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Parametric Optimization to Reduce Stress Concentration at Corner Bends Of Solid and Hollow Stabilizer Bar

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Abstract

The function of stabilizer bars in motor vehicles is to reduce the body roll during cornering. This project looks into the performance of stabilizer bar with respect to their stress variations at corner bends and weight optimization. The focus is on the stress concentration at the corner bends of a stabilizer bar, that is designed for an automotive vehicle, which is reduced by optimizing the shape of the critical regions in two types of stabilizer bars, one is solid and other is hollow. In order to do this, parameters which constitute the geometry of the stress concentrated regions are determined. The effect of these parameters on stress concentration is evaluated by using Design of Experiments (DOE) approach and parametric correlations. Possible design options and their corresponding mass and maximum equivalent stress values are obtained by using Finite Element Analysis. The results are assessed by means of response surfaces generated by FEA software. FE analyses showed that it is possible to decrease the maximum equivalent stress at the critical regions for solid and hollow stabilizer bar by 11% and 12% respectively with a mass increase of 3.75% and 3.45% respectively. Transition form that gives optimum stress concentration is determined.

Keywords- Stabilizer Bar; Stress Concentration; Design of Experiments Method; Parametric Optimization; Finite Element Analysis.

1. INTRODUCTION

The function of stabilizer bars in motor vehicles is to reduce the body roll during cornering. The body roll is influenced by the occurring wheel load shift and the change of camber angle. Decisive is the steering performance which may be purposefully adjusted towards understeer or oversteer when designing the stabilization. So the stabilizer bars increases the travelling comfort and the driving safety to a considerable extent [1]. Stabilizer bars also called as Antiroll bar or Sway bars. During the vehicle service life, dynamic forces produce dynamic stresses which may cause fatigue failure of a stabilizer bar which is one of the basic parts of a suspension system used to reduce the roll tendency of the vehicle body. Recent studies showed that satisfying the static strength conditions does not mean that the mechanical part has infinite fatigue life [2]. Because of this, during the design process of a mechanical element, it is vital to take the fatigue life assessment into account. As long as the material and/or the manufacturing process have not been changed, tensile strength and therefore fatigue strength of a mechanical part cannot be altered. Reducing stress concentration at the critical regions of a mechanical element is an effective alternative way to obtain a longer fatigue life. On the other hand, one of the main targets to be reached in the design of vehicle suspension components is to keep the unsprung mass as small as possible with a homogenous stress distribution on the part body [3]. Therefore an optimal design process of the mechanical part is inevitable.

In this study, an anti-roll bar which is used in the front axle suspension of a vehicle is redesigned to minimize the stress concentration at the corner bends. In order to do this, regions that are under stress concentration during the roll motion of the vehicle body are determined by using finite element analysis. Geometric parameters which constitute the form of these regions are assigned. In the light of the results obtained from the stress analysis that were carried out for the different values of these parameters, stress and mass alteration are assessed by using Design of Experiments (DOE) approach. Design of experiments (DOE) is one such well-defined area of operation research. This method enables one to analyze the experimental data and build empirical models to obtain the most accurate representation of the physical situation. The effects of these modifications on stress concentration and anti-roll bar mass are studied. Response surfaces for maximum von Misses stress and the mass of the anti-roll bar are also constructed by ANSYS Workbench™ V13.0 commercial finite element software package.

2. FINITE ELEMENT ANALYSIS OF STABILIZER BAR

2.1 FE Models

A typical ANSYS analysis has three distinct steps, first build the model, second Apply loads and obtains the solution and third Review the results. After starting the ANSYS session, the bar will be meshed with SOLID187 which is a higher order three dimensional solid element which has quadratic displacement behaviour and is well suited to model irregular meshes. The element is defined by 10 nodes having three translational DOF at each node. Actually, the anti-roll bar can be analyzed with solid, beam or shell elements (in case of hollow cross-section). However in this study solid elements are preferred. The solid elements are used to create a mathematical three dimensional idealization of a 3-D structure. They offer computationally efficient solutions when compared to beam and shell elements. For the limited deformation of the bushings, behaviour of the material was assumed as linear isotropic. Anti-roll bar will be manufactured from 50CrV4 (51CrV4) spring steel that is suitable for highly stressed spring design. Basic mechanical properties of this material after quenching are given in Table 1.

Table 1 Mechanical properties of the 50CrV4 (51CrV4) steel

Standard	EN 10 083
Material number	1.8159
Modulus of elasticity, E (GPa)	200
Poisson's ratio, ν (-)	0.3
Yield strength (min), S_y (MPa)	800
Ultimate strength (min), S_{ut} (MPa)	1000

In the FE model, it was assumed that elastic bushings are made of rubber. In Report, modulus of elasticity is given as E_{max}= 0.1 (GPa) and Poisson's ratio $\nu \approx$ 0.5 for this material. For the limited deformation of the bushings, behaviour of the material was assumed as linear isotropic. The element types and material properties are consider same for all geometric shaped solids as well as hollow antiroll bars.

When designing tubular stabilizers, attention is to be paid to the fact that the weight must not be reduced at the cost of the component rigidity that may be roughly calculated by the equation bellow,

$$R = \frac{G_{mod} I}{\eta}$$

Where,

R - Rigidity G_{mod} - Rigidity modulus

I - Area moment of inertia η - Geometry matrix

Furthermore, it is important that the target weight reduction is not greater than 40 to 45 % of the weight of a comparable solid stabilizer bar. Weight savings going beyond this lead to no more acceptable wall thicknesses, thus to additional stresses caused by tube warping. These warping strains have an extremely negative impact on the lifespan of a component. The outer and inner diameters may be calculated using the following equations [4],

$$d_{i,tube} = d_{solid} \sqrt{\frac{1 - k^2}{2k}}$$

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$$d_{o,tube} = d_{solid}\sqrt{\frac{1+k^2}{2k}}$$

Where,

$d_{i.tube}$, $d_{o,tube}$- inner and outer tube diameter, d_{solid} - diameter of the solid stabilizer bar, k - Weight portion of the tube (k=0.6)

The next step in the analysis is generating a finite element model (nodes and elements) that adequately describes the model geometry. The geometry of any anti-roll bar, even with irregular shapes, can be created in ANSYS WORKBENCH. There are two alternatives exist at this stage, creation of the model in ANSYS Design Modular or importing the model from an IGES (Initial Graphics Exchange Specification) file. The import of the model from IGES file can only be performed in AUX15 processor of ANSYS. In case of creating the model in ANSYS, "from the bottom up" method is used. First of all, keypoints, which are the "lowest-order" solid model entities, must be created. Then these keypoints are connected with straight lines which are then connected to each other with fillets. Now the geometry imported option is used for the project. In that case geometry was created in regular commercial software CATIA V5 R20 and converted into IGES model to import in ANSYS WORKBECH 13.

A detailed CAD model of the anti-roll bar is given in Fig.1. The anti-roll bar is mounted to the vehicle body (H and H') and axle beam (F and F') via elastic bushings. The stress analysis is performed via ANSYS® Workbench™ V13.0 commercial finite element software. To build this model, CAD model of the anti-roll bar is meshed using SOLID187. For the optimization the joint stresses are not considered, so only the corner bends of the antiroll bars which are the major stress concentration areas are considered. Therefore only flat surfaces are taken for the loading conditions at arm ends of the stabilizer bar and displacement loads are given on these surfaces.

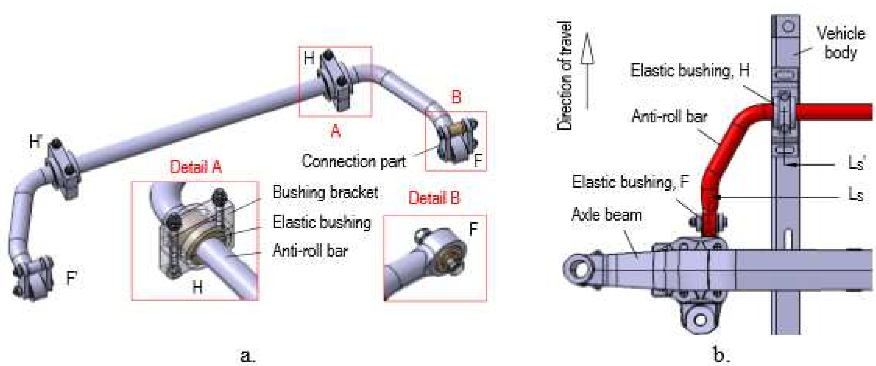

Fig.1 a. Elastic bushings of the anti-roll bar, b. Position of bushing points

2.2 Applying Boundary Conditions and Loads

The displacement constraints exist at two locations that is 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. 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, is shown in Fig.2.

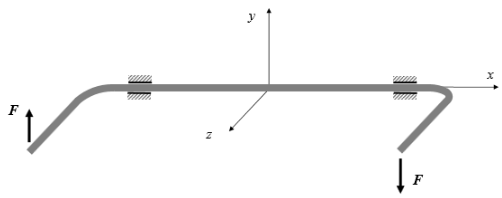

Fig.2 Load Step

For the second load step the force loads of the first load step are removed and displacement loads are given that is maximum suspension deflection are applied to the bar ends again in opposite directions. The analysis carried out for a completely reversed cycle and displacement loads with 40 mm applied at ends in opposite direction. Also the outer surface of rubber bush is completely constrained with surface to surface contact between stabilizer bar outer surface and rubber bush inner surface. The axial movement of the bar, set of center nodes at middle of stabilizer bar are coupled to a common reference point and axial movement is constrained. The stress analysis of the anti-roll bar is carried out for one cycle. For the analysis, the displacement $Z_1 = Z_2$ = 40 mm is applied on arm bearings F and F' in opposite directions.

2.3 Finite Element Analysis

The anti-roll bar analysis is conducted with the boundary conditions as vertical displacement 40 mm on both ends of the bar in opposite directions; the equivalent stress distribution on the bar is given in Fig.3. Here, it is observed that the stress near end of the bar is much lower than the maximum stress on the bar

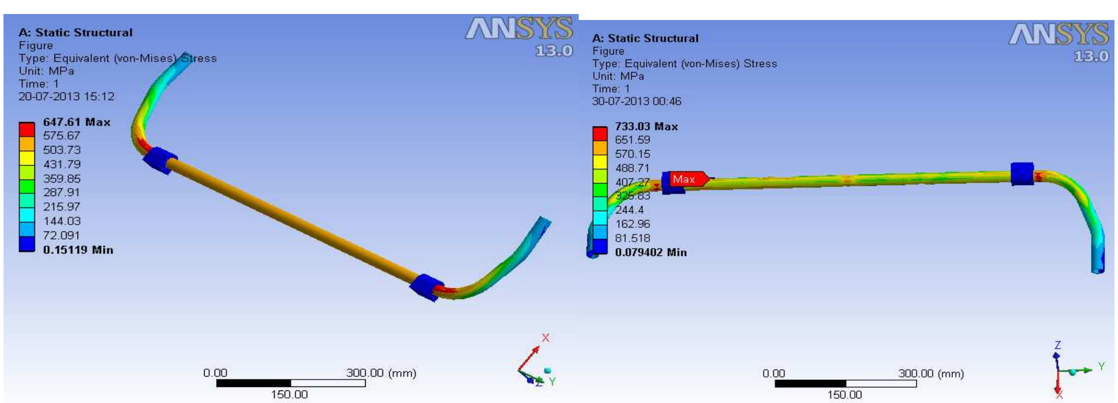

Fig.3 Equivalent stress distribution for u shaped curved solid and Hollow stabilizer bar

Table 2 Finite Element Model Properties

	Bush 1	Stabilizer bar	Bush 2
Solid bar			
Mass	5.4232e-002 kg	6.9709 kg	5.4232e-002 kg
Nodes	4023	62353	4023
Elements	700	13432	700
Hollow bar			
Mass	5.4232e-002 kg	3.5975 kg	5.4232e-002 kg
Nodes	927	3420	927
Elements	140	1724	140

The results of the stress analysis are also shown in Fig.3. According to these results, equivalent von Mises stress between the bearings H and H' is about σ= 550 (MPa). The maximum equivalent stress is determined at the bending section is $\sigma_{max} \approx$ 647.65 (MPa); which is 80 % of the yielding strength of material for solid bar and 90% for hollow bar. Therefore, application of the Parametric Optimization Process to minimize the stress concentration at these regions of the anti-roll bar that is subjected to dynamic loading during the service life of the vehicle is very much essential.

3. PARAMETRIC OPTIMIZATION

3.1. Introduction to Optimization

In this paper, a methodology is presented to investigate the possibility of obtaining reduction in stresses at corner bends as well as weight reductions in 51CrV4 Spring Steel Stabilizer Bar while fulfilling certain criteria such as deflections and stresses occurring in the bar. For the ANSYS DesignXplorer the procedure for optimization can be summarized in the following steps [5],

1. Read the ANSYS APDL file into DesignXplorer and record the input and output parameters.
2. Through DesignXplorer central composite DOE scheme create candidate designs (Automatic design points).
3. Create response surface using second order polynomial based regression analysis using the candidate designs and the true responses.
4. Define design goals for the optimization such as allowable constraints etc.
5. Create new design points through sample generation from the specified goals
6. Select the best candidate/candidates from tradeoff study and verify the validity of the candidate design points by running analyses on candidate designs in simulation and thereby creating reference design points.

In order to optimize the anti-roll bar shape and obtain the minimum value of stress concentration at the critical regions determined via the primary stress analysis, DesignXplorer™ module of ANSYS® Workbench™ V13.0 commercial finite element software is used. To build the solid model, Catia V5 R20 commercial software is also used. To carry out the optimization process, firstly input and output parameters are determined. Then DesignXplorer™ module is started and Design of Experiments (DOE) method is chosen. Nine automatic design points are generated for two input parameters; R and L_2 by the software package in pursuance of the workflow of DesignXplorer™ that is given in Fig.4. Stress analysis that corresponds to these points is carried out. By using the results of this analysis, 3-D response surfaces for maximum von Mises stress and anti-roll bar mass are also generated by the FEA software. In the next step, in light of the design targets and limitations, "Goal Driven Optimization (GDO)" approach is utilized to predict the best combination of the chosen parameters by using the results obtained in the earlier step. Targeted design point that corresponds to optimal geometry of the anti-roll bar is estimated.

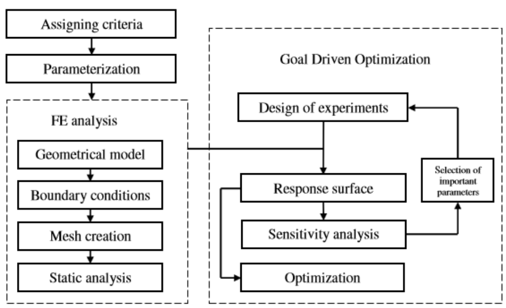

Fig.4 Flow chart showing the steps and their interactions

The design criteria for deflections and stresses were based on global strength of the stabilizer bar with equal and opposite displacement loads acting on stabilizer end legs. Also for this investigation stresses at the end connections are not taken into account, which are important as a design criterion from a local strength point of view. Friction is neglected between the antiroll bar and bushes while carrying out the analysis. The investigation focuses on the analysis of a stabilizer bar corner bends in a particular location in a vehicle. A stabilizer bar in a different position in the vehicle might have different dimensions and will be subjected to different accelerations. This results in a completely different loading condition. Therefore, the conclusions of this paper might not necessarily be directly applicable to stabilizer bar which is different than the one presented here.

3.2 Determination of geometric parameters

In order to reduce the stress concentration, design enhancement of anti-roll bar corner bends has been proposed. This change has been found based on two geometric parameters. To what extent the modification in these parameters alter the stress concentration and mass is also examined. The geometric parameters determined at the critical regions are,

- Transition radius, R
- Transition length, L2

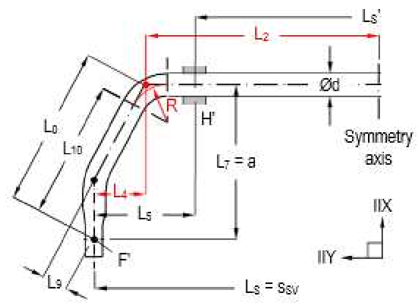

Fig.5 Basic dimensions of antiroll bar

After importing the parametric solid body from CAD model to ANSYS WORKBENCH, firstly change is made in all settings like named parameters and geometric coordinate system which use the Parameter and key to filter out the specific parameters of interest as shown in fig.5.

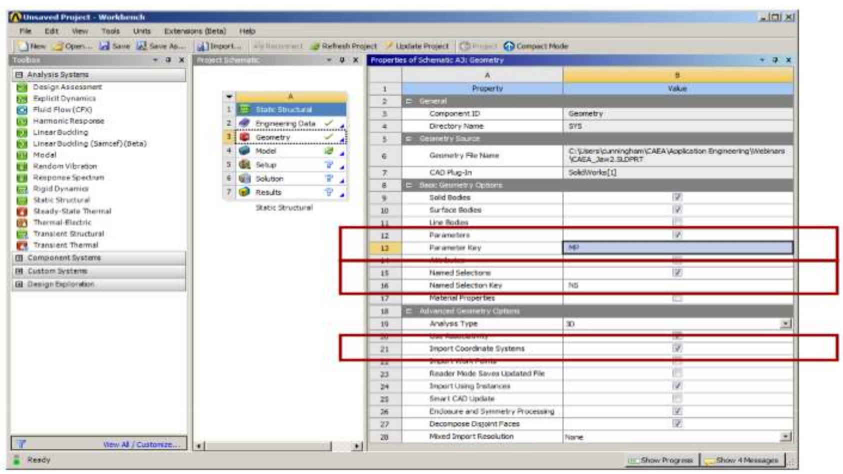

Fig.5 Geometry property filter out view in ANSYS WORKBENCH 13

3.3 Model Setup

Apply all loads and supports in the Mechanical interface. The Named Selections that were imported from the CAD model are used to define loads and supports. If any of the load magnitudes are varied parametrically add them to the Parameter Set by clicking on the box to the left of item capital P in checkbox. Output parameters consist of the result quantities that are to be kept track off. Output quantities consists of the following,

- Mass and Volume from the Details menu of a body (in Geometry)
- Minimum or Maximum quantities for standard result items in the Solution folder (deformation, strain, stress, reaction force, user defined Results, contact pressure etc.)

- Scalar parameters defined in a post-processing command block shown in fig.6

To perform the deterministic analysis of the stabilizer bar for the following parameter ranges,

- Transition Length : 360 mm to 440 mm
- Transition Radius : 90 mm to 110 mm

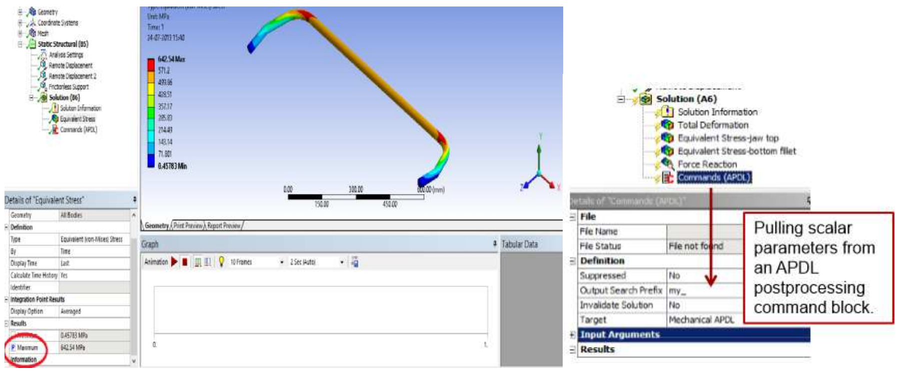

Fig.6 Output parameter selection with command block

Then DesignXplorer™ module is started and Design of Experiments (DOE) method was chosen. Nine automatic design points are generated for two input parameters; R and L2 by the software package in pursuance of the workflow of DesignXplorer™ that is given in Fig.4. Stress analysis that correspond to these points is carried out. By using the results of this analysis, 3-D response surfaces for maximum von Mises stress and anti-roll bar mass are also generated by the FEA software. As a next step, in the light of the design targets and limitations, “Goal Driven Optimization (GDO)” approach is utilized, to predict the best combination of the chosen parameters, by using these results. Targeted design point that corresponds to optimal geometry of the anti-roll bar is estimated.

4. RESULTS AND DISCUSSION

Following results are obtained from the analysis of the solid curved stabilizer bar, Mass =7.07 kg, Max. Principal Stress = 537.76 MPa, Max. Equivalent Stress = 647.61 Mpa. The variation of equivalent stress along bend bar portion is plotted which is as shown in Fig.7.

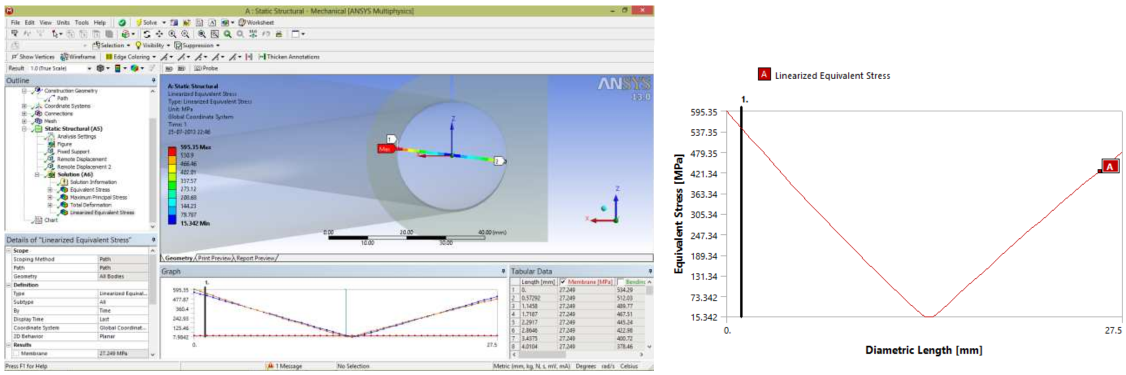

Fig.7 Stress variation at bend section of solid curved stabilizer Bar

In this analysis the peak stresses are observed in the bend portion of the bar and in the bushing locations. The most obvious effect of using hollow section is the reduction in mass of the stabilizer bar. However, maximum stresses on the bar have increased. Almost 50% mass reduction is obtained at the cost of 22% increase in stresses. Bushing location is another parameter of anti-roll bar design. By placing the bushings closer to the center of the bar, the stress at the bushing becomes higher than the stresses at the bend portion.

Table 3 Calculated maximum stress and mass values for specified design points of solid bar

Design Point	1	2	3	4	5	6	7	8	9
Transition Length (mm)	400	360	440	400	400	377.16	422.84	377.16	422.84
Transition Radius (mm)	100	100	100	90	110	94.29	94.29	105.71	105.71
Equivalent Stress Maximum (MPa)	642.54	716.56	580.42	644.58	643.88	685.31	607.48	685.53	600.92
Solid Mass (kg)	6.94	6.76	7.27	6.92	6.97	6.82	7.08	6.84	7.13

Maximum von Mises stress and anti-roll bar mass values calculated for the design points specified by parametric optimization software are given in Table 3, with that results generate the response surfaces for as functions of transition radius R and transition length L_2 parameters of solid and hollow type bar that are given in Fig.9 and Fig.11. The effect of the transition length L_2 on stress concentration and mass for selected values of the transition radius R are shown in Fig.8.

Table 4 Calculated maximum stress and mass values for specified design points of hollow bar

Design Point	1	2	3	4	5	6	7	8	9
Transition Length (mm)	400	400	400	360	440	377.16	377.16	422.84	422.84
Transition Radius (mm)	100	90	110	100	100	94.29	105.71	94.29	105.71
Equivalent Stress Maximum (MPa)	667.11	671.6	663.93	726.13	603.67	707.96	713.66	634.6	631.24
Solid Mass (kg)	3.4	3.39	3.41	3.31	3.56	3.34	3.35	3.47	3.49

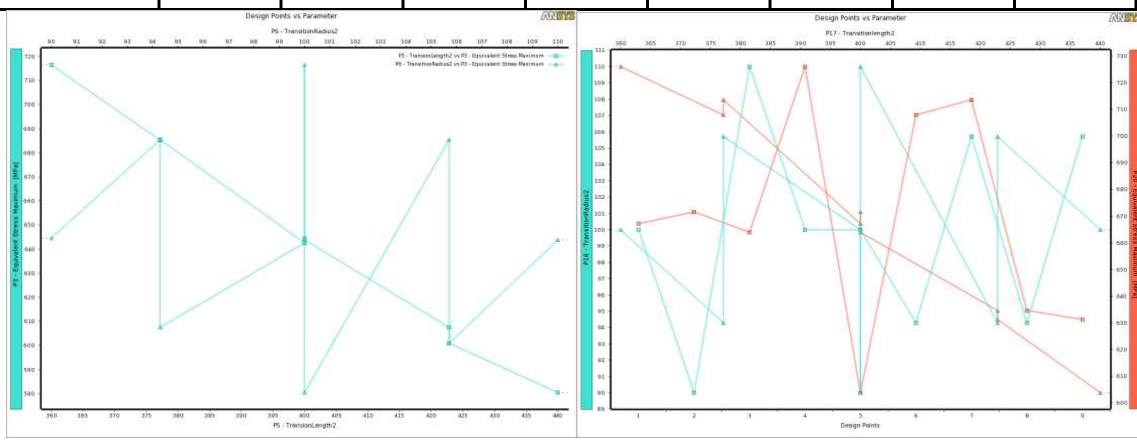

Fig.8 Plots of Design Points versus Input and Output Parameters for solid and hollow bar

Analyses showed that, reduction of stress concentration is possible by increasing the geometric parameter L_2 from 400 to 440 (mm). The calculated decrease is 12.2% for R= 90 (mm) and 8.18% for R= 110 (mm) which are the two limit values for the transition radius.

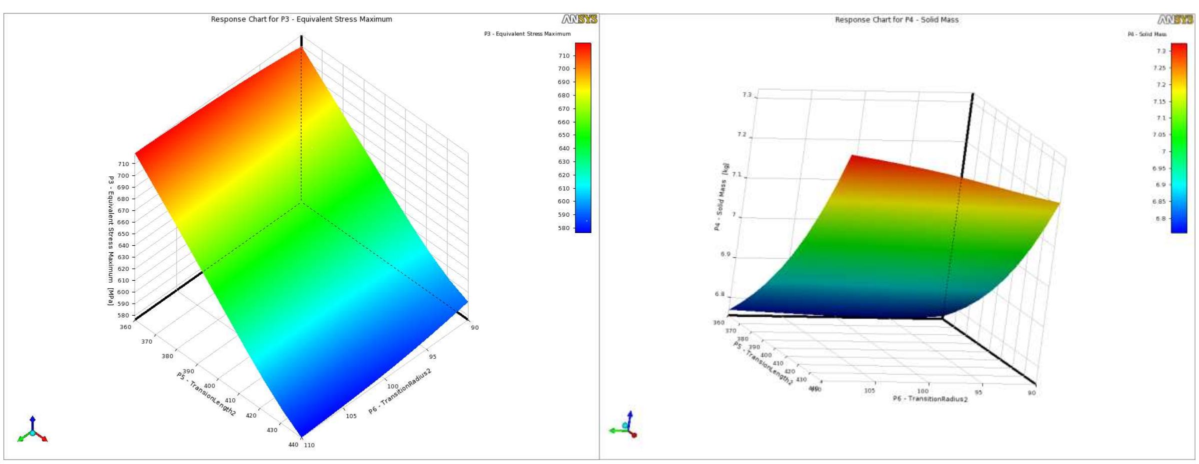

Fig.9 3D Response Surface for Equivalent Stress and Mass for solid bar

Table of Schematic D4: Optimization

	A	B	C	D	E
1		P5 - TransionLength2	P6 - TransitionRadius2	P3 - Equivalent Stress Maximum (MPa)	P4 - Solid Mass (kg)
2	Optimization Study				
3	Objective	No Objective	No Objective	Minimize	No Objective
4	Target Value				
5	Importance	Higher	Higher	Higher	Default
6	Candidate Points				
7	Candidate A	439.96	108.12	576.95	7.3136
8	Candidate B	439.24	99.678	581.62	7.2631
9	Candidate C	439.4	90.615	591.21	7.2081
10	Verification C			586.7	7.2088

Fig.10 Optimized Best Value for Minimized Equivalent Stress

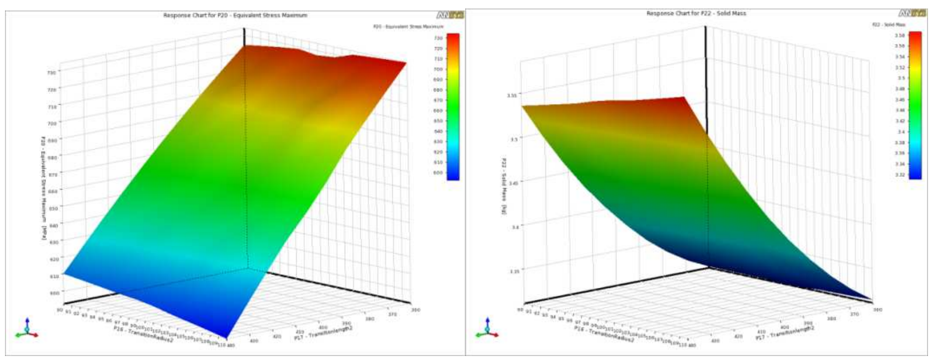

Fig.11 3D Response Surface for Equivalent Stress and Mass of Hollow Bar

It is also observed that higher L_2 value causes a mass increase of about 4.75% for R= 90 (mm) and 3.8% for R= 110 (mm). The effect of the transition radius R on stress concentration and mass is also shown in Fig.8. The analysis pointed out that increasing the transition radius acts as a stress riser, over an optimum R value. From the analysis it is found that a larger corner bend radius reduces anti-roll bar mass to a certain extent. The calculated reduction in mass is 2.95% for L_2= 440 (mm) and 3.4% for L_2= 360 (mm).

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Table of Schematic D4: Optimization

	A	B	C	D	E
1		P16 - TransitionRadius2	P17 - Transitionlength2	P20 - Equivalent Stress Maximum (MPa)	P22 - Solid Mass (kg)
2	Optimization Study				
3	Objective	No Objective	No Objective	Minimize	No Objective
4	Target Value				
5	Importance	Higher	Higher	Higher	Default
6	Candidate Points				
7	Candidate A	109.19	439.34	★★★ 594.7	3.5805
8	Candidate B	100.23	439.88	★★★ 603.66	3.5625
9	Candidate C	91.27	438.79	★★ 611.41	3.5334

Fig.12 Optimized Best Value for Minimized Equivalent Stress for hollow bar

Table 5 Comparison of Parametric Stabilizer bar Results

Bar type	Output parameter	Before optimized value	After optimized (DoE) value	FEA optimized value
Solid bar	Max. Equivalent stress (MPa)	642.54	576.95	582.05
	Mass (kg)	6.94	7.27	7.27
Hollow bar	Max. Equivalent stress (MPa)	667.11	594.7	602.97
	Mass (kg)	3.40	3.55	3.55

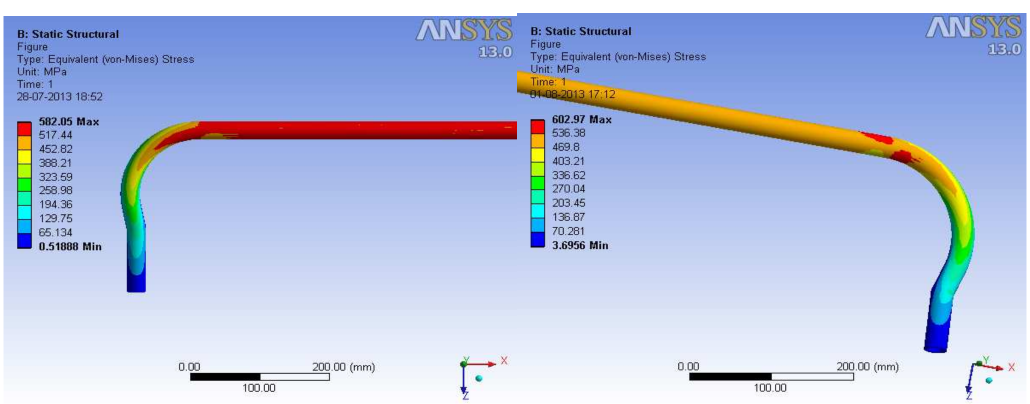

Fig. 13 FEA of Optimized parametric values of solid and hollow bar

The optimal values of the chosen parameters are estimated via Design of Experiments method as R= 108.12 mm and L_2= 439.96 mm for solid type stabilizer and R= 109.19 mm and L_2= 439.34 mm for hollow type stabilizer bar as shown in fig.10 and fig.12. To validate this result, a CAD model of the anti-roll bar that corresponds to these new values is prepared and finite element analysis is carried out for this model. The results of this analysis showed that the difference between the predicted and calculated values for maximum von misses stress in solid type bar is 0.86% and 1.22% for hollow type stabilizer bar, since increasing L2 decreases the maximum stress. The comparison of the DOE prediction and finite element analysis is given in Table 5.

The total decrease in the maximum stress σ_{max} at the critical regions for this design is calculated as 11% with a mass increase of 3.75% for solid type stabilizer bar. A decrease of 12% in the maximum stress with increase in mass 3.45% for hollow type stabilizer bar is noticed. The shape of the anti-roll bar corner bends that provides optimum stress concentration for determined design limits is shown in fig.13.

5. CONCLUSIONS

A stabilizer bar that will be used in the front axle suspension of the vehicles is redesigned to minimize the stress concentration at the corner bends for given structural limits. For this purpose, the effects of two design parameters; the transition length (L_2) and the transition radius (R) that constitute the geometry of the critical regions are studied for solid as well as hollow stabilizer bar. Locating the bushings closer to the center of the bar increases the stresses at the bushing locations. Also the weight of the hollow anti-roll bar is less than the solid bar but the stresses on the hollow bar are higher.

The parametric optimization is applied via ANSYS Workbench V13.0 commercial finite element software by using Design of Experiments (DoE) approach. FE analyses showed that it is possible to decrease the maximum equivalent stress at the critical regions for solid and hollow stabilizer bar to 11% and 12% with a mass increase of 3.75% and 3.45% respectively.

The results obtained can be summarized as: An increase of the transition length (L_2) decreases the equivalent Von misses stress at the corner bends, however raises the anti-roll bar mass. Also the effect of increasing the transition radius R raises the equivalent stress and also the notch effect at the critical regions over an optimum R value. Increasing the transition radius R also decreases the anti-roll bar mass.

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