Structural Analysis and Validation of Anti Roll Bar

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***Abstract* — Anti-roll bar is the part of an automobile suspension system that connects the left and right wheel members through short lever arms and is clamped to the vehicle chassis with rubber bushes. It forces each side of the vehicle to lower, or rise, to similar heights, to reduce the sideways tilting (roll) of the vehicle on curves, sharp corners, or large bumps. It also resists roll or swaying of the vehicle which occurs during cornering or due to road irregularities. In this study FEA is carried out to perform structural analysis of Anti-roll bar using ANSYS Parametric Design Language (APDL). The proposed work is focus on to prepare the methodology for analysis of the anti-roll bar, and validate it by analytical method.**

Keywords— Structural analysis, APDL.

1 Introduction

Anti-roll bar is also known as anti-sway bar, sway bar or stabilizer bar. It is the part of many automobile suspension systems, which helps to reduce the body roll of a vehicle during fast cornering or over road irregularities. It is a rod or tube which is usually made of steel, that The bar's torsional stiffness (resistance to twist) determines its ability to reduce body roll, and is named as “Roll Stiffness” [1]. 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. The anti- roll bar, as being a suspension component, is used to improve the vehicle performance [3]

Fig.1. Fitment picture of Anti-roll bar on vehicle [3]

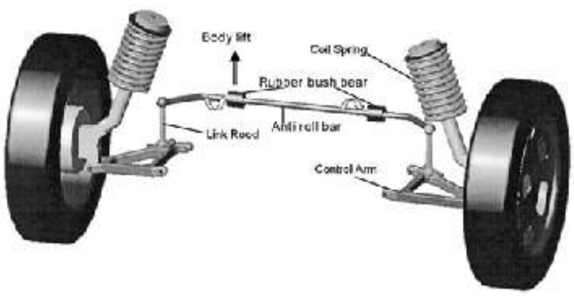

Fig.2. Typical Anti-roll bar

Principle of working

The anti- roll bar is a torsion spring that resists body roll motions. It is usually constructed out of a cylindrical steel bar. It is formed as “U” shape that connects to the body at two points and at the left and right side of the suspension. If the left and right wheels move together, the bar rotates about its mounting point but if the wheels move relative to each other, the bar is subjected to torsion and forced to twist [4]. The ends of the bar are connected to an end link through a flexible joint. The anti- roll bar end link connects in turn to a spot near a wheel which transfers forces from a heavily loaded axel to the opposite side. So in anti- roll bar forces are transferred from heavily loaded axle to the connected end link via a bushing, to the anti- roll bar via a flexible joint to the connected end link on the connected end link on the opposite side of the vehicle to the opposite axel.

Connections of Anti-roll bar

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 [1].

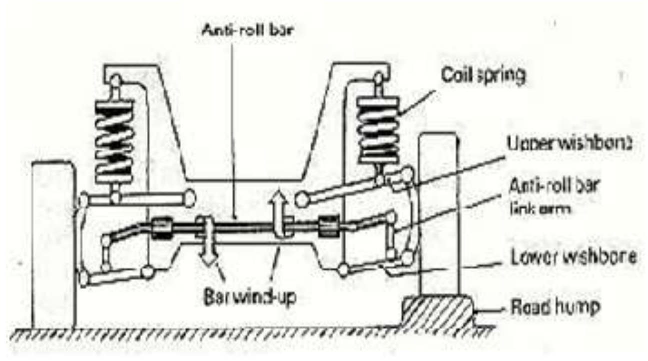

Fig.3. An anti-roll bar attached to double wishbone type suspension. (The vehicle is crossing over a road bump on one side)[1]

OBJECTIVES

An objective of this proposed work is to do the structural analysis of an existing anti-roll bar using APDL, obtained the ANSYS results for deformation and equivalent stresses and calculate the stresses by analytical method and validate the result.

METHODOLOGY

From the literature review, Anti-roll bar and its design study is carried out. In this study Finite element modeling and analysis of the existing anti-roll bar is performing using the ANSYS software. Initially the parametric model of anti-roll bar is prepared in ANSYS using the ANSYS Parametric Design Language (APDL) and analysis of anti-roll bar is performed. By analytical method the stresses with the ANSYS results will be found out and compare. Experimental validation of the analysis will be done using parametric model. Finally the optimization of the Anti-roll bar is performed using the above validated parametric model. The proposed work is for the overall methodology for the analysis of anti-roll bar and its optimization.

Problem Definition

The geometry of the Anti-roll bar is depends on the design and location of other chassis components. If the design of chassis components is to be changed then the design of Anti-roll bar is changed because it is simple as compared to design other components. But it is to be finalized for best parameters, and hence there is need to work on such parameters those will reduce the weight of the anti-roll bar without failure i.e. stress generated should be within permissible limit.

Anti-roll bar model information

Parameter	Initial Design	Unit
Length **L**	1100	mm
Width	230	mm
Outer diameter $\mathbf{D_o}$	21.8	mm
Thickness	2.9	mm
Bush Length	40	mm
Position of Bush from center	±390	mm

Analysis of existing Anti-roll bar

The structural analysis of the existing anti-roll bar is carried out using the APDL i.e. ANSYS Parametric Design Language, following model is prepared using input listing method and the boundary conditions and load is applied as shown in figure

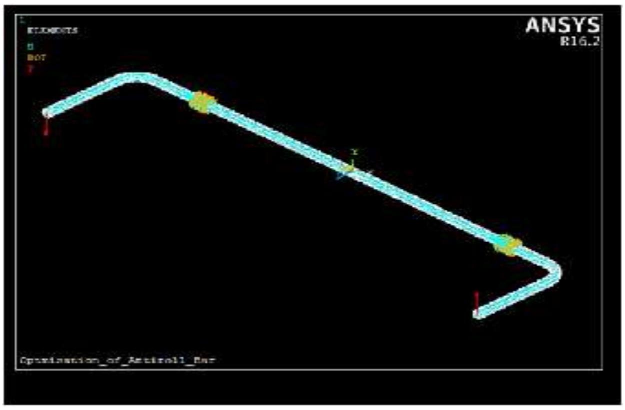

Fig.4. Meshed model of anti-roll bar with boundary conditions

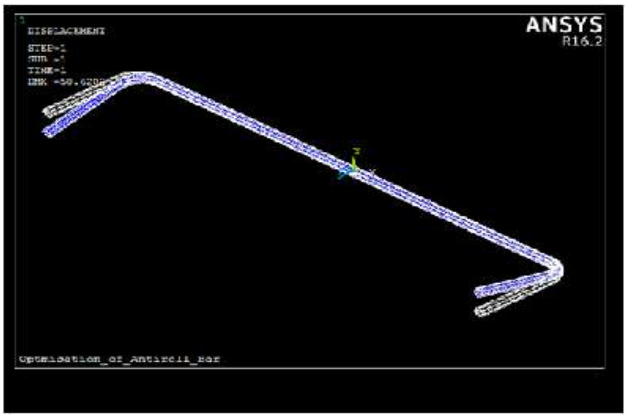

Fig.5. Displacement of Anti-roll bar in Y-direction

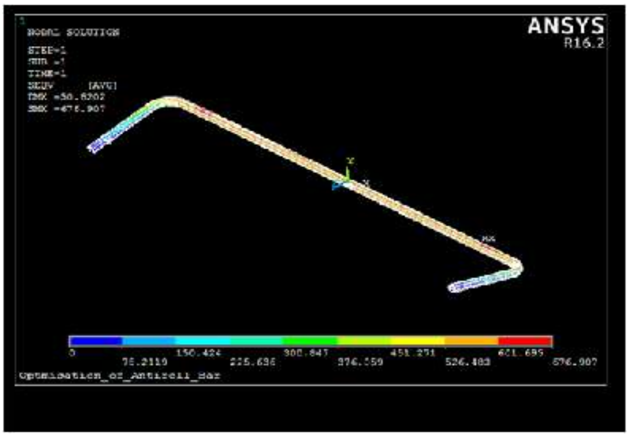

Fig.6. Equivalent Stresses in Anti-roll bar

By structural analysis the von mises stresses obtained as 676.907 MPa and the deformation is 50.62 mm

Analytical method

The transmission shafts are subjected to axial tensile force, bending moment or torsional moment or their combinations. Most of the transmission shafts are subjected to combined bending and torsional moments. When the shaft is subjected to axial tensile force, the tensile stress is given by,

$$\sigma_t = \frac{4P}{\pi d^2}$$

When the hollow shaft is subjected to pure bending moment, the bending stresses are calculated as

$$\sigma = \frac{32\, D_o M}{\pi(D_o^4 - D_i^4)}$$

When the shaft is subjected to pure torsional moment, the torsional stresses are calculated as

$$\tau = \frac{16 D_o T}{\pi(D_o^4 - D_i^4)}$$

In our case the central portion of the antiroll bar is subjected to bending and torional moments both. So according to the Mohr's circle , the principal stresses are calculated as

$$\sigma_1, \sigma_2 = \frac{\sigma}{2} \pm \sqrt{(\sigma/2)^2 + \tau^2}$$

Putting the values σ and τ calculated above, we get

$$\sigma_1, \sigma_2 = \frac{16Do}{\pi(Do^4 - Di^4)}\left(M \pm \sqrt{M^2 + T^2}\right)$$

$$\sigma_1 = \frac{16Do}{\pi(Do^4 - Di^4)}\left(M + \sqrt{M^2 + T^2}\right)$$

$$\sigma_2 = \frac{16Do}{\pi(Do^4 - Di^4)}\left(M - \sqrt{M^2 + T^2}\right)$$

The bending moment for the Anti-roll bar case study is the multiplication of force (2000 N) and the distance of bush from the corner (140 mm)

Bending moment (M)

$= 2000 \times 140 = 28 \times 10^4$ Nmm

The twisting moment for the Anti-roll bar case study is the multiplication of force (2000 N) and the width (230 mm), that is causing the twisting in the bar

Twisting moment (T)

$= 2000 \times 230 = 46 \times 10^4$ Nmm

The maximum and minimum principle stresses are given by

$$\sigma_1, \sigma_2 = \frac{16Do}{\pi(Do^3 - Di^3)}\left(M \pm \sqrt{M^2 + T^2}\right)$$

$$\sigma_1 = 554.94\ Mpa$$

$$\sigma_1 = -175.263\ Mpa$$

The von mices stress is to be calculated for obtaining the combined result of bending stresses and torsional stresses

$$\sigma_{Von} = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2}$$

$$\sigma_{Von} = \sqrt{554.94^2 + (-175.263)^2 - [554.94 \times (-175.263)]}$$

$$\sigma_{Von} = 660.25\ Mpa$$

This is the value of the maximum stress as per Von Mises criterion.

Conclusion and Future work

The von mises stresses obtained from ANSYS are 676.907 MPa and by analytical method the von mises stresses value is 660.25 MPa. i.e. the percentage error is 2.46%. This means the analysis process of parametric modeling is correct. We are still working over the project for optimization using APDL. We will further do the validation by experimental method.

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