Analysis on Stability Bar

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A B S T R A C T
Vehicle anti-roll bars are suspension components used for limiting body roll angle. They have a direct effect on the handling characteristics of the vehicle. 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. Finite Element Analysis (FEA) can be effectively used in design analysis of anti-roll bars. However, due to high number of repeated design analyses, the analysis time and cost problems associated with the use of general FEA package programs may create considerable disadvantages in using these package programs for performing anti-roll bar design analysis.

In this study, an automated design program is developed for performing design analysis of vehicle anti-roll bars. The program is composed of two parts, the user interface and the FEA macro. The FEA macro includes the codes for performing deformation, stress, fatigue, and modal analysis of anti-roll bars in ANSYS 10. The user interface, which is composed in Visual Basic 8.0, includes the forms for data input and result output procedures. By the developed software, the FEA of the anti roll bars is simplified to simple data entry via user interface. The flow of the analysis is controlled by the program and the finite element analysis is performed by ANSYS at the background. The developed software can perform design analysis for a wide range of anti-roll bars: the bar centerline can have any 3D shape, the cross section can be solid or hollow circular, the end connections can be of pin or spherical joint type, the bushings can be mounted at any position on the bar with a user defined bushing length. The effects of anti-roll bar design parameters on final anti-roll bar properties are also evaluated by performing sample analyses with the automated design program developed in this study.

Introduction
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.

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 tensional resistance. But it resists relative movement between the bar ends, such as shown in Figure 1.

Figure 1 - Antiroll/Sway Bar

The bar’s tensional stiffness or resistance to twist determines its ability to reduce such relative movement and it’s called as "roll stiffness".
The main goal of using anti-roll bar is to reduce the body roll. Body roll occurs when a vehicle deviates from straightline 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. Body roll also occurs when a wheel crosses a bump at one side only, which was the case in Figure 2.

![Figure 2 - A vehicle experiencing body roll during cornering.](image)

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. Thus, the driver cannot drive the vehicle with confidence. Second reason is its effect on the camber angle of the tires, which is the angle between the central plane of symmetry of the wheel and the vertical plane at the center of the contact patch. 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. The smaller the contact patch of the tire, the less traction exists against the road surface. Therefore, body roll should be prevented. The first way to prevent body roll is to eliminate its source, roll moment. This moment can be reduced by increasing the roll center heights of the front and rear suspensions. But, this will cause considerable lateral wheel displacements during bump and rebound with track variations during operation. Another negative effect is the higher camber angle change. Another method for preventing excessive body roll is to use stiffer suspension springs, thus making it harder for the suspensions to move in opposite directions at the same time. This however, reduces the ride comfort. A compromise solution is to use softer suspension springs to provide ride comfort, lower roll centers to avoid lateral wheel displacement and anti-roll bar(s) to reduce body roll.

Anti-roll bars serve two key functions. First they reduce body roll, as explained above, and second provide a way to redistribute cornering loads between the front and rear wheels, which in turns, gives the capability of modifying handling characteristics of the vehicle. This can be done by arranging the roll stiffness’s of the anti-roll bars at the front and rear suspensions. If a firmer anti-roll bar is installed at the front, then the distribution of lateral load transfer increases toward the front tires, since a firmer anti-roll bar allows less deflection, thus transfers lateral loads at a faster rate. And the overall result is additional under steer effect. Adversely, increasing roll stiffness at the rear by using firmer anti-roll bar will create a overseer effect. Thus, anti-roll bars are also used to improve directional control and stability.

**Introduction to FEM**

Steps involved in FEM

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**Modeling the Anti-Roll Bar**

In case of creating the model in ANSYS, “from the bottom up” method is used. First of all, key points, which are the “lowest-order” solid model entities, must be created. Then these key points are connected with straight lines which are then connected to each other with fillets. The final step in creating the bar centerline is to combine all these lines and fillets. Now the bar centerline can be meshed with BEAM189 elements. However, there is still a problem, which is locating the bushings on the bar. In order to model the bar correctly nodes are needed at the midpoints of the bushings. Actually, this problem can be solved by creating hard points at the bushing locations on the bar centerline and then meshing the line. The program will automatically create nodes at hard point locations. But, this method can only be used with free meshing. In an ANSYS analysis, mapped meshing, if it is possible, is always preferred to free meshing. A free mesh has no restrictions in terms of element shapes, and has no specified pattern applied to it while a mapped mesh is restricted in terms of the element shape it contains and the pattern of the mesh. Here, the concern is on the pattern of the mesh and it’s preferred to have elements with equal size. Thus, another method is employed by dividing the centerline into 3 lines, using the bushing positions as division points. During meshing of these lines the ANSYS program will define a node at the ends of all three lines and merge the coincident nodes. Thus, nodes at the midpoints of both bushings are created.

Before meshing the line, some attributes to be associated with the generated beam elements must be defined. These attributes include:

- The beam element type
- The material set number
- The cross section ID
- The real constant set number
- The orientation key points

The first three of these attributes are defined as: Element Type 1, Material Set Number 1 and Cross-section ID 1, for all three lines.
No real constants are required for the BEAM189 so the fourth attribute is not defined. The orientation key points are used to determine the orientation of the cross section with respect to the beam element axis. This property is actually not used for circular cross-sections since a circle’s orientation is meaningless. But an orientation key point is asked by the program whatever the cross-section is. Defining a key point at a far distance in y axis will solve this problem. There exists a single node at the intersection points of lines, which means the model, will behave like single beam. Number of elements used for meshing is an important issue. The results of some sample analyses performed with different number of elements are discussed in Section 3.2.5. As a result of the discussion, use of an element number between 100 and 200 is suggested. This case may change according to the geometry but 200 elements will guarantee a good solution and using more elements will only increase the analysis time. The real shape of the anti-roll bar; the bar ends. This is a simplification which is based on the fact that anti-roll bars have spherical joints or bar and bayonet type (pin) connections at the end points both of which are moment free. Thus, stresses at the ends are not an issue. This assumption is validated with some sample analyses. It should be also noted that, the end connections -either pin or spherical joints- can be designed using the reaction force data without requiring finite element solution.

![Figure 3.10 - Detailed View of Anti-roll Bar Bushing Model](image)

### 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 its preferred to review the results of these two loadings together, SOLUTION processor must be selected for applying loads. A load step can be defined as one set of loading conditions for which the solution is obtained. By using multiple load steps the structure’s response to each loading condition is isolated. There exists another loading for determination of the natural frequencies and the mode shapes of the bar. However, this is a different analysis type - modal analysis - and cannot be solved as a third load step of the static Analysis.

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.

Where F is Force Applied, fA is Deflection and Angle of deflection.

### Stability solid bar analysis report

#### Input data

**Geometry (Dimensions are in mm)**

**Keypoints Coordinates**-

1) 550 0 230
2) 460 0 0
3) -460 0 0
4) -550 0 230

**Connections**-

Bushing Type = x-movement free
Location on +x side = 390 mm
Bushing Location on -x side = 390 mm
Bushing Stiffness = 1500 N / mm
End Fixture Type = Spherical Joint

**Material** -

Material Type= SAE 5160
Modulus of Elasticity = 206000 MPa
Poison’s Ratio = 0.27
Yield Strength =1100MPa
Ultimate Tensile Strength=1400 MPa
Endurance Limit=706 MPa
Density =7800 MPa

**Analysis Type**-

Linear Analysis
υ = 0.27

**Cross Section**-

Type= Hallow Circular
(R) = 10 mm
(I) = 08 mm

**Bar Properties**-

Roll Stiffness= 432.0 Nm/deg
= 45.08 Nm/deg
= 40. 98 N/mm
Length = 1394 mm
Mass = 1.872 kg
Fillet Radii:
50
50

Bushing Locations:
\[ x_1 = +390 \]
\[ x_2 = -390 \]

Mesh Density:
Number of Elements = 100

Loading:
Load Applied = 1000 N
Max. Suspension Deflection = 50 mm

Endurance Limit Modification Factor:
ksize = 1.003
ksurface = 0.661 (Cold Drawn)
kmisc = 1

Maximum Stress Strain Result:
Maximum Principal Stress = 606 MPa
Equivalent Stress = 683 MPa
Maximum Principal Strain = 0.319 %
Equivalent Strain = 0.390 %

Fatigue Life:
N = 93595 Cycles

Natural Frequency:
1st Natural Freq. = 95.94 Hz
2nd Natural Freq. = 71, 10 Hz
3rd Natural Freq. = 88, 82 Hz
4th Natural Freq. = 109, 78 Hz
5th Natural Freq. = 165, 28 Hz
Roll Stiffness = 438.6 Nm/deg

Validatation:
Loading = 1000 N
Permissible Deflection = 27 mm

\[ 2 \text{ times the deflection} = 27 \times 2 = 47 \text{ mm} \]

Roll Stiffness \((K) = \frac{\text{Load} \times \text{Yield Strength}}{\left[ 1/\tan \left( \text{deflection/length} \right) \right]} \]

\((K) = 1000 \times 1100 / \left[ 1/\tan \left( 27/550 \right) \right] \)

\((K) = 432.0 \text{ Nm/deg} \)

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

- 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. The size factor used for endurance limit modification is also affected from the diameter of the bar.
- 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. The size factor is also adversely affected from the outer diameter of the anti-roll bar.
- Increasing the bushing stiffness increases the anti-roll stiffness. The stresses are again increased.
- Constraining the x movement of the within the bushing increases the roll stiffness if the amount of this displacement is high in the unconstrained case.
- Locating the bushings closer to the center of the bar increases the stresses at the bushing locations while roll stiffness of the bar decreases.
- 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.
- Required roll stiffness can be obtained with a lower weighting bar by changing the bar by changing the bar material.

Recommendations for Future Work
The following recommendations can be made for future studies:

- In order to obtain desired anti-roll bar properties, the automated design software must be run with trial parameters. Optimization can be utilized for finding the value of input parameter (bar diameter) to obtain a specified property (roll stiffness) while keeping the other input parameters same.
- In some situations zooming may be very useful in post-processing the results. Thus, zooming capability can be made available in the post-processor.
- Although their use is not common, the analysis of bars with variable cross-section can be added to program capability.
- A database of bushing materials with material properties can be made available to user.
- Fatigue analysis can be performed by using real road data.
- Automated design software can be developed for other machine components.
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