

Performance Optimization of Tie rod using FEA

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Abstract:- Structural performance of any mechanical component is measured basically in terms of its natural frequency, deformation, stiffness, maximum stress level, fatigue life etc. In case of vehicle suspension system; however tie rod is mainly under compressive and fluctuating forces encounter from steering and bumping of vehicle. When steering acts to turn the vehicle, tie rod comes under compressive load. And when vehicle running on rough road condition, fluctuating forces.

When we design the tie rod we need to know the design space available. The prime objective is minimize weight and cost. In order to achieve these targets we have optimize the parameters that affects the structural performance of Tie rod. This paper emphasizes on optimistic way of designing the tie rod that ultimately yields the effective and efficient performance of Tie rod.

Keywords:- FEA, Modal FEA, Natural frequency, stiffness.

I. INTRODUCTION

A tie rod is an important member in vehicle suspension system. It performs an important task of transferring the motion from steering system to suspension system. In a car's steering wheel is connected to the steering gear steering wheel turn the wheels. The steering gear is connected to the wheels via the tie rod ends. 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 studies and practical observations, it is found that tie rod is primarily encounter under compressive loads and hence fails in buckling. Moreover due to suspension components fluctuating loads are also coming on tie rod due to random loads coming on suspension of vehicle.



Figure1: Location of tie rod in suspension system

A. Role of tie rod in suspension system

Failure of tie rod may occur in terms of plastic (permanent) deformation or a fatigue failure. And it may cause the running vehicle in an unstable condition which may cause an accident. So it's important to validate the tie rod in every aspect of structural loading. The load coming on tie rod is mostly compressive. The efforts required where car is moving are comparatively less with stationary car. The working strength of the tie rod is that of the product of the allowable working stress and the minimum cross-sectional area.

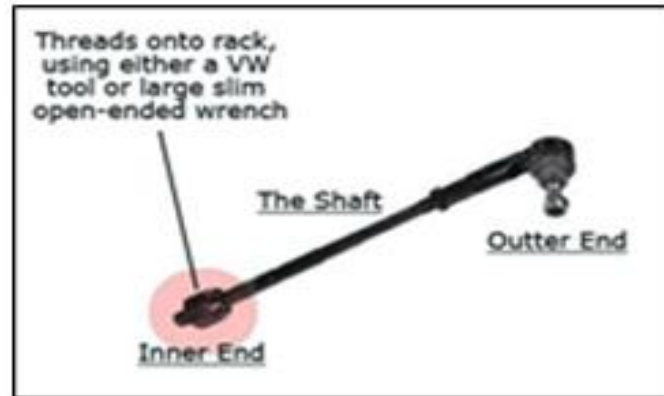


Figure2: Location of tie rod in suspension system

Rods are often made thicker at the ends and to keep its strength high at tie rod ends. It is advisable that steering and suspension systems are checked regularly, at least once a year along with a complete wheel alignment. A worn tie rod can cause loss of grip of vehicle, drifting, erratic steering and also major tire wear. If a tie rod is replaced, then a wheel alignment will also be required. Because tie rod replacement will disturb the alignment setting. 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.

II. PROBLEM DEFINITION

Initially the theoretical study of Tie rod is done. The main task in this study is to find the critical buckling load for the existing design. Observe the deformation and stresses induced in the Tie rod. Set up the benchmark for the proposed design. Try to evaluate the possibility with different material combinations for optimized design. Target is to minimize the weight keeping the critical buckling load constant. The stress targets are also need to be achieved.

A. Aim and Objective

An objective of this project 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 compressive loading. Evaluate the sensitivity of structure against different input parameters. Decide the optimization scope for weight reduction. Construct the design of new optimized model.

B. Linear buckling analysis:

In linear static analysis, a structure is normally considered to be in a state of stable equilibrium. As the applied load is removed, the structure is assumed to return to its original position. However, under certain combinations of loadings, the structure may become unstable. When this loading is reached, the structure continues to deflect without an increase in the magnitude of the loading. In this case, the structure has actually buckled or has become unstable; hence, the term "instability" is often used interchangeably with the term "buckling."

C. Parameters affecting the critical buckling load of a structure.

We know that, critical buckling load is given by

$$P_{cr} = (\pi^2 EI) / (4L^2)$$

Here, π is constant.

E, I and L are variables which can control the critical load value.

In order to get the higher P_{cr} value

'E' and 'I' must be higher

'L' should lower

'L/D' ratio shall be minimum

Now, $I = \pi [D_o^4 - D_i^4] / 64$

So the value D value affects in quadratic.

Hence for design against the buckling loads, one should select the material having maximum E value, keeping length as minimum as possible, select maximum outer diameter in order to maximize the inertia value and lower the L/D ratio.

In case of tie rod we have carried out many such combinations / iterations that will give the maximum value of critical load with lowest invest of mass (optimized solution).

D. Sensitivity analysis:

In order to study the buckling performance of structure, each parameter affecting the buckling performance is studied. The inertia of structure is plotted for various diameters incrementally. The critical buckling load is also calculated for these diameters. When both the graphs plotted together, it is interesting to observe that, for small increase in inertia there is significant rise in the critical load. This signifies that, inertia is very sensitive to the critical buckling load. Hence it can be concluded that the structure should be selected such that it has maximum inertia value.

Further there can be other parameters also which are sensitive to the buckling performance of a structure. For circular cross sections inertia is quadratically dependent on the outer diameter. Hence if we change the outer diameter by small amount; there is significant change in the inertia value of structure.

Moreover the young's modulus of material is directly proportional to the critical load. Hence it is recommended to prefer the material having high value of young's modulus.

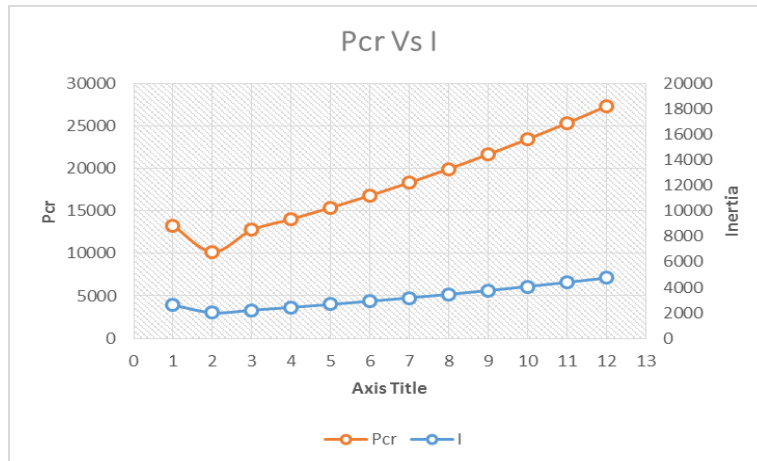


Figure 3: Critical buckling load (Pcr) Vs Inertia (I)

III. PERFORMANCE EVALUATION OF TIE ROD USING FEA

Nastran Linear Statics Analysis User's Guide give the fundamental of linear buckling analysis. Finite element approach including the effect of the differential stiffness to the linear stiffness matrix. The differential stiffness results from including the higher-order terms of the strain-displacement relationships. Eigenvalue extraction method used in linear buckling are Inverse power, Enhanced inverse power, Lanczos method. Assumptions and limitations of linear buckling analysis need to be consider before analysis.

A. Existing and Proposed Tie rod FE analysis.

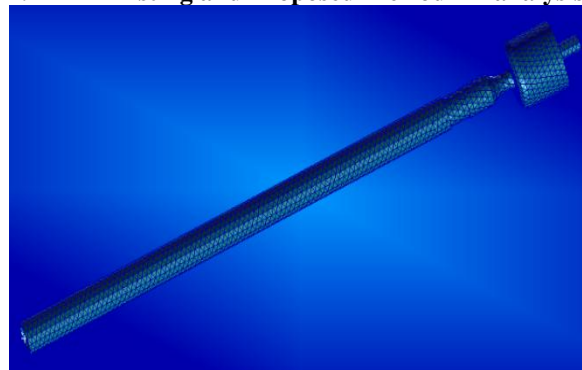


Figure 4: FE Model of Tie rod - DS3855241

Table 1: Structural details of Tie rod

Sr. No.	Quantity	Unit	Existing design	Proposed design
1	Outer diameter	mm	15.9	18.5
2	Inner diameter	mm	10.6	15.5
3	Length	mm	345	345
4	Thickness	mm	2.65	1.5
5	Mass	kg	0.503	0.422
6	Modulus of Elasticity	N/mm2	210000	210000

B. Extraction of Natural frequency and mode shape:

Table 2: Results of Normal mode analysis

Natural Frequency	Frequency (Hz)	
	Existing	Proposed
1	626.95	736.03
2	627.96	737.20
3	1636.36	1819.41
4	1654.7	1820.05

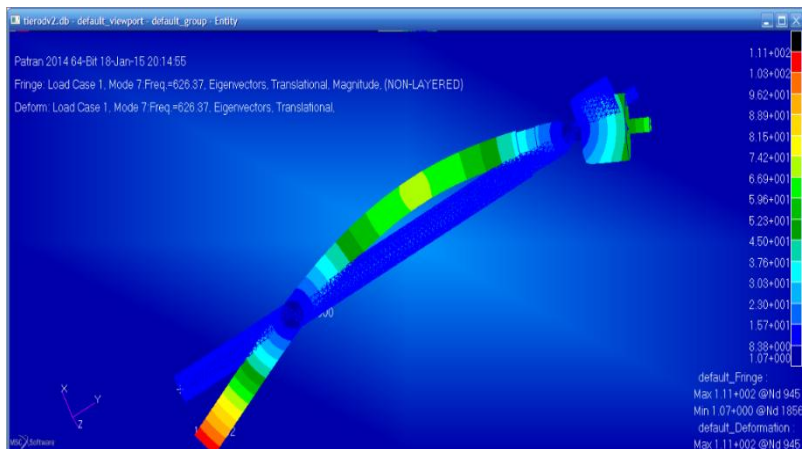


Figure 5. Eigenvector plot - First mode shape – 626.37 Hz of Existing design

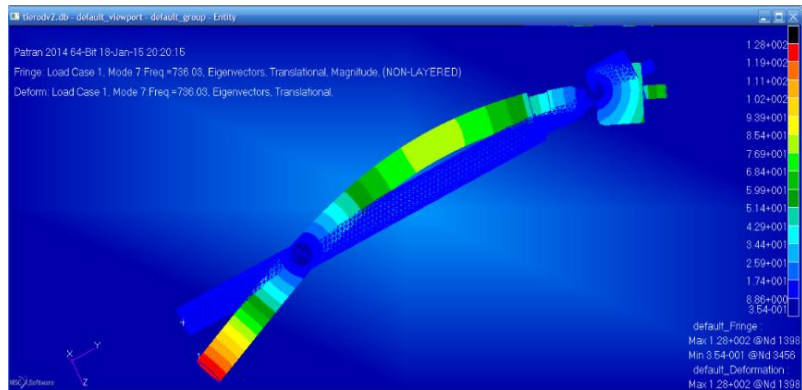


Figure 6. Eigenvector plot - First mode shape – 736.03 Hz of Proposed design

C. Critical buckling load evaluation

Theoretical calculations (approximate) for critical buckling load

Linear buckling of Euler column.

For clamped-free boundary conditions the critical load is:

$$P_{cr} = (\pi^2 EI) / (4L^2)$$

Where,

$$E=10.5E6,$$

$$I=8.333 \times 10^{-5},$$

$$L=320$$

$$P_{cr} = 13158.0$$

D. Calculating critical buckling load with SOL 105

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 Eigenvalue = 1.608350E+02

Applied load = 100 N

Critical buckling load (Pcr) = Eigenvalue * Applied load = 16083.5 N

IV. RESULT SUMMARY AND INTERPRETATION

A. Summary of results

Result Table 5: - Buckling load – Theoretical (Approx.)

Sr. No.	Quantity	Unit	Existing design	Proposed design
1	Critical buckling load	N	13158.27	14776.32

Result Table 6: - Linear buckling analysis – SOL 105

Sr. No.	Quantity	Unit	Existing design	Proposed design
1	Critical buckling load	N	16083.01	17707.04

B. Result interpretation

We have carried out the theoretical calculations as well as the FEA linear buckling analysis for tie rod. From the above result tables, we can observe that, the critical buckling load calculated from the theoretical method is comparatively lower than the linear and non-linear buckling analysis. This is due to the fact that, in theoretical calculations the tie rod is assumed thoroughly hollow and of uniform section. The variations in cross sections cannot be accounted in theoretical procedure. This is the limitation of theoretical procedure and the reason why we do the finite element analysis. As the FEA can take care of all these issues which analytical method cannot consider.

From linear buckling analysis the critical buckling load value obtain is around 16kN for existing design and around 17.7kN in proposed design. The rise in the critical buckling load is attributed to the rise in inertia value of proposed design.

Looking at the displacement plot of the first natural frequency of both existing and proposed design; we can observe that the location of maximum displacement is at the centre of tie rod.

V. CONCLUSIONS

A research starts with study of various types of sections proposed for design of tie rod. An objective of this research is a weight optimization. Here we are proposing the optimized design of Tie rod that has a minimum weight and maximum critical buckling load carrying capacity.

From results of normal mode analysis, it is seen that the natural frequency of existing design is 626 Hz and proposed design is 736 Hz. So there is ~ 18% rise in natural frequency of proposed design. Further the second and onwards the frequencies of proposed design are increased compared to existing design. So overall the dynamic performance of tie rod is being improved.

The weight of existing tie rod is ~ 0.498 Kg and proposed design is ~ 0.421 Kg. So there is approximately around 14% weight reduction is achieved. Apart from weight savings, it is interesting to note that the critical buckling load of existing design is 16.08 kN against critical buckling load of 17.7 kN of proposed design. Hence there is 10% rise in load carrying capacity of tie rod, which is quite significant.

VI. FUTURE SCOPE

1. In this research the tie rod has been evaluated under critical buckling load conditions. A linear buckling analysis have been carried out successfully. As we know the tie rod is a suspension component. Hence it is continuously exposed to fatigue loading condition. Hence the fatigue life estimation under full vehicle condition is recommended to be performed.
2. One can initially decide the expected fatigue life of tie rod component. Based on targeted life, an optimization can be performed to minimize the weight keeping the performance unaltered. This can be done with advanced tools like solver embedded fatigue.
3. Material having higher modulus of elasticity are suitable under buckling load conditions. Hence the composites or high carbon alloy steels having higher E value can be implemented.
4. Circular cross sections are most suitable under fatigue loading conditions. But other shapes of cross sections can also be tried e.g. elliptical shapes.
5. Completely hollow sections can be suitable for tie rod design. But for actual OEM, with the availability of good set up and for large scale manufacturing, full hollow shaft can be manufactured. Which can further reduce the weight.

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