Performance Evolution of Tie Rod in Suspension System of Car using Finite Element Approach

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Abstract— Tie rods or Track rods are an integral part of vehicle's steering system. Tie rod ties vehicle's steering rack to the steering arm. Tie rod may get fail due to varying forces and bumping of vehicle during steering. The forces from the steering are also considered during the static condition of car. Vibration and buckling of Tie rod has been continuously a concern which may lead to structural failure if the resulting vibration and stresses are undesirable and excessive. Present paper is aimed to assess buckling strength and compare buckling performance of Tie rod for different proposed dimension of Tie rod with constant length and same material. Finite element models of the Tie rod also analyzed to obtain stiffness and stress distributions in it. The mode shape and natural frequency results for different proposed dimensions of tie rod obtained in the normal modal analysis and in buckling analysis, the buckling load factor obtained for that tie rod are compared and critical buckling load is calculated. Results getting from the Finite element analysis are validated by using the theoretical results.

Key words: Tie Rod, Buckling, Critical Buckling Load, FEA

I. INTRODUCTION

Form the several years a great deal of research work has been invested to find out the critical buckling loads of columns. Theoretical and experimental research has predicted that geometrical imperfections and modified boundary conditions greatly influence the critical buckling load magnitudes and scatter of columns. A Tie rod contains such geometrical imperfections and modified boundary conditions from a perfect cylindrical rod, since a Tie rod consists of outer and inner ends threaded into a middle rod body, with changing end conditions. So it is important to accurately find out the buckling loads of Tie rods, especially ones that are critical in compression used in automobile industries.

Depending upon the design criteria of minimizing compression margin safety coupled with the degree of difficulty to predict buckling characteristics and accurately calculating the critical buckling load is of high importance. This research work gives an analytical study and that gives a designer a systematic approach to accurately predict the buckling load of a Tie rod. To accomplish the level of accuracy, a linear FE analysis is carried out. The goal of this work is to establish an acceptable method of predicting the stiffness, natural frequency and buckling load of a passenger car Tie rod due to axial compression.

The main work of tie rod is to transmit the motion from steering arm to steering knuckle in order to turn the wheel in conventional suspension system. And in macpherson strut suspension system force is transmit from rack to steering knuckle to turn the wheel. Tie rod mostely subjected to compressive load. Failure of tie rod may cause instability of vehicle and cause the accident. so it's important to check the strength of tie rod.

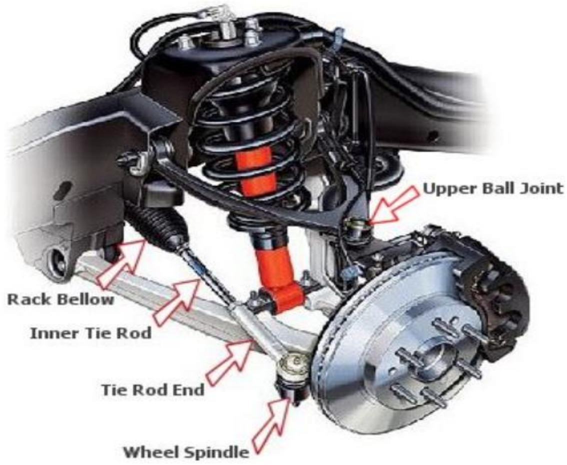

Fig. 1: Location of Tie rod

Pradeep chavan et al [1] FE analysis of Tie rod done to evaluate the performance of tie rod in suspension system. They assess buckling strength and compare the buckling performance of tie rod for different material. Tie rod also analyzed to obtain stiffness and stress distribution in each component. They concluded from his work that carbon steel material is good for manufacturing of tie rod for automobile vehicle as it shows better mechanical properties compared to cast iron and aluminum alloy material.

Raghvendra K et al [2] concluded from his work that, results decreases displacement, stress, mass count and increases buckling eigenvalue then buckling load will be increases compare to actual tie rod. Mild steel SAE 1020 ITR 2 proposal gives better results like stress, displacement, mass of the model and buckling load when compare to other iterations of tie rod.

Manik patil et al [3] concluded that, distribution of deformation and stress do not exceed the yield strength value and that there were neither damages nor failure of Tie rod. The correctness and accuracy of computed results was still dependent on the selection related to various modeling parameters. Some of the most important aspect such as boundary conditions or correct mesh and type of elements were performing a decisive role in achieving of correct results. According to deformation, stress and natural frequency result tie rod taking for analysis is safe.

George Campbell [4] tested on a large scale finite element model of a tie rod in NASTRAN. The static buckling load of a tie rod was analyzed. The results of the finite element model were compared with experimental results. The analysis was performed in three steps. First, linear buckling was analyzed with SOL 105. Second, a nonlinear static analysis with arc length method was performed in SOL 106 to determine the instability behavior of the structure. In the last step, a nonlinear buckling analysis was done with restart into SOL 106 to determine the nonlinear buckling load. The tie rod has a strongly nonlinear behavior which was due to material yield and geometric nonlinear effects.

II. PROBLEM DEFINITION

At first the theoretical study of Tie rod is done. The main work of Tie rod is to transmit the motion from steering arm to steering knuckle and sustain the forces and vibrations caused by bumps from tires due to uneven road surfaces. The main task in this paper is to find the deformation and stresses induced in the Tie rod for different dimensions. The 3-D model is prepared for Tie-rod. Different values of diameter for tie rod with same material and length are assigned and analysis is carried out using finite element analysis software. The results are compared to optimization of tie rod.

Fig. 2: Tie rod

III. OBJECTIVES OF THE RESEARCH WORK

The main Objectives of this research work are:

– Theoretical calculation for the critical buckling load of the car Tie rod for different dimension will be done.
– To generate 3D model of Tie rod initially accurate measurement of the existing Tie rod component will be done.
– Using that measured dimension of existing design and proposed design tie rod, 3D modeling of the car Tie rod will be done using modeling software.
– Determination of displacement and stresses generated will be carried out through Static Analysis.
– Buckling analysis done for finding the critical buckling load.
– Compare results for optimization of tie rod.

Parameter	Unit	ExistingDesign
π		3.14
E	N/mm^2	210000
I	mm^4	2598.32
L	mm	320
D	mm	16.0
d	mm	10.5
A	mm2	112.86
volume	mm3	36115.2
density	Tonns/mm^3	7.90E-09
mass	Tonns/mm^3	2.85E-04
mass	kg	0.290
Thickness	mm	2.75
Pcr	N	13275

Table 1: Parameter Details of existing tie rod

IV. FE ANALYSIS OF EXISTING TIE ROD

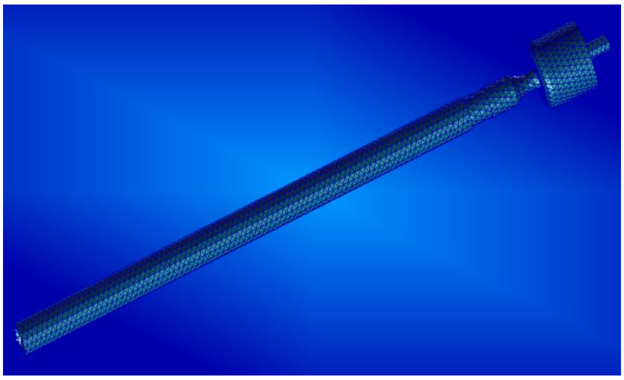

Fig. 3: FE Model of Existing Tie Rod

A. Extraction of Natural Frequency and Mode Shape of Existing Design Tie Rod

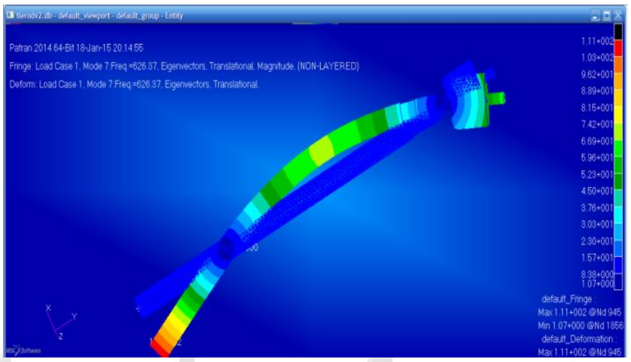

Fig. 4: Eigenvector plot - First mode shape – 626.37 Hz of Existing design

Above figure shows the plot of eigenvector for the first natural frequency of the existing design in free-free mode. The value of frequency is ~ 626 Hz. From the result it can be seen that the location of maximum displacement is at free end and second is at the center of rod. The mode shape is bending at center.

Natural Frequency	Frequency (Hz)
1	626.95
2	627.96
3	1636.36
4	1654.7

Table 2: Results of Normal mode analysis of Existing Tie rod

B. Stress Plot for Existing Design:

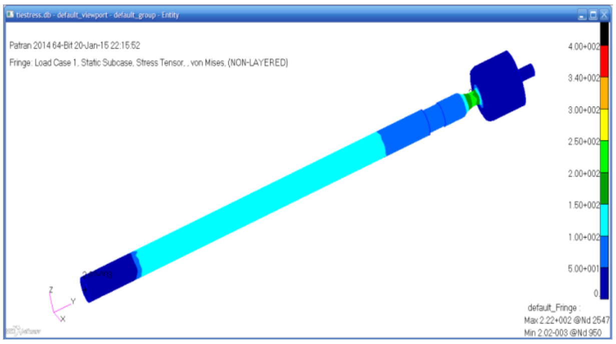

Fig. 5: Von-Misses stress plot -Existing design

Above figure shows the Von-Misses stress plot for existing design. The magnitude of Maximum stress is 222MPa and location of maximum stress is at the necking area below the ball joint.

C. FE Analysis of Proposed Tie Rod:

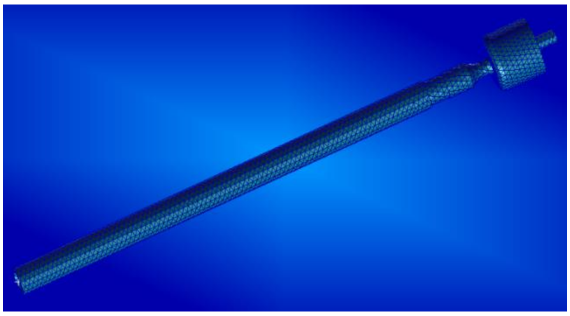

Fig. 6: FE model of proposed tie rod

Parameter	Unit	Optimized Design
π		3.14
E	N/mm2	210000
I	mm4	2917.70
L	mm	320
D	mm	18.5
d	mm	15.5
A	mm^2	80.14
volume	mm^3	25645.71
density	$Tonns/mm^3$	7.90E-09
mass	$Tonns/mm^3$	2.03E-04
mass	kg	0.202
Thickness	mm	1.50
Pcr	N	14748

Table 3: Structural Details of Proposed Tie Rod

D. Extraction of Natural Frequency and Mode Shape of Proposed Tie Rod:

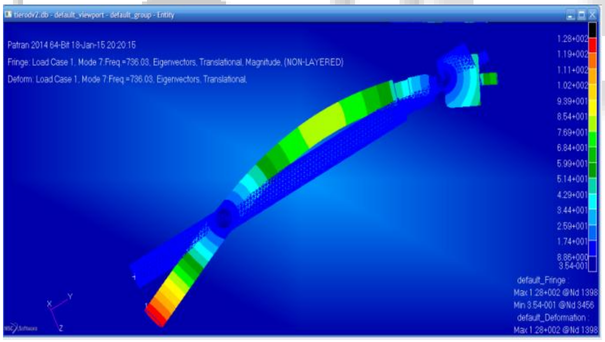

Fig. 7: Eigenvector plot - First mode shape – 736.03 Hz of Proposed design

Above figure shows the plot of eigenvector for the first natural frequency of the proposed design in free-free mode. The value of frequency is ~ 736 Hz. From the result it can be seen that the location of maximum displacement is at free end and second is at the center of rod. The mode shape is bending at center; which is in line with the existing design.

Natural Frequency	Frequency (Hz)
1	736.03
2	737.20
3	1819.41
4	1820.05

Table 4: Results of Normal mode analysis of proposed Tie rod

E. Stress Plot for Proposed Design:

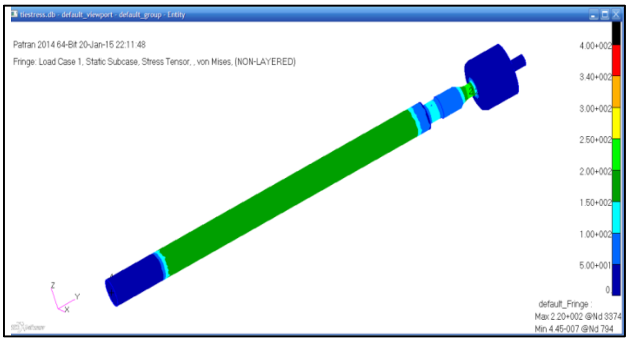

Fig. 8: Von-Misses stress plot -Proposed design

Above figure shows the Von-Misses stress plot for proposed design. The magnitude of maximum stress is 220 MPa and location of maximum stress is at the necking area below the ball joint.

V. CALCULATING CRITICAL BUCKLING LOAD WITH SOL 105 OF EXISTING TIE ROD

From f06 file we receive the first eigenvalue as shown below.

0 SUBCASE 2

REAL EIGENVALUES

MODE NO.	EXTRACTION ORDER	EIGENVALUE	RADIANS	CYCLES	GENERALIZED MASS	GENERALIZED STIFFNESS
1	1	1.608350E+02	1.268207E+01	2.018415E+00	3.986552E-01	6.411771E+01
2	2	1.611996E+02	1.269644E+01	2.020701E+00	3.992774E-01	6.436336E+01
3	3	1.286315E+03	3.586523E+01	5.708129E+00	1.181076E+00	1.519236E+03

Here the Eigen value = 1.608350E+02
Applied load = 100 N
Critical buckling load (Pcr)
= Eigen value * Applied load
= 1.60835 * 100
= 16083.5 N

A. Calculating Critical Buckling Load with SOL 106 of Existing Tie Rod:

Find CRITICAL BUCKLING FACTOR (ALPHA) = …..Value (obtain from f06 file).
In given example Subcase 102 gives ALPHA = 0.7037

REAL EIGENVALUES

MODE NO.	EXTRACTION ORDER	EIGENVALUE	RADIANS	CYCLES	GENERALIZED MASS	GENERALIZED STIFFNESS
1	1	6.606301E-01	8.127915E-01	1.293598E-01	3.911617E+00	2.584132E+00
2	2	1.344575E+00	1.159558E+00	1.845494E-01	7.892585E-01	1.061218E+00
3	3	-1.533607E+00	1.238389E+00	1.970957E-01	-3.948311E+00	6.055158E+00
4	4	-3.012476E+00	1.735649E+00	2.762371E-01	-7.961774E-01	2.398466E+00
5	5	-1.521872E+01	3.901117E+00	6.208821E-01	-5.841976E+02	8.890738E+03
6	6	1.532112E+01	3.914220E+00	6.229675E-01	5.306149E+02	8.129614E+03

*** USER INFORMATION MESSAGE 9040 (SUBDMAP NLSTATIC)
CRITICAL BUCKLING FACTOR (ALPHA)= 7.037617E-01
*** USER INFORMATION MESSAGE 4114 (OUTPX2)
DATA BLOCK OUG1 WRITTEN ON FORTRAN UNIT 12 IN BINARY (LTLEND) FORMAT USING NDDL DESCRIPTION FOR OUG, TRL =
101 0 183120 0 0 0

This is how we would calculate buckling Load.
General Formula: Pn + Alpha * (delta P)…..eq. (2)
Where,
Pn = Total Load at Subcase Id where Param, buckle, 2 is applied
Delta P = is the (Delta Load or (P (n) - P (n-1)) / No_of_increment.
Let us consider Subcase 102 (Alpha = 0.7037)
Total Load Pn = 20.00 (subcase 102)
Delta Load = (16083.0–15278.0)
= (Load @ (Subcase 2 - Subcase 1)
= 805.0
Pcr = Pn + (Delat_Load)/No_of_incr) * Alpha

= 16083.0 + (805.0/2) * 0.7037
= 16366.24 N

Description	Critical Buckling Load (N)	
	Existing Tie Rod	Optimized Tie Rod
Linear Buckling analysis SOL - 105 Result	16083.01	17707.04
Non Linear Buckling Analysis SOL - 106 Result	16366.24	17791.21

Table 5: Buckling analysis result for critical Buckling load of Tie rod.

VI. Results and Discussion

Sr. No.	Natural Frequency (Hz)	
	Existing Tie rod	Optimized Tie rod
1	626.95	736.03
2	627.96	737.20
3	1636.36	1819.41
4	1654.7	1820.05

Table 6: Results of Normal mode analysis of Tie rod

From the mode analysis of tie rod 1st mode natural frequency of proposed tie rod is 736 Hz which is 15% more than natural frequency of Existing tie rod 627 Hz.

Sr. No.	**Results**	**Critical buckling load (N)**	
		Existing	**Optimized**
1	Theoretical	13275	14748
2	Linear Buckling analysis SOL 105	16083	17707
3	Non Linear Buckling analysis SOL 106	16366	17791

Table 7: Compare Theoretical and buckling analysis result for critical Buckling load of Tie rod.

From linear buckling analysis the critical buckling load value obtain is around 16kN for existing design and around 17.7kN in proposed design. Results of non-linear buckling analysis show that, there is a some rise in critical buckling load compared to linear buckling analysis. As we know that, the non-linear analysis is more accurate than linear analysis. Hence SOL 106 results can be considered as the ultimate. Theoretical calculated critical buckling load for existing tie rod is 13.16kN and for proposed tie rod value obtained 14.7kN. The rise in the critical buckling load is attributed to the rise in inertia value of proposed design

VII. Conclusion

The finite analysis result shows that in mode analysis natural frequency of proposed tie rod 15% more than natural frequency of existing tie rod. Hence the life of proposed tie rod increases than existing for fatigue due to vibration. And from the theoretical calculation of critical buckling load critical buckling load of proposed tie rod is more than critical buckling load of existing tie rod. In case buckling analysis From linear buckling analysis the critical buckling load value obtain is around 16kN for existing design and around 17.7kN in proposed design. Results of non-linear buckling analysis show that, there is a some rise in critical buckling load compared to linear buckling analysis. As we know that, the non-linear analysis is more accurate than linear analysis. The critical buckling load of proposed tie rod is more than existing tie rod in buckling analysis and also the weight of proposed tie rod reduced than existing tie rod which is the today's automobile requirement. So the proposed tie rod is suitable for suspension system of passenger car.

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