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# Non-Linear Finite Element Analysis of Rubber Bush for 2-Wheeler Rear Shock Absorber for Prediction of Fatigue Life

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**ABSTRACT**

*Suspension bushing is an important link in shock absorber and it must be stiff enough to maintain alignment of suspension under loads. It is crucial for ride and handling characteristics and must be durable under highly variable loading. Such elastometeric bushings exhibit non-linear behaviour depending upon excitation frequency, amplitude and level of pre-load. Designing of Rubber bush is performed using suitable modelling software and FEA analysis is performed to predict behaviour of component against load applied. The different alternatives are suggested to get better model to satisfy endurance limit proposed by Gabriel India Ltd, Nashik. To satisfy testing criterion of Rubber Bush necessary design changes and relevant experimentation validation has carried out in GIL, Nashik Testing Lab.*

***Keywords****-Rubber bush, the maximum principle strain, Non-linear FE analysis, Fatigue life, Endurance test*

## I. INTRODUCTION

Rubber Bushing plays an important role in vehicle suspensions, with three main functionalities. First, it connects the structures to transmit the weight of vehicle and loads. Second, it attenuates bumpiness from rough roads and ensures comfort. Third, it provides good control stability. Material parameters of rubber are very important for design of bushing. Simple material tests, such as uniaxial test and biaxial test, can be used to obtain some parameters of the constitutive models. But, all bushing suppliers failed to provide these data appropriately. The elastomer composites used in these applications have to sustain high deformations under applied static load, and energy losses from hysteresis under cyclic deformation. Thus, elastomeric compounds used in bushings often are much harder than compounds used in mounts. [5]

In an effort to achieve significant economies of scale, most major automotive companies design global architectures that can be sold in every region of the world. Meeting the needs of consumers in very different markets creates significant challenges for noise vibration and harshness (NVH) engineers. In countries where road conditions are poor, maximizing overall system durability is critical. At the same time, consumers in developed markets demand excellent ride quality and handling performance. [3]

Rubberlike material exhibits a highly non linear behaviour characterized by hyper elastic deformability and incompressibility. Rubber withstands very large strain from 100% to 1000% without any permanent deformation. The fatigue life is predicted by combining test of material properties and finite element analysis (FEA). Analytical and FEA results are validated by Eye Bush Durability test in laboratory. Thus attempt is made to obtain a rubber bush model with considerable durability limit of $3 \times 10^5$ cycles as specified by Gabriel India Ltd, Nasik. A double tapered rubber bush used in 2-wheeler rear shock absorbers is considered in this study. Figure 1 shows three dimensional diagram of this rubber bush assembly.

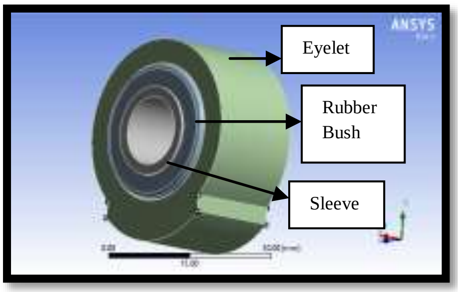



***Fig. 1 The three dimensional diagram of rubber bush assembly***

## II. NATURAL RUBBER MATERIAL PROPERTIES

### 2.1 The hyperelastic model of natural rubber

The different types of models are used for defining hyperelastic material such as Mooney Rivlin, Yeoh, and Ogden etc. Yeoh model is used for this study. The Yeoh model is known as the third-order reduced polynomial form describes isotropic incompressible rubber-like materials. The strain energy density function W, used in most of the general finite element programs able to handle hyperelastic materials is given by the following equation

$W(I_1 I_2 I_3) = \sum_{i,j,k=0}^{\infty} C_{i,j,k}(I_1-3)^i(I_2-3)^j(I_3-1)^k$ (1)

If total incompressibility is assumed ($I_3$= 1) the equation (1) changes to

$W(I_1 I_2) = \sum_{i,j,k=0}^{\infty} C_{i,j,k}(I_1-3)^i(I_2-3)^j$ (2)

Where $C_{i,j}$ are unknown constants

The third order of deformation material is found by taking terms that include $I_1, I_2, {I_1}^2, {I_2}^2$ and $I_1 I_2$.The dependence on the second invariant is very weak for carbon-black filled natural Rubbers. Thus, by leaving out terms in (2) that include $I_2$ it is possible to get the strain energy function for the Yeoh model, which gives a good fit to experiment carried out on filled rubbers. Consequently, this strain energy function is given by the following expression

$W = C_{10}(I_1-3) + C_{20}(I_1-3)^2 + C_{30}(I_1-3)^3$ (3)

The rubber material parameter $C_{i,j}$ can be obtained approximately from the initial shear modulus. The approximate relations are given by the following equations

$C_{10} = \frac{G}{2},\ C_{20} = -\frac{G}{20}, C_{30} = \frac{G}{200}$. (4)

Where G is initial shear modulus. The parameters $C_{10}$ and $C_{30}$ influence the behaviour of the rubber at low and high strain rate, respectively. [7]

### 2.2 The Fatigue properties of natural rubber [1]

Generally two methods are used to character the fatigue properties of the material; one is the fatigue life equation and curve of stress amplitude vs. fatigue life (S–N equation and curve), second is the fatigue life equation and curve of strain amplitude vs. fatigue life (e–N equation and curve). Due to hyperelastic nature of natural rubber, the natural rubber's curve of strain vs. stress shows highly nonlinear characters. So the e–N equation and curve were selected to character the fatigue properties of the natural rubber material. Fatigue tests of 3-D dumbbell specimen were performed by Qian Li et.al to evaluate the fatigue properties of natural rubber. A dumbbell specimen with sectional dimension shown in Fig.2 was used for the fatigue tests. Fatigue tests were conducted in an ambient temperature of 25°C. The dumbbell specimen was acted with constant amplitude, sinusoidal cycling displacements with a mean displacement of zero, and frequency of 10 Hz. With increasing cycles in the initial phase, the maximum load decreased little by little. When the crack grew over the critical size, the maximum load decreased suddenly and the final failure reached. The fatigue failure was defined as the number of cycles at which the maximum load dropped by 20%. The e–N in the double logarithmic coordinates can be treated as a line. So the fatigue life equation of the natural rubber can be expressed as:

$\varepsilon^m N = C$ (5)

Where m and C are the moduli determined by test of material properties. m and C were determined by the fatigue life test results using the least square method as 6.211 and 323072, respectively. So the fatigue life equation of the natural rubber can be expressed as:

$\varepsilon^{6.211} N = 323072$ (6)

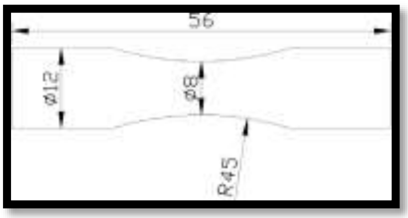



***Fig. 2 The sectional dimension of the dumbbell specimen***

Eq. (6) can be transformed as:

$m \log \varepsilon + \log N = \log C$ (7)

So the fatigue life equation of the natural rubber in the logarithmic coordinates is

$\log \varepsilon = 0.887 - 0.161 \log N$ (8)

The total principal strain at the critical region determined from the FEA was used for evaluating the fatigue damage parameter of the natural rubber. The fatigue life of the rubber bush at given load had obtained by substituting the maximum total principal strains into the fatigue life equation of the natural rubber material.

## III. FINITE ELEMENT ANALYSIS OF RUBBER BUSH ASSEMBLY MODEL

General method of FE analysis was performed to carry out non linear analysis of rubber bush assembly. Initially base model is used for the FE analysis. Figure 3 shows the general shape of rubber bush with ID 16mm; OD 25mm and hardness 60 Shore A.

***Fig. 3 The general shape of rubber bush***

ANSYS 14.5 Workbench is used for analyzing the Rubber bush assembly mode. There are three distinct domains of materials, first is outer eyelet with material aluminium alloy, second is middle layer with natural rubber material (as per GIL standard) and inner sleeve with Structural steel. Meshing is performed in Hypermesh 11.0 with 1st order brick element. For base model initial shear modulus is calculated from Classical Theory of Bushing [2] as 0.899 MPa. Total assembly load is used for determining principle stress in FEA analysis. As shock absorber is mounted to vehicle body through eyelet inner surface of sleeve is treated as fixed support.

Total assembly load (GIL Standard) = 2364.99 N

The principle strains at critical regions of base model are as shown in the figure 4. These regions are critical regions where crack tends to occur. This value of maximum principle strain is used in eq. 8 to evaluate the fatigue life of rubber bush.

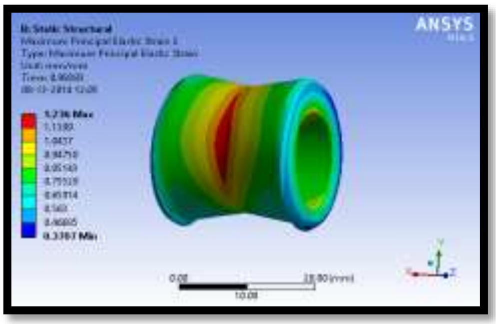



*Fig. 4 Maximum principle strain in mm/mm (60 Shore A)*

From the figure 4, it can be observed that maximum strain zone is occurring at central region of double tapered.

By referring formula (8) fatigue life of rubber bush base model is predicted as:

***Table 1. Analytical Calculation for base model***

| Strain(ε) (mm/mm) | Log e | Log N | No of cycles |
|---|---|---|---|
| 1.236 | 0.0920 | 4.9377 | 0.867 x $10^5$ |

From the above data is clear that base model had failed to achieve endurance criterion of 3x$10^5$ cycles. The main reason of failure from FE analysis is higher value of principle strain at double tapered region as shown in figure 4. Therefore to get desired output different alternatives are tabulated as follows

***Table 2. Redesign Alternatives***

| SN | Model | Material Hardness (Shore A) | Design | Remark |
|---|---|---|---|---|
| 1 | Base model | 60 | Double Taper | Failed in subject test |
| 2 | New | | | |
| | A | 60 | change shape | Change in eyelet and bush design, increase in mfg cost<br>Time consuming process |
| | B | Change | Double Taper | Need not to change Shape, Change in rubber material properties,<br>Vary hardness and check for FEA result |

By considering feasibility of alternatives change in hardness in the specified range of GIL standards seems to be more appropriate. So that alternative B has selected for future work.

## IV. FINITE ELEMENT ANALYSIS OF ALTERNATIVE MODELS

As per GIL standard hardness for bushing applications can be varied in the range of 45 Shore A to 75 Shore A. So that FE analysis to get minimum principle strain of models with change in hardness values had performed. Analytical fatigue lives of different models by using FE results are tabulated.

***Table 3. Result Summary***

| Model | Hardness (Shore A) | Shear modulus Mpa | FEA result Max princ. strain mm/mm | Fatigue life (Analytical) | Remark |
|---|---|---|---|---|---|
| Base Model | 60 | 0.899 | 1.243 | 0.894 x$10^5$ | < 3x$10^5$ failed |
| New (B) | Change | | | | |
| 1 | 65 | 1.066 | 0.9422 | 4.58 x$10^5$ | >3x$10^5$ |
| 2 | 70 | 1.264 | 0.6920 | 3.23 x$10^6$ | >3x$10^5$ |
| 3 | 75 | 1.489 | 0.6800 | 3.55 x$10^6$ | >3x$10^5$ |

From the above result table it is clear that base model was failed in the subject test and model with hardness 65, 70 and 75 shore A satisfy desired endurance criterion of 3x$10^5$ cycles. Thus to validate results testing was performed on base model as well as on model 2 with hardness 70 shore A.

## V. EYE BUSH ENDURANCE TEST

The test was performed on Stroking Endurance RR machine at GIL, testing lab. The total assembly load on specimen without oil was applied with a speed of 150 rpm/ 2.5 Hz. Test setup had run for 3 x$10^5$cycles with full stroke of shock absorber. After completion of test, specimen was dismantled from setup and kept for observation.

Acceptance Criteria for this test is that there should not be any physical damage to parts. The experimental setup is as shown in figure 5.

*Fig. 5 Endurance test setup*

After completion of Endurance test the results shown in the figure 6 and figure 7 had observed.

***Fig. 6 Test result for Base Model (Hardness= 60 shore A)***

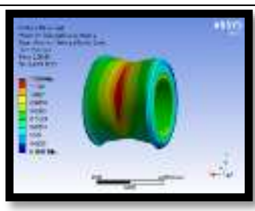
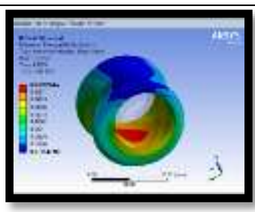
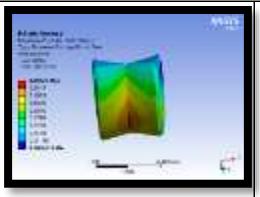
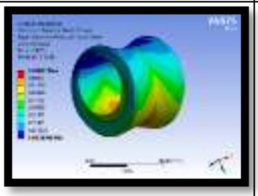
***Fig. 7 Test Result for Model 2 (Hardness= 70 shore A)***

From figure 6 we can observe abnormal cut marks and wear and tear of rubber bush at central region of double taper, thus this model had not satisfied endurance criterion so that it had failed in a subject test. For Model 2 with Hardness 70 Shore A, we had not found any cut marks on the critical region of rubber bush model. Thus this model has passed in a subject test by fulfilling acceptance criterion proposed by GIL, Nashik.

**6.1 Comparison of FEA and Testing Results**

For the sake of simplicity comparison of analytical solution with aid of FEA results and testing results are tabulated in subsequent section.

***Table 6. Overall Comparison Sheet***

| Hardness Shore A | FEA Results | Analytical Life (No of Cycles) | Testing Results |
|---|---|---|---|
| (Base model) 60 |  | $0.89 \times 10^5$ |  |
| (Model 1) 65 |  | $4.58 \times 10^5$ | |
| (Model 2) 70 |  | $3.23 \times 10^6$ |  |
| (Model 3) 75 |  | $3.55 \times 10^6$ | |

The above summary sheet gives brief idea about analytical fatigue life prediction by using FE analysis results and eye bush testing results resemblance. FE results are validated by testing results. Thus from the above table we come to know that with increasing hardness strain value decreases and thus increasing fatigue life of the rubber bush

## VI. CONCLUSIONS

- Finite element analysis of Rubber bush gives the behaviour of material on application of load.
- Higher value of strain was a prime reason for failure of base model.
- As hardness increases, strain at the taper portion goes on reducing and hence we have got increasing fatigue life of component which is verified by the Endurance test.
- Analytical solution with aid of FEA result matches well within a range of testing results.
- The proposed method can reduce product design cycle, decrease design and production cost and helps to recommend better alternative for given application.

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