stresses occur on the bar for the same bar end deflection.
2. The weight of the hollow anti-roll bar is less than the solid bar having the same roll stiffness. However, the stresses on the hollow bar are higher.
3. Locating the bushings closer to the centre of the bar increases the stresses at the bushing locations while roll stiffness of the bar decreases.
4. If the pin joints are used at the bar ends, the Stresses near the ends are increased. The roll stiffness of the bar is increased while the maximum stresses are decreased due to distribution of the stresses along the length of the bar.

REFERENCES

1) M. Murat TOPAÇ, H. Eren ENGİNAR, N. Sefa KURALAY, "Reduction of stress concentration at the corner bends of the anti-roll bar by using parametric optimisation", Mathematical and Computational Applications, Vol. 16, No. 1, pp. 148-158, 2011

2) Tadashi Skai ,Shinichi Nishizawa, akitoshi Hamamoto, "Bend Shapes For Anti roll bar" United States Patents Application Publication, Pub. No. US 2011/0101630A1,Pub Date May5,2011

3) Tim Webb, Karen Curtis, "Abaqus FEA and fe-safe™ Used for Automobile Stabilizer Bar Analysis", INSIGHTS January/February 2010

4) Adam-Markus WITTEK*, Hans-Christian RICHTER, "STABILIZER BARS: Part 1. CALCULATIONS AND CONSTRUCTION", PROBLEMY TRANSPORTU, Krasińskiego St. 8, 40-019 Katowice, PolandVolume 5 Issue 4,2010

5) P.Senapathi, S.Shamasundar, G. Venugopala Rao and B.M.Sachin "Endurance testing and FE analysis of four wheeler automobilestabilizer bar" SAE technical papers 2009

6) Emanuele Guglielmino ,Tudor Sireteanu ,Charles W.Stammers ,Gheorghe Ghita ,Marius Giuclea "Semi-active Suspension Control", Springer-Verlag London Limited, Library of Congress Control Number: 2008927606, British Library Cataloguing in Publication Data, ISBN 978-1-84800-230-2,2008,

7) Adrian Chapple, Dr. Adam Towse, "Delivering World Class Chassis Design", ThyssenKrupp Automotive Tallent Chassis Ltd, Altair Engineering 2007

8) L. N. B. Gummadi ,H. Cai , " Bushing Characteristics of Stabilizer Bars" SAEPaper Number: 2003-01-0239,

9) SAE Spring Committee, "Spring Design Manual", 2nd Ed.,SAE, pp. 215-267, 1996