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Optimization & FEA of Inner Tie Rod for Light Motor Vehicles (LMV)

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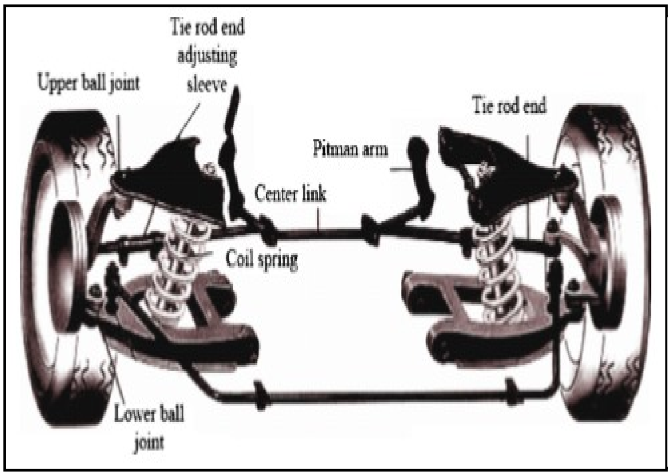
Abstract

The ratio of Tie Rod length to the radius of gyration of its cross section is normally quite large, it would likely buckle under the action of compressive forces. If it becomes worn out, steering will be producing clunking noise and also the vehicle will typically be pulling or (dragging) to either side (left or right) which will cause the accident. Which is not safe for passenger life seating in the car. The aim of the project is to analyze tie rod for the active improvementin the mass and buckling load of tie rod. This paper has the intention to evaluate buckling strength and compare buckling performance of Tie rod for different dimensions, theoretically calculate the critical buckling load of Tie rod with different diameter of it and keeping the same material and length. We have been compared& validated the theoretical results with the experimental test results obtained by the natural frequencies on FFT analyzer.

Keywords: Tie Rod, Load Carrying Capacity, FFT analyzer

I. Introduction

Design of suspension components in an automotive is very critical as they are constantly under varying loads. While designing the component we must ensure the safety. Apart from design prospective it is important to focus on the weight and cost of an individual component. The tie rod is an important part of suspension system. It connects the steering to the suspension in order to transform the motion. In Conventional suspension system tie rods connect the centre link to the steering knuckle as shown in fig.1.1. In MacPherson strut suspension and rack and pinion steering gears, tie rods connect the end of the rack to the steering knuckle as shown in fig.1.2. A tie rod consists of an inner and an outer end as shown in both figures. Tie rods transmit force from the steering centre link or the rack gear to the steering knuckle, causing the wheels 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 toes, a critical alignment angle, sometimes referred to as the caster and camber angles. A vehicle's steering and suspension systems should be checked regularly, at least once a year along with a complete wheel alignment.



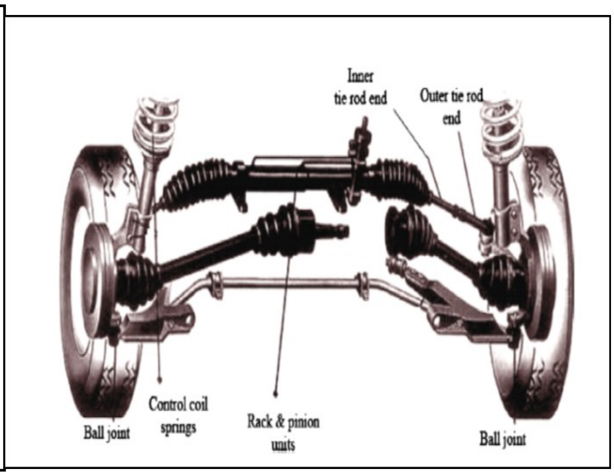


Fig.1.Conventional Suspension [1] Fig.2. McPherson Suspensions with Rack and Pinion [1]

The job of the tie rod end is to ensure the wheels are aligned. It provides the adjustment for wheel alignment that keeps the tires from wearing out on the inner and outer edges. If they wear out, the wheels will lose alignment and you may find that the tires and steering wheels are shaking when you drive the car. To evaluate the structural performance of tie rod, we need to consider the loads coming on tie rod. From various theoretical studies and practical observations, it is found that tie rod is primarily encounter under compressive loads and hence fails in buckling. Apart from this because of suspension components fluctuating loads are also coming on tie rod due to random loads coming on suspension of vehicle.Design &

Optimization of existing Inner Tie Rod for sustaining 22,750 N load without failure under 3G burm condition, with reduction in weight of around 15%. Suggest an alternative material for optimized design.An objective of this proposed work is to do the performance evaluation of an existing tie rod. This is done in terms of displacement, stress, and strain analysis. Enlist the parameters affecting the performance of tie rod under different loading condition. Evaluate the sensitivity of structure against different input parameters. Determine the optimization scope for weight reduction. Construct the design of new optimized model. Take the necessary physical tests like Compression test on Universal Testing Machine, Modal analysis test by using FFT Analyseron the tie rod in order to validate it against the actual physical conditions

Design Calculations

Theoretical Calculations:-

Parameter	Initial Design
Material	Steel SM45C
E	$210x10^3$ MPa
L	330 mm
D	15.61 mm
Density	$7700Kg/mm^3$
Tensile Yield Strength	360 MPa
Tensile Ultimate Strength	569 MPa

Table 1. Tie Rod Material Properties

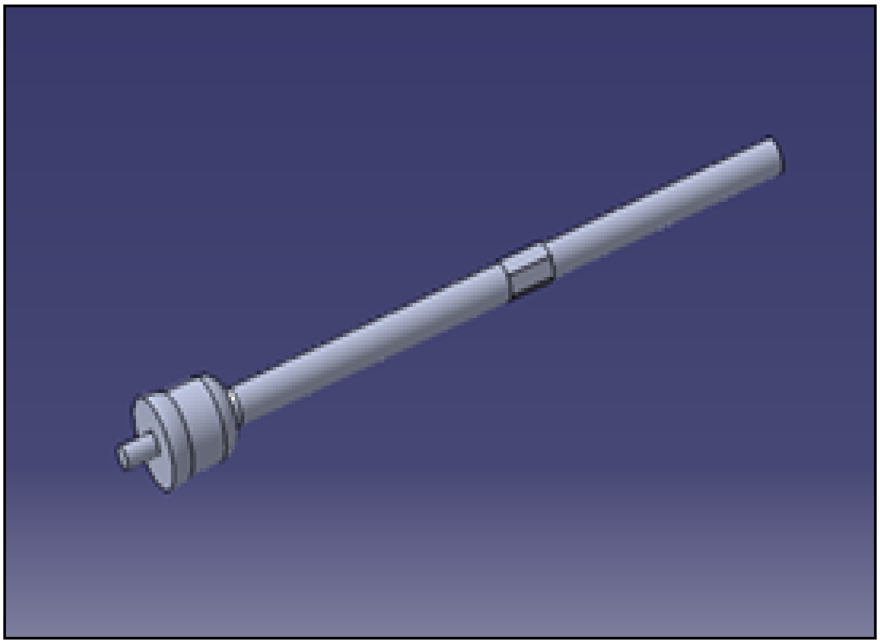


Fig. 3. Tie Rod CATIA Model

Steering movement ratio:

Where, R = 170 mm is the radius of the steering wheel.
And the output rack movement is:
Xo = 2 π r ---------------------------- (iii)
40 = 2 π r
Then, the movement ratio (MR) can be calculated as input movement over output:
MR = Xi/Xo--------------------------- (iv)
= 2 π R/2 π r
= 1068.11/40= 26.70therefore the movement ratio is 26.70: 1.
For an effort of 40 N applied by both hands on the steering wheel and considering no friction, the output load will be:
Fo = Fi * MR
= 40*26.70= 1068.11 N
Therefore the load transmitted to the tie rods is 1068.11

Cross Sectional Area (A):-

A=πr^2 = π*7.8052 = 191.38 mm2---------------------- (v)

Moment Of Inertia (I):-

I= π/64*d4 = π/64*154 = 2914.617 mm4 ---------------------- (vi)

Least Radius Of Gyration (K):-

K=√(I/A) = √(2914.617/191.38) = 3.902 mm---------------------- (vii)

Slenderness Ratio:-

L/K = 0.7l/K = 0.71*330/3.902 = 60.87---------------------- (viii)

So we use Rankine's formula, because Slenderness ratio is in between 60 to 120 for steel.

Critical Load (Pcr) (One end fixed and other end is hinged)

$$P_{cr} = \frac{a * \sigma_c}{1 + c\left(\frac{L}{r}\right)2} = 27498.29 \text{ N}$$

Stiffness (K):-

K= Load/Deflection = 27498.29/0.2258---------------------- (ix)

= 121.785 KN

Deflection (δ):-

δ= P*L/A*E =27498.29 * 330 / 191.38 * 210x103 ---------------------- (x)

= 0.2213 mm

FEA Analysis of Tie Rod

Load case scenarios	Fx(N)	Fy(N)	Fz(N)	Load
1G Static	0	0	2912	Compressive
7G Bump	15529	0	23293	Tension
1.10G brake	4454	0	4049	Tension
Brake and bump	16287	0	18430	Tension
1.30G cornering	0	-6564	5049	Compressive
Cornering and bump	16954	-6564	22430	Tension
3G bump	0	-22711	7750	Compressive
1.20G acceleration	-2005	0	1671	Compressive
Acceleration and bump	12696	0	22052	Compressive
1.00G Reverse Brake	-1878	0	1878	Compressive
Reverse Brake and bump	-16717	0	22259	Compressive
4G Ditch Hook	0	20036	774	Tension

Table 2. Different load case scenarios effects on the tie rods.

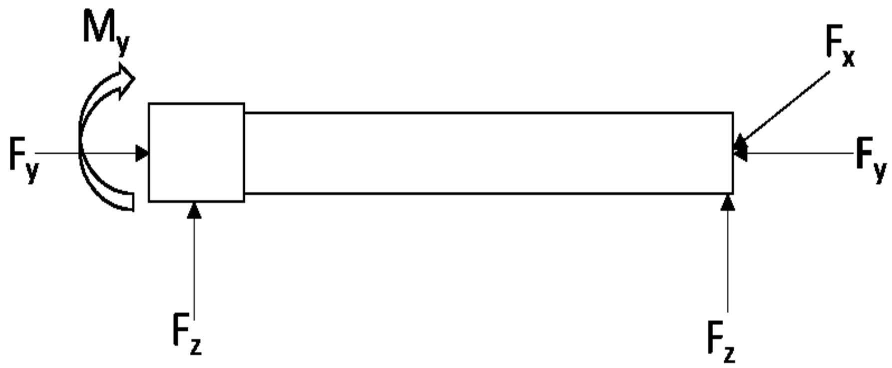
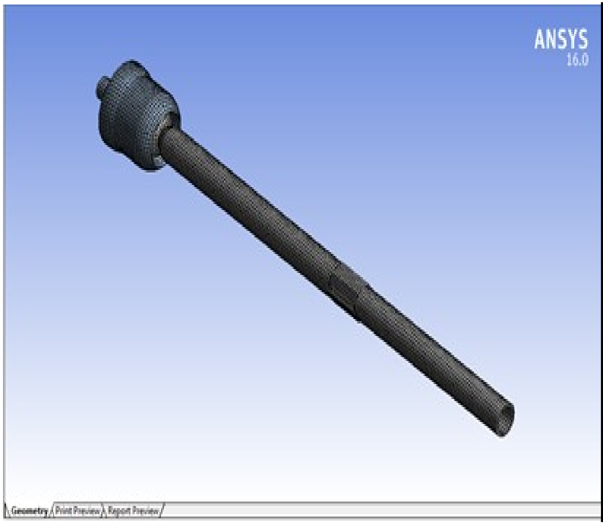


Fig. 4. Free Body Diagram of Inner Tie Rod

Based on the observations on load case scenarios table it is clearly observed that most of the loads are compressive in nature. In 3G berm condition high compressive force i.e. 22750 N is coming on Tie Rod. So, we choose 3G berm Condition for design & analysis purpose.



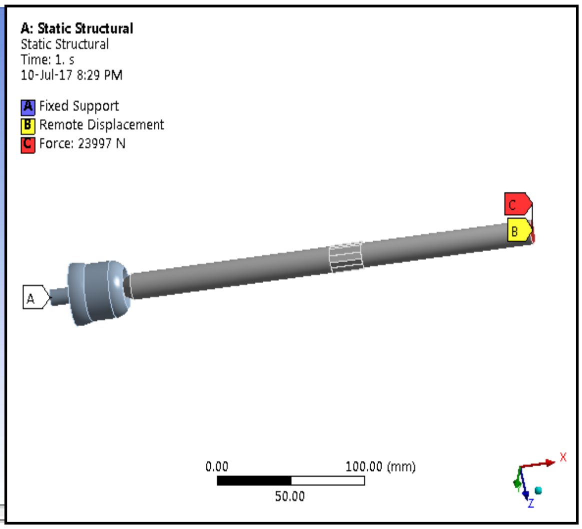
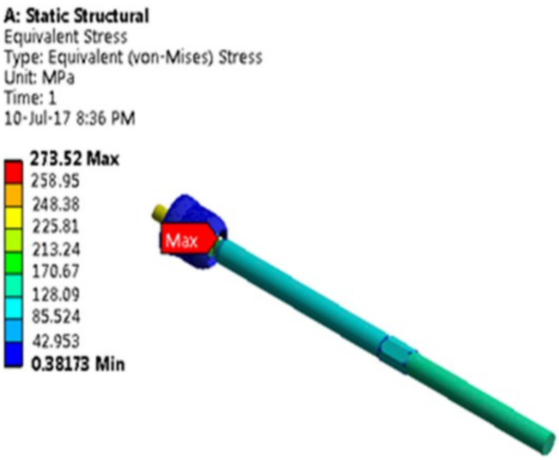


Fig.5. Meshed Model Fig.6.Boundary Conditions

Existing Tie Rod Analysis – 3G Berm Condition



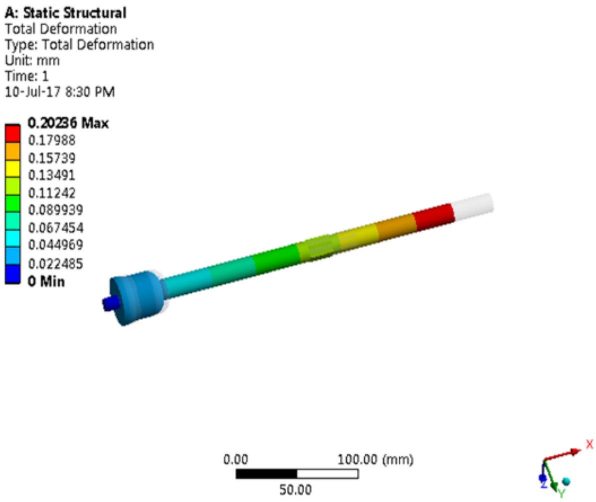
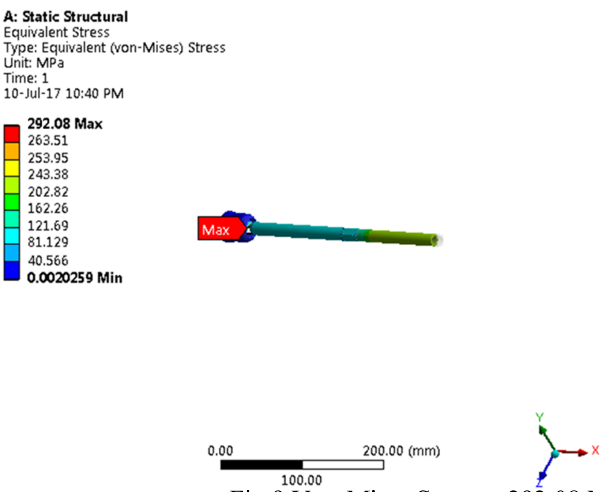


Fig.7.Von-Mises Stress - 273.52 MPa Fig.8. Total Deformation – 0.20236 mm

from this result we can say that our design is safe, because stress generated i.e. 273.52 Mpa < 299.25 Mpa. (Allowable Stress)

Optimized Tie Rod Analysis – 3G Berm Condition



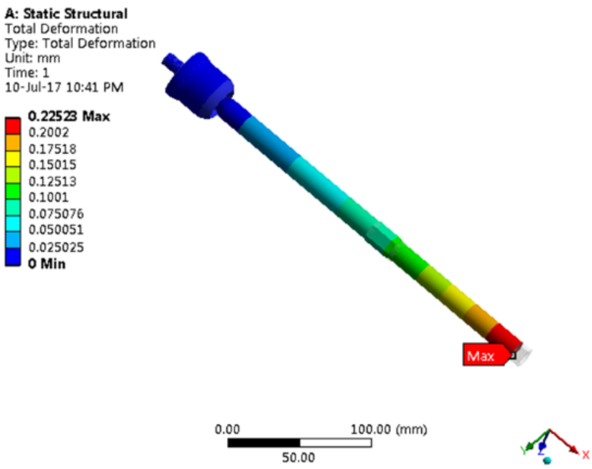


Fig.9.Von-Mises Stress – 292.08 MPa Fig.10.Total Deformation – 0.22523 mm

From this result, we can say that our design is safe, because stress generated i.e. 292.08 Mpa < 299.25 Mpa.(Allowable Stress)

Testing & Validation

Modal analysis was performed on optimized model of tie rod to determine critical natural frequency of tie rod and to avoid resonance.

Fig.11.Modal Analysis Test on FFT Analyser

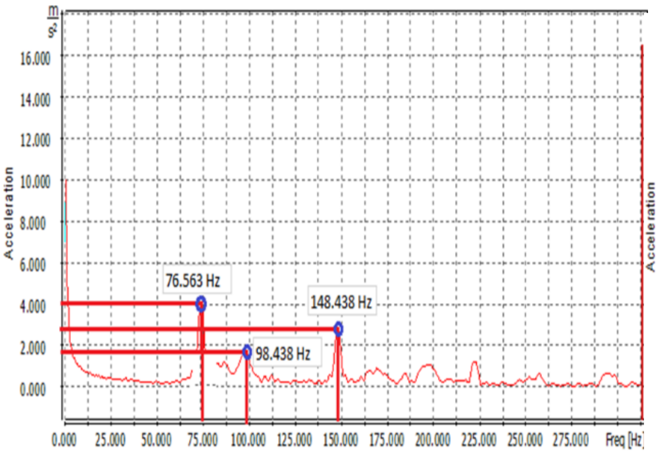


Fig.12. Existing Tie Rod

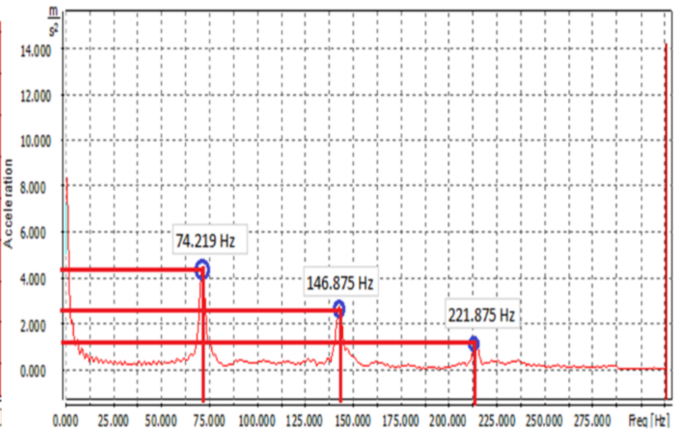


Fig.13.Optimized Tie Rod

The Modal analysis of Existing Tie Rod obtained by analysis is in the range of85 Hz. To 2167.7 Hz and the Modal analysis obtained by experimentally is in the range of 75 Hz. To 150 Hz. The Modal analysis of Optimized Tie Rod obtained by analysis is in the range of 100 Hz. To 3340 Hz and the Modal analysis obtained by experimentally is in the range of 75 Hz. To 230 Hz.

Parameter	Natural Frequencies
Existing Tie Rod	87.73
Optimized Tie Rod	100.87

Table 3. Natural Frequencies Comparison

Condition	**ID, mm**	**Material**	**Parameters**			
New Design	12.0	Steel- SM45C (Hollow)	Stress	Deformation	Weight	Original Weight
			292.08	0.22523	0.580 Kg.	0.689 Kg

Table 4. Obtained results for Optimized Tie Rod

Conclusion

Tie rod plays important role in steering system and should be carefully selected.The results we got for selected Optimized Tie Rod are showing good improvement compare to Solid tie rod in terms of weight, High Strength.Optimized Tie rod with ID 12.0mm is selected for optimization purpose.Overall (Compare to existing model with solid steel Tie rod) change in weight is 17.19 % for Steel- optimized tie rod.The finite analysis result shows that in modal analysis natural frequency of proposed tie rod is 12.7% more than natural frequency of existing tie rod. Hence the life of proposed tie rod increases than existing tie Rod. We can use Al6082 (Aluminium Alloy)as an alternative for Steel SM45C.Only 6% deviation in the buckling Deformation results, which is acceptable as per industry norms. Only 7 – 8% deviation in theoretical, Practical &

Validation part. So, from the obtained results we can conclude that the optimized tie rod is capable of carrying 22750 N compressive load in 3G Berm condition.Similar analysis approach can be followed for any type of Tie rod, in general for heavy and Construction vehicles.If the complete assembly is considered we can perform dynamic analysis using Flexible Body Dynamics.Design & develop the Quarter Car Test rig model as per the requirement & specification to test it on actual conditions. Fatigue life estimation of each component can be performed for car Tie Rod.

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