ANALYTICAL DESIGN OF BUS PASSENGER TIE ROD

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ABSTRACT

This paper focuses on the study of buckling load on the Tie rod of steering system that undergoes an axial compression. Because of the external factors like road condition, different driving situations, different road adhesion, traffic conditions, vibrations and sudden jerks are sets up in tie rod. Tie rod generally buckle under the action of compressive force due to the large ratio of tie rod length to its radius of gyration. When it becomes worn out, steering will become more difficult and the vehicle will also typically be pulling or dragging to either side. Thus the aim of the project is to analyse tie rod for to improve the mass and buckling load of tie rod and to find out maximum deformation and stress. Present research is divided in two parts. First, to conduct survey amongst the buses, examine the causes of failure and second is to design and analysis to recommend best possible alternatives of Tie Rod with the aid of advanced design tools like CAD. Tie Rod failure is one of the major problems facing for MSRTC workshop supervisor.

Keywords : Buckling Load, Compressive Load, CAD, FEM, MSRTC Bus, Tie Rod

I. INTRODUCTION

Tie rod is the part of steering system of an automobile. Tie rod is slender in structure and used to tie and sustain compressive and tensile loads. The primary function of tie rod is to transfer the motion from steering arm to steering knuckle. Though maximum weight of vehicle is sustained by the suspension system, Fluctuating Forces and vibration due to bumps form automobile vehicle transmits to tie rod which may cause it to fail. The most percentage of forces on tie rod is compressive. Structural failure also takes place due to high severity of vibration and forces. Force required in static condition of passenger bus is more as compared to moving bus. Research shows that geometry and boundary conditions highly affected the magnitude of buckling and scatter of cylindrical column. Tie rod also contains imperfection in geometry and boundary conditions. So to find out buckling load is important in tie rod that undergoes critical compression in vehicle. The diameter of the rod is determined by considering the rod in two failure cases. 1] Compression failure 2] Buckling failure. So aim of this paper, to predict the buckling load analytical study will be done which gives a systematic approach for tie rod. Tie rod must therefore be strong enough to bear the stresses, deflection and vibration. In present work focus is given on optimization of existing tie rod by using suitable CAD software and its model analysis will also be done using ANSYS workbench.

Fig 1: Tie Rod for Passenger Bus

Fig 2 Tie rod Assembly (TCIC 1512)

II. LITERATURE REVIEW

V.D.Thorat, Prof. S.P.Deshmukh (2015) Developed rigid multi body dynamic analysis approach in design. The applications of this methodology simplify design process and give correct result. For the case study here work of design done on Ackerman steering mechanism for TATA tipper. In this first according to Ackerman conditions basic geometry is designed and then optimized it for static loading, modal analysis and then for dynamic forces generated on steering linkages while turning using Rigid Dynamics tool in Ansys. Results shown rigid dynamics approaches for design reduces time for optimization, simulation and provide the chance to take most corrective action. Author concluded that rigid dynamics approach is used in modern design techniques for various domains.

Prof Raghvendra K, Ravi K (2014) developed the Design iteration like different materials, shape, size and buckling load factor & studied theoretical, experimental and modal analysis of tractor Tie rod by analysis software. Author concluded with Results decreases displacement, stress, mass count and increases buckling Eigen value then buckling load for Mild steel SAE 1020 when compared to Aluminum A6016 and Cast iron C1540.

Mr. P M Chavan, Prof. MM Patnaik
(2014) studied buckling strength and compared performance of buckling for Tie rod for different materials. It is found that the mode shape, natural frequency , stiffness value and capacity for buckling load are high for carbon steel. So author concluded and validates that Carbon steel material is suitable for the manufacture of the Tie rod of vehicle as it shows better mechanical properties compared to cast iron and aluminum alloy material.

Mr. Patil, Prof. Chavan, Prof. Agade (2013) studied on the Tie rod that undergoes continuous vibrations when vehicle is running .The analysis of Tie rod is carried out with FEA to check its natural frequency, maximum stress analysis and deformation in Ansys software, Author concluded that the distribution of deformation and stress did not exceed the yield strength value and that there were neither damages nor failure of Tie rod. Author concluded that the tie rod was taken for analysis is safe.

Mr. Yongsheng Li (2013) study on steering tie rod for cracking with strength and buckling analysis theory, which showed a low risk of failure. A "necking down" method was used to optimize the length and location by Arc Length Algorithm, proved by pressing and impact test. Simulation results are consistent with tests. Concluded that "necking down" can alleviate cracking and improve quality efficiently on premise of mere decrease in pressure resistance.

Mr. Sankanagoudar, Dr. Amaranth, Prashant D., M.Thakur (2014) designed and developed mechanism for the deployment of Equipment panel of a spacecraft. Analysis is done in UG NX 7.5. The linear Buckling analysis method is used. Since, the tie-rod is the main drive element for tilting of the equipment panel, Analytical and FE results compared with experimental values & concluded that induced stresses are within permissible limits and hence, the design is safe.

Mr. Lei Zhang, En'guo Dong (2012) studied the comparison of the dynamics characteristics of multi-body rigid model and the rigid-flexible torque during bumping found maximum in three vehicle conditions i.e. the steering course, the driving straight course, and the bumping course, the torque during a bumping found higher. Finally author concluded that the stress due to bumping should be calculated to avoid failure. Coupling model built using the software of ADAMS and ANSYS.

Mr. Anthony DeLugan (2012) Developed nonlinear analysis method to accurately predict the buckling load of tie rod in axial compression. Tie rod end of an aerospace that contains two outer corrosion resistant (CRES) steel rod ends and hollow aluminium rod body is selected. Both a linear analysis and nonlinear analysis is performed. Results indicated that the Arc Length method among all three nonlinear analysis techniques is found suitable to accurately predict the buckling load.

III. IDENTIFICATION OF PROBLEMS

Maharashtra State Road Transport Cooperation (MSRTC) Passenger Buses has failure of tie rod due Cracking to and Buckling as shown in Fig: 5.1 & 5.2. As the Tie rod fails the bus will be pulling or dragging to either side. It is revealed that the Tie rod failure is genuine problem facing to MSRTC workshop supervisor. So it's important to design the tie rod for its strength.

3.1 Excessive compressive load in bumping

3.2 Unknown excitation due to vibration

3.3 Cracking of tie rod

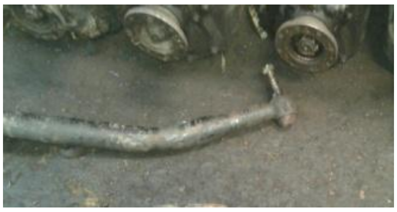

Fig. 3 Tie Rod Buckle

Fig. 4 Tie Rod Crack

IV. SCOPE OF PROBLEMS

When a ST-BUS moves on the road, the stresses will be developed in the Steering system due to road conditions, improper servicing, maintenance and lubrication which causes failure of TIE ROD, which is a frequent problem faced by ST-BUS. Due to the failure of TIE ROD .The present status of urban roads and its rapid detenoration are partly attributed to the excessive cost associated with maintenance of the Buses .Structural failure also takes place due to high severity of vibration and forces. Vibration and fatigue of Tie rod has been continuously a concern which may lead to failure if the resulting vibration and stresses are severe and excessive. Tie rod material is sometimes may lead to cracking of rod at assembly joint with tie rod end. Deterioration of material may be due to faulty material compositions, selection. When material becomes worn out, steering will become more difficult there by producing clunking noise. The vehicle will also typically be pulling or (dragging) to either side (left or right). Hence TIE ROD failure is genuine problem. This study helps to highlight the causes of TIE ROD failure and suggest the design modification to reduce failure of TIE ROD, this study will help to minimize the inconvenience to Passengers, ST-BUS Employee and Worker

V. STATIC ANALYSIS

- Out of the total weight of vehicle, fixed load of vehicle (mainly engine) is acted on front axle 4700 Kg while the pay load (variable load like material, passengers) is acted on rear axle.
- A steering system begins with the steering wheel or steering handle. The driver's steering For a MSRTC Bus under consideration, the torque on output shaft of gear box is. **4447.5 KN-mm**. This output shaft is connected to the drop arm (pitman arm) of length 250 mm whose other end is attached to the Push rod.
- Maximum force that can be transmitted by the drop arm is found out as,
- Force = moment / length = **4447.5 KN-mm / 250 mm = 17.79 KN**
- Push rod force is nothing but the steering arm force. Push rod is connected to the stub axle. So, the same maximum force is transmitted on the stub axle. Stub axle swivels around king pin which is 450 mm away from it. Generated torque about king pin is . TORQUE = 17.79 KN X 550 mm = **9784.5 KN-mm**
- Tie rod arm is connected to the king pin on one side and tie rod on other side through ball joints.
- Maximum force on tie rod which is at ball joint can be found out by
- Di viding torque by the length of tie rod arm 205 mm.
- Maximum force on tie rod = **9784.5 KN-mm / 205 mm = 47.729 KN**

VI. KINEMATIC ANALYSIS

Considering maximum weight on four wheel of Bus = Gross Vehicle Weight + Passenger Weight + Luggage Weight

= 150 KN + 30 KN + 20 KN

= 200 KN

Therefore, load on each wheel = 200 / 4 = 50 KN

Using coefficient of friction μ , Frictional force on tire exerted can be calculated as

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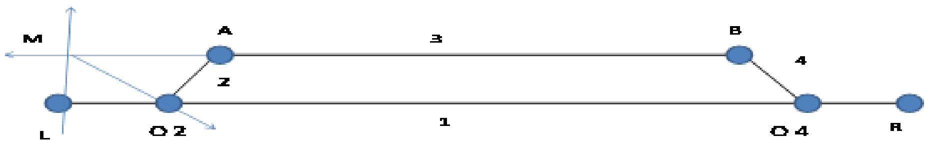
Frictional force = μ x 50 = F_R KN i.e. Force required to turn wheel is F_R KN.

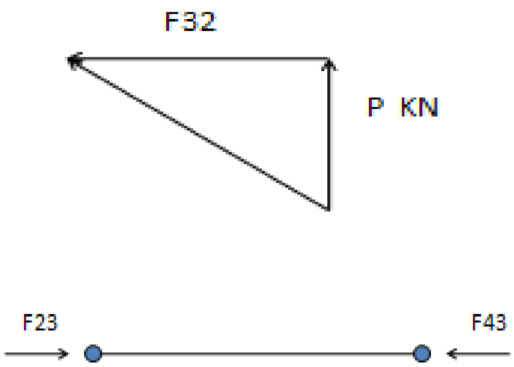
Considering different conditions as given below to find out Maximum Load on tie rod

Condition I: When Vehicle is straight

When vehicle is turning form straight to angular in rest position with F_R KN frictional forces, load on Tie rod is calculated with help of Triangle Law of forces



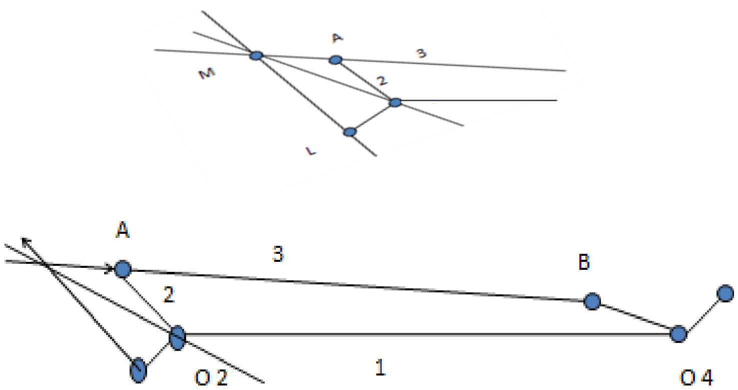
Condition I: Turning from Straight to angular Condition



Force polygon for Straight Condition

Condition II: When Vehicle is turning from extreme left to straight

When vehicle is turning form extreme left to straight in rest position with F_R KN frictional forces, load on Tie rod is calculated with help of Triangle Law of forces.



Condition II: Turning from Left to straight Condition

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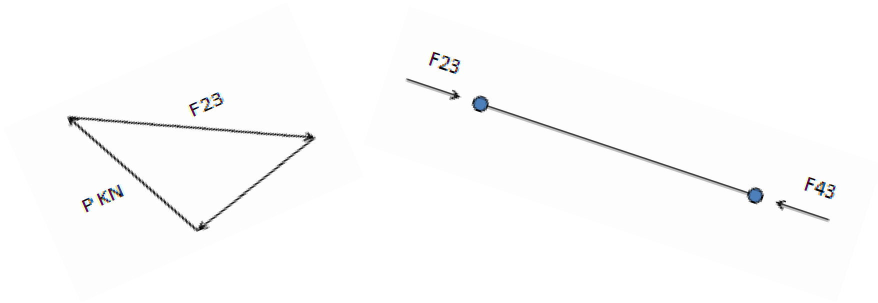
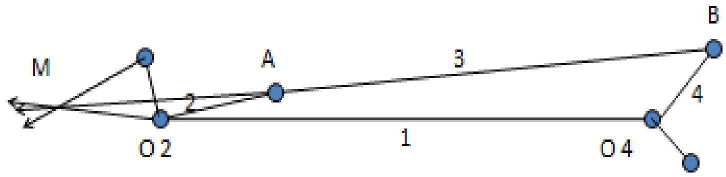


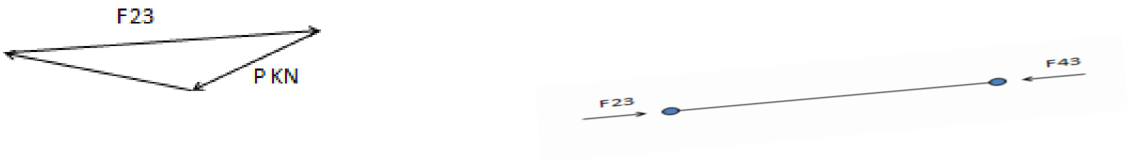
Force polygon for Left to straight Condition

Condition III: When Vehicle is turning from extreme right to straight

When vehicle is turning form extreme right to straight in rest position with F_R KN frictional forces, load on Tie rod is calculated with help of Triangle Law of forces .



Condition III: Turning from Right to straight Condition



Force polygon for Left to straight Condition

Similarly Force on Tie rod is calculated with different coefficient of friction 'μ' in all the three conditions

Parameters / Cases	μ	F_R (KN)	Load in Straight condition (KN)	Load in Left condition (KN)	Load in Right condition (KN)
1	0.3	15	9.83	19.33	33.33
2	0.4	20	13.11	25.77	44.44
3	0.5	25	16.3	32.2	49.55
4	0.55	27.5	18.02	35.44	52.55
5	0.6	30	19.6	38.6	58.06

Load on Tie rod in different Steering Conditions

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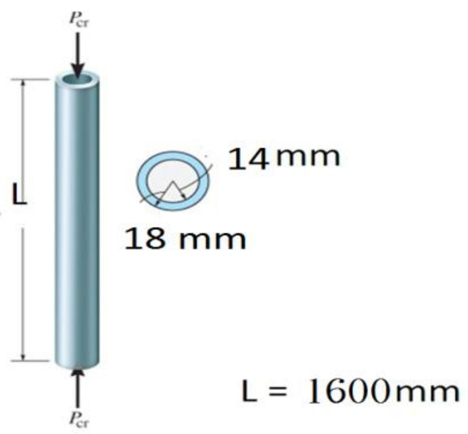
VII. ANALYTICAL CALCULATION OF EXISTING TIE ROD

L = 1600 mm , Outer Radius = 18 mm , Inner Radius = 14 mm ,Material: - AISI 1019 steel tube Modulus of Elasticity E = 200 GPa. End conditions: Pin-ended column. the maximum allowable axial load the column can support so that does not buckle is calculated as follows.



MATERIAL PROPERTY AISI 1019			
mechanical properties	symbol	unit	value
young's modulus	E	GPa	200
shear modulus	G	GPa	80
poission's ratio	ν	---	0.29
density	ρ	Kg/m3	7870
yield strength	σ_y	MPa	370
ultimate strength	σ_U	MPa	800

Now,

Solution: - Slenderness Ratio = Total Length / Radius of Gyration

Where, K = Radius of Gyration = (Moment of Inertia / Cross sectional Area) ^{½}

I = Moment of Inertia A = Cross sectional Area

I = (π/4) X $[0.018^4 - 0.014^4]$ = (π/ 4) X $[0.018^4 - 0.014^4]$ = 5.2276 X 10^{-8}

A = π X $[R_o^2 - R_i^2]$ = π X $[0.018^2 - 0.014^2]$ = 4.02123 X 10^{-4}

To find moment of inertia ,

K = (Moment of Inertia / Cross sectional AREA) $1^{/2}$ = (5.2276 X 10^{-8} / 4.02123 X 10^{-4}) $^{1/2}$ = 0.0114 m

Slenderness Ratio = Total Length / Radius of Gyration

S.R. = Total Length / K = 1.6 m / 0.0114 = 140.35

As Slenderness Ratio exceeds 80 for long column, Euler's formula will be used

Use of Euler's equation to obtain critical load with E_{st} = 200

$$P_{cr} = \frac{\pi^2 EI}{L^2}$$

$$= \frac{\pi^2 * \left[200\left(10^6\right)\text{ kN/m}^2\right] * \ 5.2276 * 10^{-8}}{(1.6\text{ m})^2}$$

$$= 40.308\text{kN}$$

Maximum force on tie rod through Kinematic analysis (56 KN) and static analysis (48 KN) calculations are more than Critical load of existing tie rod . Therefore tie rod buckle at extreme condition of steering and bumping. As frictional resistance increases load coming on tie rod will increase, as in punctured tire condition. So tie rod Design with AISI 1019 material is failed.

VIII. CONCLUSION

The analysis of TIE ROD **has** been carried out considering various steering conditions , frictional coefficient and various parameters like critical load, strength, stiffness and deflection. It is required to redesign it. It is decided to change the material and dimensions to withstand the required load. Further it is planned to validate the design using CAD and FEM.

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