To Study the Design Optimization of the Tie Rod with Mass Reduction using CAD/CAE Tools

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Abstract— In the present work, a three dimensional model of a Car tie rod is prepared in CAD modelling tool SOLIDWORKS and imported into FEM assisted CAE tool, ANSYS. A proper description of FEM is also presented to describe the working of this technique. First of all the Failure analysis of the tie rod is done to determine the sensitive region of the tie rod and failing forces. The material assigned to the model is AISI 8620 alloy steel. The meshing of the tie rod is done and then proper boundary conditions are applied and the equivalent stress and deformation is calculated. The failure analysis shows that the highest stresses are produced at the edge of the threaded joint of tie rod and the under the compressive forces of 4205 N, the stresses surpass the compressive yield strength of the material and tend to fail the tie rod. The changes in the sensitive part of the tie rod are done in order to reduce the stresses induced in the tie rod. The fillets are added to the edge of the threaded joint of the tie rod and for each case, the structural analysis using same boundary condition and forces is done to determine the values of equivalent stress and mass of the tie rod. The compressive forces kept are 4200 N since this was the highest value at which tie rod remains safe during failure analysis. The modification is done only in the sensitive part of the tie rod. After the structural analysis of each case of change, it is found that the maximum stress reduction is 30.014 and the mass reduction percentage is 0.27 when compared to the original model.

Key words: Universal joint, Finite Element Methods, Optimization

I. Introduction

The tie rod is part of the steering mechanism in a vehicle. A tie rod is a slender structural rod that is used as a tie and capable of carrying tensile loads only. A tie rod consists of an inner and an outer end. The spokes on a bicycle's wheels are tie rods. As the ratio of its length to the radius of gyration of its cross section is normally quite large, it would likely buckle under the action of compressive forces. The tie rod transmits force from the steering center link or the rack gear to the steering knuckle. This will cause the wheel to turn. The outer tie rod end connects with an adjusting sleeve, which allows the length of the tie rod to be adjustable. This adjustment is used to set a vehicle's alignment angle. The working strength of the tie rod is that of the product of the allowable working stress and the minimum cross-sectional area.

Fig. 1.1: A vehicle tie rod

Tie rods are connected at the ends in various ways. But it is desirable that the strength of the connection should be at least an equal strength to that of the rod. The tie rod ends can be threaded and then passed through drilled holes or shackles (this is a U-shaped piece of metal that is secured with a pin or bolt across the opening), and then retained by nuts than are screwed on the ends.

A. Forces acting on the Tie rod

While driving through uneven street surface, the suspension framework is subjected to a few conditions that outcome in wear and tear of different parts that makes up its edge work. Be that as it may, the tie rod which is one of such parts is inclined to experience the ill effects of the vertical, flat and sidelong strengths following up on the suspension framework when the street surface is loaded with potholes and other hindering conditions that will actuate such powers on the vehicle suspension framework.. The tie bar co-ordinate framework is as per the following;

1) X-bearing is radially outward horizontally
2) Y-bearing is radially outward vertically
3) Z-bearing is along the length of the rod

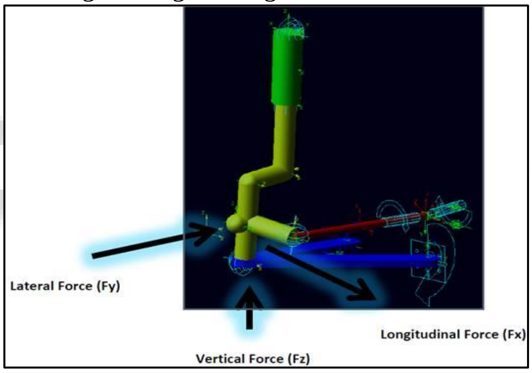

Fig. 1.3: Forces acting on the connecting rod

In the proposed work, the structural analysis of the tie rod is done by fixing the both ends of the tie rod and the applying the compressive forces on it from the both end joints of the tie rod (Figure 6.5).

B. Materials used in Tie rod

The actual model of the car taken into account was made of AISI 8620 alloy steel. AISI 8620 is an alloy carburizing steel when thermally treated builds up a hard external case with a soft ductile core. Precisely controlled extents of chromium, nickel and molybdenum are in charge of the broad use as a carburizing compound steel, with important elements including extraordinary surface hardenability and great inside quality. Has least twisting and development qualities. 8620 has great machinability and reacts well to polishing.

II. Objectives of the Present Work

The objectives of the proposed work are as following:

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1) Study of the effect of the forces acting on the tie rod and load distribution over the tie rod structure.
2) Model preparation in CAD tool SOLIDWORKS and converting into STEP format to import it into CAE tool named ANSYS to run the structural analysis under considered boundary and loading conditions to validate previous results[1].
3) Running the static analysis and determination of von-mises stress and deformation occurring due to applied forces and determination of sensitive parameters.
4) Modifications in the sensitive parameters to optimize the tie rod and finalize the optimum model possessing less weight with lower load impact over it.

III. CAD MODELLING

The CAD models of the tie rod are prepared in SOLIDWORKS. The AISI 8620 alloy steel was the material used for the universal joint. The mass of the Tie Rod is 0.66189 Kg.
The drawings of the universal joint cross and yoke are given below:

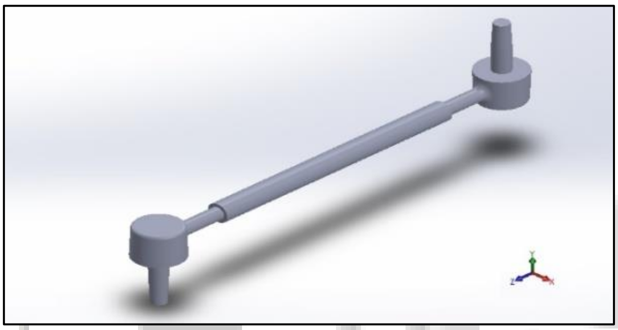


Fig. 3.1: The original 3D model of TIE ROD

IV. STATIC STRUCTURAL ANALYSIS

The structure examination of tie rod was performed in the ANSYS 14.5. Mechanical properties of structural steel and boundary conditions are given below;

PROPERTY	VALUE
DENSITY	7.85 g/cm3
YOUNG'S MODULUS	2E+11 Pa
POISSON'S RATIO	.30
ULTIMATE TENSILE STRENGTH	540 MPa
YIELD STRENGTH	390MPa

Table 4.1: Properties of structural steel

The model was then imported into the ANSYS WORKBENCH using Geometry interface. The model imported was in the STEP. format. Then through the Model interface, assignment of material, meshing and the structural analysis was done.3. The meshing of the tie rod model which is the basic operation of the Finite element modelling was done.The number of nodes and elements formed were 32051 and 19049 respectively which is quite good.

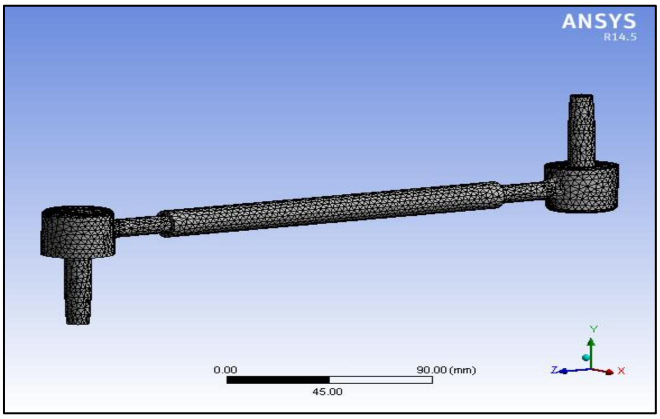


Fig. 4.1: meshing

A. Boundary conditions

The boundary conditions were applied to the tie rod model in order to run the structural analysis of the tie rod.

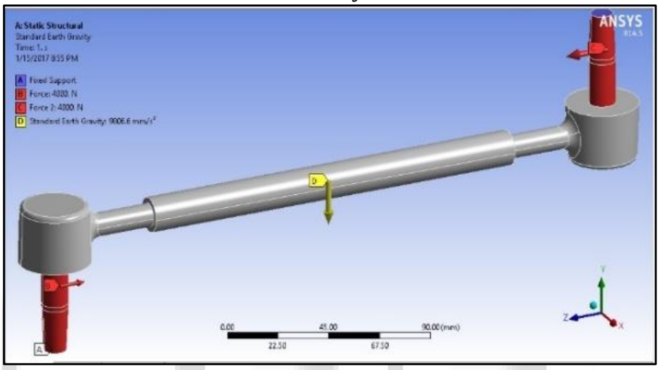
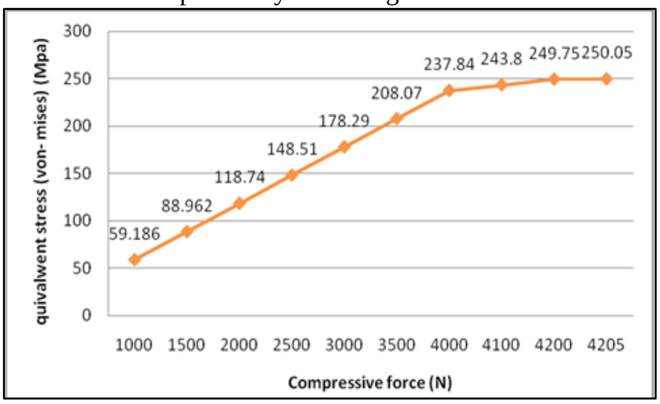


Fig. 4.2: boundary conditions and forces acting on the tie rod

The both end threaded parts were fixed and as it is known that the compressive forces including with the self-weight acts on the tie rod, the forces were applied

B. Failure analysis

The failure analysis of the Tie rod was done so that the maximum force at which the tie rod could fail be determined. The force which exceeds the compressive yield strength will be considered as the failing force. In this step, the identical forces but with different magnitude are applied on the tie rod keeping the same boundary conditions. The procedure was repeated until the stress value did not increase the compressive yield strength.



Graph 4.1: compressive force vs. Equivalent stress (von-mises)

The above graph shows that the structural failure of the tie rod occurs at the compressive force exceeding 4200 N. Thus the tie rod is safe under the 4200 N compressive force considering the body weight.

V. OPTIMIZATION OF TIE ROD

The structural analysis was run in order to find the magnitude of the forces at which the equivalent stress value exceeds the ultimate tensile strength value. Further the optimization of the tie rod model in order to reduce the mass and stress value is approached at the maximum value of force at which the tie rod is safe.

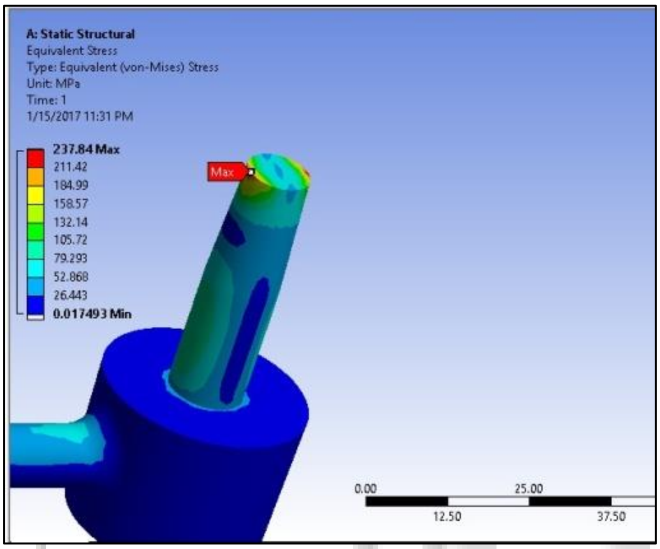


Fig. 5.1: Maximum equivalent stress (von-mises)

Since from the failure analysis, it is clear that the tie rod is safe up to the magnitude of 4200 N of compressive force beyond which it will fail as the equivalent stress will exceed compressive yield strength of 250 MPa in case of AISI 8620. Also from the structural analysis, we have found the most sensitive part of the tie rod (Figure 6.1)

A. *Addition of fillets at the sensitive edges*

The edges of the constrained parts were found to be under highest stress (figure 6.1). So fillets were tried at these edges of which the results along with figures are shown below

Fillet Radius	Equivalent stress (von-mises)	Mass
1 mm	325.47MPa	.66184 kg
2 mm	263.83MPa	.66158 kg
3 mm	256.76MPa	.66119 kg
4 mm	200.87 MPa	.66067 kg
5 mm	174.49MPa	.66005 kg

Table 5.1: Equivalent stress and mass of tie rod for respective fillets at sensitive part

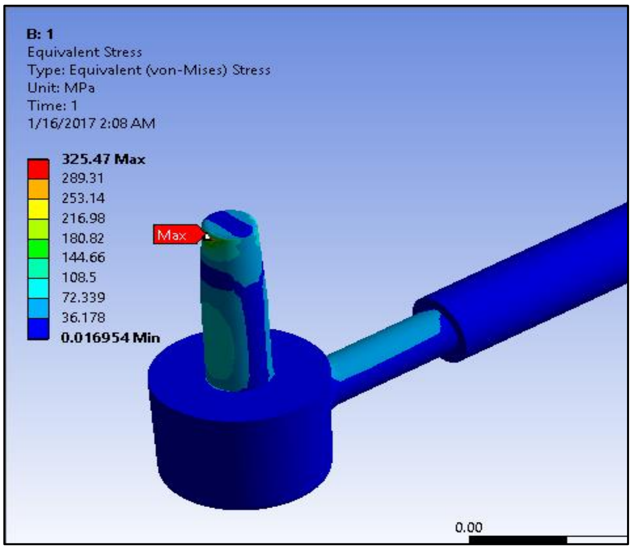


Fig. 5.2: Equivalent stress for filleting the edge by 1 mm radius

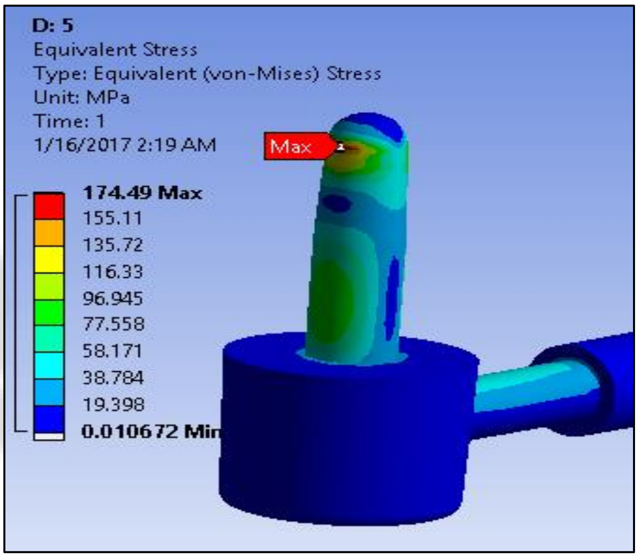


Fig. 5.3: Equivalent stress for filleting the edge by 5 mm radius

Calculated entities	Original model	Optimized model
Equivalent stress (MPa)	249.75	174.49
Mass (kg)	0.66189 kg	0.66005
Stress reduction percentage	-	30.014 %
Mass reduction percentage	-	0.27%

Table 5.2: comparison of stress and mass results of original and optimized tie rod

VI. CONCLUSION & FUTURE SCOPE

A. *Conclusion:*

The conclusion drawn from the present work is presented below:

– The Tie rod dimensions were actually taken from an original Tie rod of a car. The Failure analysis of the Tie rod was done in order to determine the maximum force it can undergo. Since the compressive yield strength of AISI 8620 steel alloy is 250 MPa, therefore from the

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failure analysis, it was found that the safest compressive force magnitude was 4200 N above which the tie rod would fail.

- The most sensitive parts of the tie rod where the maximum equivalent stress (von-mises) occurs are the open end edges of the tie rod joints. During the optimization process, the stress reduction and mass reduction was done from that sensitive part. During the structural analysis, the maximum compressive force was kept 4200 N in each case.
- The stress reduction and mass reduction percentages came out to be 30.014% and 0.27%. The sensitive are which was a flat edge is provided with a fillet of radius 5 mm.
- Further the higher compressive forces are applied to the tie rod with same boundary conditions, it is found that the optimized tie rod is safe up to the compressive forces of magnitude equal to 6000 N.

B. Future scope

- The mass reduction from the tie rod can further be done from the non-sensitive part of the tie rod bearing very much low stresses.
- The new materials can be assigned to the tie rod and their structural and failure analysis can be done in order to finalize a lighter and high stress bearing tie rod.

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