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Design of a Low Alloy Steel Vehicle Tie Rod to Determine the Maximum Load That Can Resist Failure

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Abstract

The theory of durability and reliability was investigated on vehicle tie rods and it was found out that buckling is the major failure mode that hampers its longevity during braking, cornering and both compressive and tensile load acting on the vehicle while going through speed bumps. To determine the maximum load required for a typical tie rod material to buckle, low alloy steel was selected using CES EduPack 2013 software and this was done on the basis of the required material attributes and the loading conditions of tie rod during operation. Using CATIA software, both ends of the tie rods (inner and outer) were subjected to different load case scenarios obtained from ADAMS software. The load cases were analysed to find the maximum loads in both directions, capable of causing the tie rod to buckle or yield in operation and the analysis showed a maximum load of 18,563N. CATIA software was used to model several designs and analyse possible areas of stress concentrations on the tie rod. The hollow design was chosen as it meets the design objectives with a mass of 4.7kg which is not too different in real life and may not result in the performance of the design being compromise.

Keywords

Automobile, Brake, Cost, Failure, Load, Reliability, Safety, Suspension System

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1. Introduction

In automotive a tie rod is the most important part of steering linkage, this is a mechanical component used as a link mechanism to connect centre link to steering knuckle transferring forces to turn the wheel in conventional suspension system and rack to the steering knuckle in McPherson suspension system [10, 12]. One of the key parameter to determine the safety and reliability of automobile is the functionality of the suspension system. Therefore, it is vital that the tie rod operates reliably during exploitation and under severe working condition, and this depends on proper design concept for optimum performance under such condition. Tie rod is a spherical rod with threaded parts consisting of outer and inner ends as shown in Figure 1 [1, 6, 7]. This paper will focus on design analysis of high performance tie rod, intended to fail in a controlled way avoiding sudden or catastrophic failure and serving as a sacrificial component that yields while saving surrounding component of higher cost.

To operate the steering system for a change in direction of a car, the steering wheel is first turned to the desired direction. The movement of the steering wheel is transferred to the steering system through the wheel shaft, and then connects to the track rod to enable motion of the tie rod, which is linked to the steering arm for rotation of the wheels and tyres [3, 8, 13]. However, tie rods are made with slender structural rods which are more capable of carrying tensile load which in most cases results in failure at a certain compressive load exerted on the tie rod, particularly when the car is subjected to breaking, cornering or exposed to speed bump [14]. A typical example of loading effect on a tie rod is a car driving

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on a bumpy road, causing upward and downward movement of the wheels attached to the connecting rod. However, the magnitude in this case may depends on whether or not the tire is striking a huge bump or a tiny speck and in situations where huge bumps are encountered, the load impact can be minimised if the suspension system is in good condition [16]. The role of a car suspension system in such case scenario would be to increase the friction between the road surface and the tires to ensure steering stability with good control, vital characteristics that can save the driver and passengers as well as the tie rod from damage [5]. The study of loads or forces acting on a vehicle in motion is related to a property known as vehicle dynamics, and it is one of the major properties that every driver should be familiar with. In other words, vehicle dynamics is a part of engineering that is based mainly on classical mechanisms such as a vehicle steering mechanism which tie rod is one of the functional elements as shown in Figure 1 [2, 4, 5, 11]. Automobile engineers consider the dynamics of a moving vehicle mainly from two perspectives including Ride, a vehicle's ability to perform impressively in a bumpy road and Handling, a vehicle's ability to safely accelerate, brake and corner without inconveniencing passengers or affecting the car's performance [16].

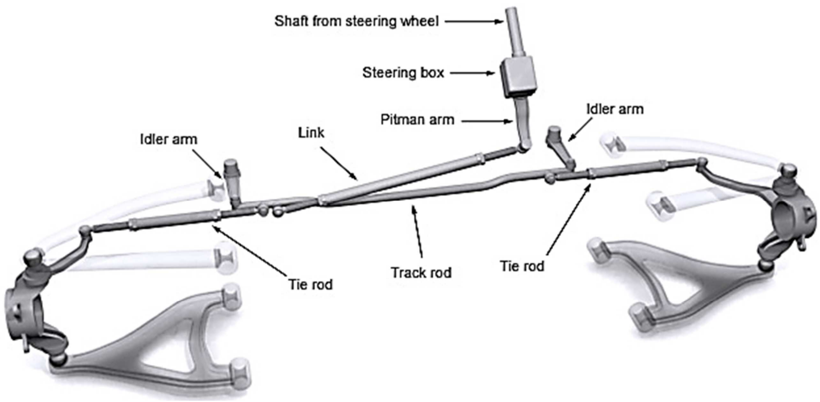

Figure 1. Typical Pitman arm steering system [9].

2. Methodology

The methodology was achieved by using ADAMS software to conduct simulation for a number of load case scenarios and the result obtained was adopted as the load applied on the tie rod using CATIA software for the design and stress analysis. Also, CES EduPack 2013 version was used for the material selection to determine a suitable material that can withstand the maximum load obtained from the simulation.

2.1. Load Case Scenarios on the Tie Rods by Using ADAMS Software

Durability in tie rods are measured by the ability of normal driving loads like acceleration, braking, cornering, etc. During driving, there are several scenarios which imparts load on the tie rod. In other words, the entire weight of the car settles on the wheel, and the loads is transferred to the tie rod through suspension system. The load cases can be divided into two; service loads which do not cause failure of the component and extreme load which leads to failure of the component. These loads are assumed to occur during a small time interval. Table 1 shows the different load case scenarios on the tie rods that were tested and the resulting load effective on the both side of tie rods (inner and outer) were calculated. The maximum results of these load cases as shown in Table 1 are used in this work as the main source of data.

Table 1. Different load case scenarios effects on the tie rods.

Load Case Scenarios	Fx (N)	Fy (N)	Fz (N)	Mx (Nmm)	My (Nmm)	Mz (Nmm)	Fz (N) (Vertical Max load) N	Load
1G Static	0	0	2,912	0	0	0	231.1326	Compressive
7G Bump	15,529	0	23,293	0	0	0	-13,340.3838	Tension
1.10G Brake	4,454	0	4,049	0	-1,336,205	0	-6,873.9779	Tension
Brake and Bump	16,287	0	24,430	0	-1,336,205	0	-16,970.6959	Tension
1.30G Cornering	0	-6,564	5,049	-1,969,138	0	0	1,040.8367	Compressive
Cornering & Bump	16,954	-6,564	25,430	-1,969,138	0	0	-13,654.3468	Tension
3G Berm	0	-22,711	7,750	-3,406581	0	0	1,740.064	Compressive
1.20G Acceleration	-2,005	0	1,671	0	0	0	2,159.7437	Compressive
Acceleration and Bump	12,696	0	22,052	0	0	0	-10,739.2577	Compressive
1.00G Reverse Brake	-1,878	0	1,878	0	563,256	0	13,197.1951	Compressive
Reverse Brake and Bump	-16717	0	22,259	0	563,256	0	18,563.7102	Compressive
4G Ditch Hook	0	27,036	774	8,110.908	0	0	-2,641.5033	Tension

ADAMS software was used to measure the impact of all load scenarios on the tie rod and the result is shown in the last two columns at the right hand corner of Table 1. The fundamental of ADAMS software is measuring the load distribution through the suspension which is not able to evaluate the failure at the component. Each load case scenarios was applied by ADAMS software and limited at 1 second time, and 500 unit steps through a static analysis in order to avoid transient response and spring oscillation. Each load case is acting on both sides of the tie rods (inner and outer) on the Vertical axis (Z-axis). Figure 2 shows the simulation carried out using ADAMS software to arrive at the Fz (N) (Vertical load) (Max) N. According to the ADAMS results as shown in the last two columns at the right hand corner of Table 1 and the positive values in Figure 2 represents compressive load while the negative values represents tension load. As shown in Figure 2, it can be observed that the 1G static in the load case scenario resulted in a minimum compressive force of 231.1326N, while 4G Ditch Hook scenario resulted in a minimum tensile load of about 2,641.5033N. Furthermore, it is concluded that braking cases has the maximum effects on the tie rods due to tension and compression. The static weight of the car has very low weight when compare with the loads on the tie rod and can be neglected. Therefore in this report only the two cases which are Reverse Brake and Bump scenario for compression and Cornering and Bump scenario for tension will be considered due to maximum values than the other which are about 18,563.7102N and 13,652.4787N respectively.

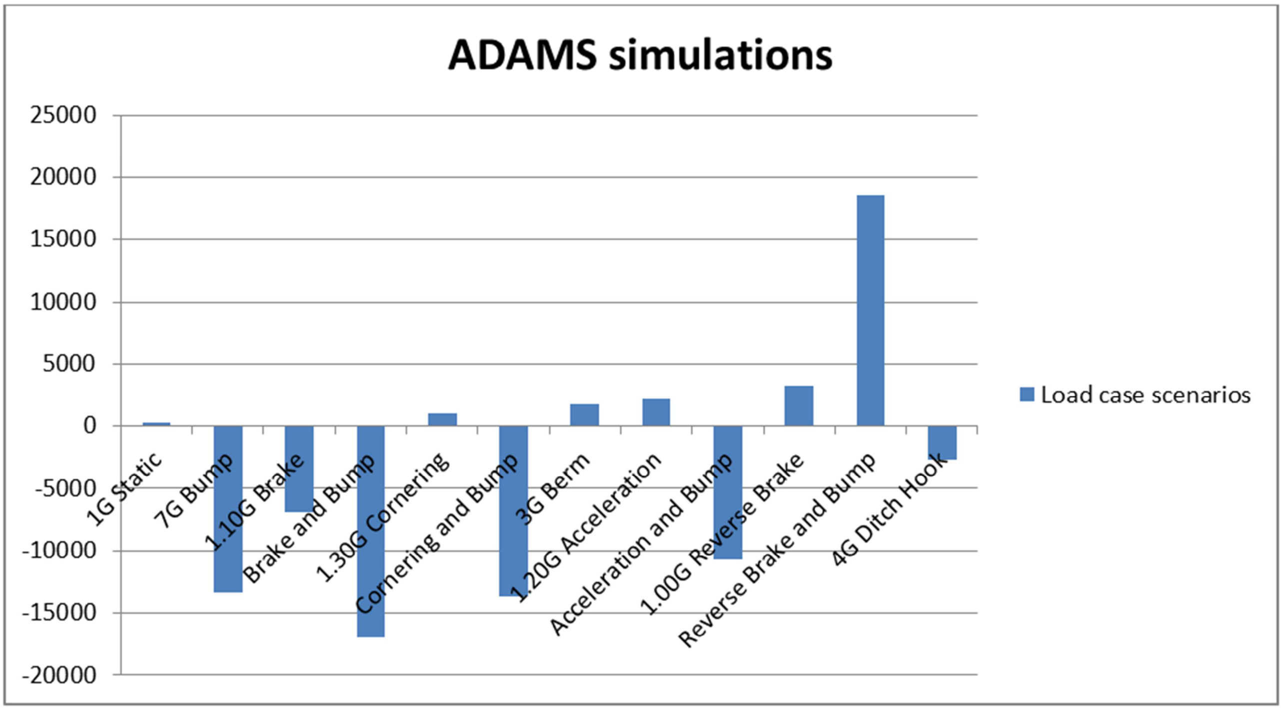

Figure 2. The ADAMS results based on the load case scenarios.

The designer must consider these forces during the designing a tie rod because these forces can cause fracture of the tie rod especially the compressive force which results in buckling. Furthermore, the lateral force and longitudinal force were reviewed during the ADAMS simulation. These forces could also cause bending of the tie rod, but its effect on the tie rod is negligible when compare to the loading in z-direction.

2.2. Materials Selection and Consideration for Tie Rod

According the CES EduPack 2013 software, there are several materials that can be selected for tie rod such as ceramics, metal alloy and composites. By using CES EduPack database level 2, a graph of young's modulus was plotted against density. As represented in Figure 3, only 28 out of 100 materials satisfied the requirements.

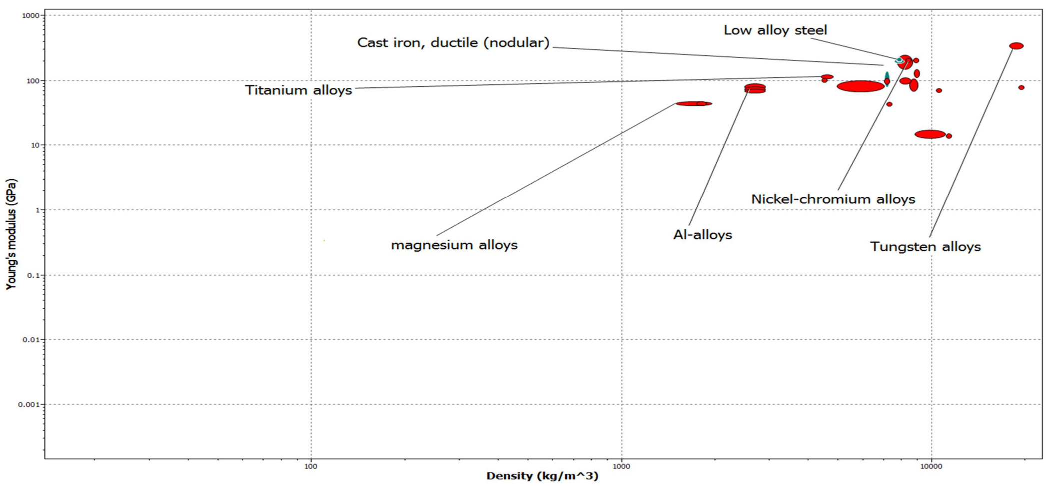

Figure 3. List of materials which is based on young's modulus and density (CES EduPack, 2013).

According to the above Figure, it can be seen that the Titanium alloys, Aluminium alloys, have a lower density than the remaining material. However, the young's modulus for them is lower than the low alloy steel. Titanium alloys are difficult to use as material for tie rod because it is too expensive. Even though, the density of low alloy steel is higher than others, but it has higher value in young's modulus. Another graph plotted for the choice of material and prices to achieve the suitable tie rod material is represented in Figure 4.

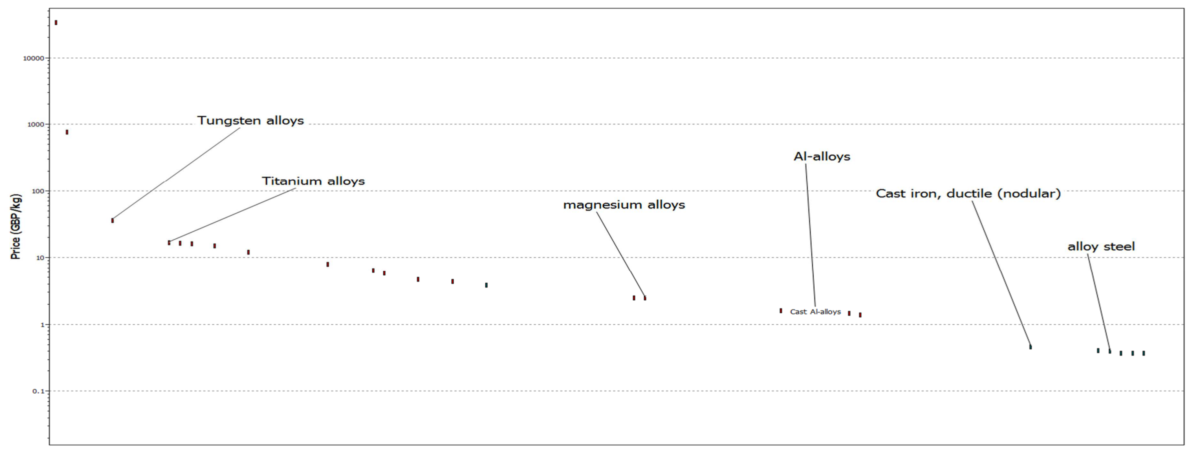

Figure 4. List of materials which is based on price (CES EduPack, 2013).

Table 2. Comparing the low steel alloys and Aluminium alloy properties (CES EduPack, 2013).

Attribute	Low Steel alloy	Aluminium alloys
General properties:		
Density (kg/m3)	7.8e3 – 7.9e3	2.5e3 – 2.9e3
Price (GBP/kg)	0.378 – 0.418	1.48 – 1.62
Mechanical properties:		
Young's modulus (GPa)	205 – 217	72 – 89
Yield strength (MPa)	400 – 1500	50 – 330
Fatigue strength at 107 cycles	248 – 700	32 – 157
Fracture toughness (MPa.m0.5)	12 – 400	18 – 35
Elongation (%strain)	3 – 38	0.4 – 10
Thermal properties:		
Melting point (°C)	1.38e3 – 1.5e3	475 – 677
Thermal expansion coefficient (µstrain/°C)	10.5 – 13.5	16.5 – 24

Comparing both Titanium and Metals alloy prices; it shows that titanium alloys are more expensive than the low steel alloy and Aluminium alloys. Although cast iron has low price, it cannot be used as tie rod material because it is more brittle than the others. Therefore low alloy steel and aluminium are the most suitable materials for the tie rod application.

From Table 2, it can be seen that low alloy steel is the most suitable material for the tie rod application than the Aluminium alloys due to the following reason:

i. The material price of low alloy steel is lower than aluminium alloys by four times, which leads to reduction in production cost.

ii. Tie rod should be ductile so that it can absorb energy and fails plastically

iii. The minimum yield strength of low alloy steel is eight times greater than aluminium alloys, which prevents the material from deformation

iv. The maximum fracture toughness of low alloy steel is lower than aluminium alloys by ten times, this helps to resists crack propagation on the surface.

v. The minimum young's modulus is high in low alloy steel as it is twice that of aluminium alloys. This helps to resist fracture even when the material is working in plastic zone.

vi. The density of low alloy steel is lower than aluminium alloys by nearly three times, this property can help to reduce the production weight. Due to these reasons, low alloy steel can be preferred as the suitable material for tie rods.

2.3. Pugh Method to Choose the Final Material

Table 3 shows the final material was selected using Pugh's decision matrix. The following weighted values (ranging from 0 to 5) were given to desirable characteristics which the material should have.

Table 3. Pugh method for material selection.

Material Characteristics	Weighted Value
Cost	5
Density	5
Young's Modulus	4
Elongation	4
Fatigue Strength	3
Fracture Toughness	3

Points were given to material characteristics based on the values above. The points were on a scale of 1 to 5 as follows:

5 – Excellent

4 – Very Good

3 – Fair

2 – Poor

1 – Very Poor

For each material characteristic, the point scored is multiplied by the weighted value given to that characteristic to get the total points due to that characteristic. The total points were then summed to find the better material as shown in Table 4 and 5 as follows;

Table 4. Low alloy steel points.

Characteristics	Weighted Value *(v)*	Points *(n)*	Total Points $(v \times n)$
Cost	5	4	20
Density	5	3	15
Young's Modulus	4	5	20
Elongation	4	4	16
Fatigue Strength	3	5	15
Fracture Toughness	3	4	12
Total Value Scored			98

Table 5. Aluminium alloy points.

Characteristics	Weighted Value *(v)*	Points *(n)*	Total Points $(v \times n)$
Cost	5	2	10
Density	5	5	25
Young's Modulus	4	3	12
Elongation	4	2	8
Fatigue Strength	3	2	6
Fracture Toughness	3	2	6
Total Value Scored			67

It is clear from the result of this performance matrix that low alloy steel is a better material to Aluminium for the tie rod. From the material review done and from the result of the Pugh's selection matrix, Low alloy steel is the material of choice and will be used for the design of the tie rod.

2.4. Other Considerations

Working Frequency and Natural Frequency

It is essential that the working frequency of a component is well below its natural frequency so as not to be excited into resonance. The working frequency of the tie rod can be gotten from equation 1.

$$f(\mathrm{Hz}) = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{1}$$

Where

f is the frequency in Hertz

k is the Load cell stiffness constant

m is the mass applied in Kg.

Hooke's law relates the force exerted on a spring material (the load cell) to the displacement produced:

$$F = Ke \tag{2}$$

Where

F is Force applied

K is the Load cell stiffness constant and

e is the displacement produced.

The fundamental frequency can then be written as

$$f(\mathrm{Hz}) = \frac{1}{2\pi}\sqrt{\frac{g}{e}} \tag{3}$$

Where

g is the acceleration due to gravity which is $9.81 m/s^2$.

The working frequency of the final design will be estimated using this formula and will then be compared with its natural frequency obtained using Catia FEA solver. The tie rods natural frequency should be sufficiently high as to be on reachable.

3. Design Concepts

Design From the theoretical analysis performed, the minimum diameter required for the tie rod to carry a load of 18564N without failure either by yielding or buckling was determined to be 12.8mm. For the first design, the diameter of the rod was made 15mm. Mesh size of 5mm was used to mesh the model before constraints and loads were applied. Since the load is compressive in nature, it was applied on the negative z-axis. Figure 5 shows the results obtained from the finite element analysis.

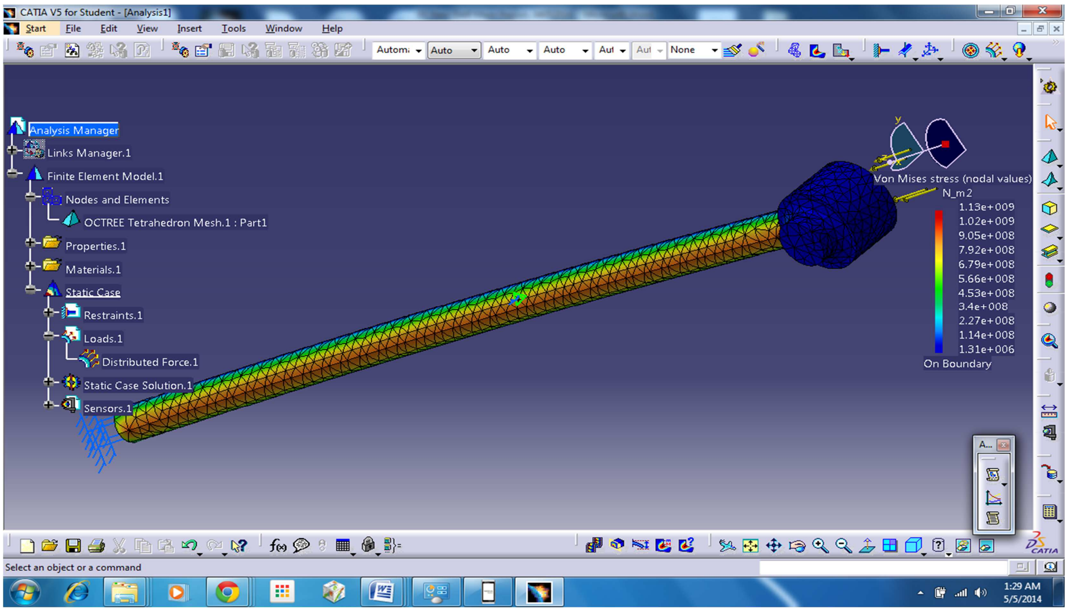

Figure 5. Analysis of Design 1.

With a maximum stress of 1130MPa, this design would be able to withstand the design load without yield. Buckling analysis was done to determine the Buckling load factor (BLF) of the tie rod model. Buckling Load Factor (BLF) was found to be 1.4, hence using the following equation

$$BLF = \frac{P_{cr}}{F_{applied}} \tag{4}$$

The critical Buckling load Pcr was determined to be 25990N. This load was applied to the model to see if it the stress it generates in the model will cause it to yield. If the stress generated is more than the yield strength of the material, the tie rod will fail by yielding instead of buckling. But if the stresses are less, the rod will buckle before it yield which is what is desired in this design. The stresses on the tie rod when the Pcr was applied are shown in Figure 6.

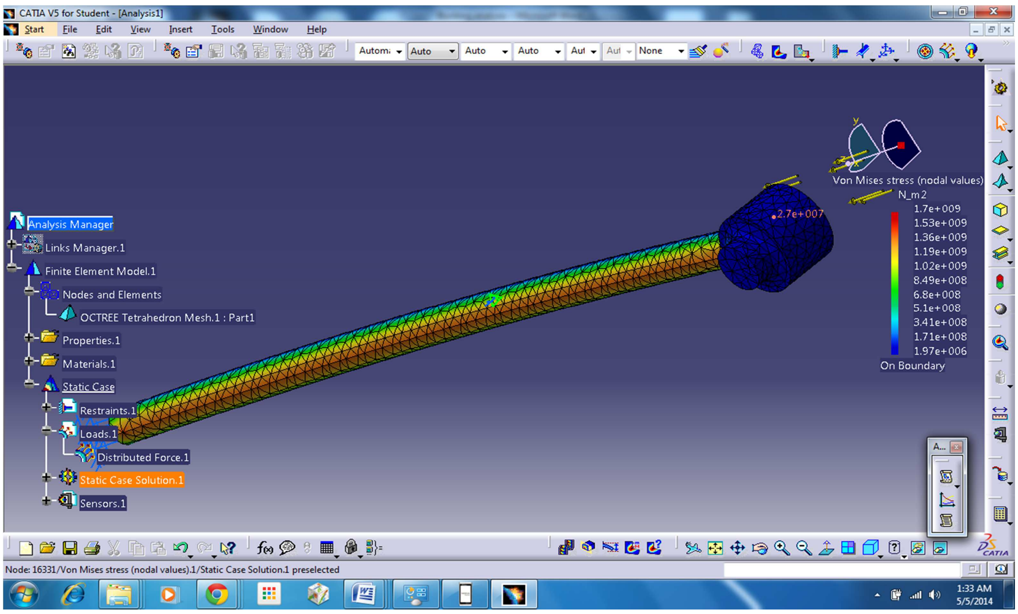

Figure 6. Stress analysis from critical buckling.

The stress generated by the critical buckling load in this case is 1700MPa. This is more than the yield strength of the material; hence the tie rod would yield before it buckles.

3.1. Design 2: Application of Arcs to increase the Tie Rod Length

From the expression for the critical load on a simply supported column,

$$P_{cr} = \frac{\pi^2 EI}{L^2} \tag{5}$$

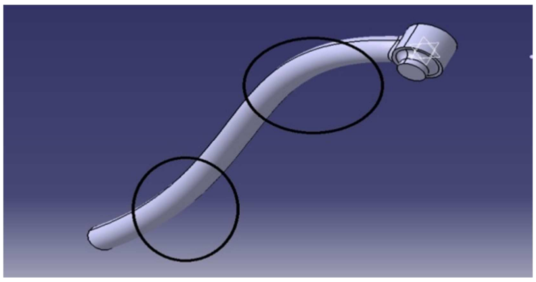

Figure 7. Bends on the tie rod.

It call be seen that increasing the length of the column decreases the critical buckling load. However, the distance between the two ends of the tie rod is a constraint in this work and is fixed at 383.3mm. Therefore in order to increase the length of the tie rod in the available desired space, introduction of arcs were considered. Hence, this conclusion informed the second design produced as shown in Figure 7.
Due to these arcs, the actual length of the tie rod became 437mm which was constrained in the 383.3mm design space. The design load of 18564N gave a stress value of 5150MPa which is far greater than the yield strength of the material. From the buckling simulation, a Buckling load factor of 1.15 was generated in the first buckling mode. Since the tie rod yielded at a load of 18564N, increasing this load will by the buckling factor will cause more yielding. Since this design will yield before it buckles, the second design was not an acceptable design. It was however noted that much of the stresses were at the lower part of the tie rod as shown in Figure 8.

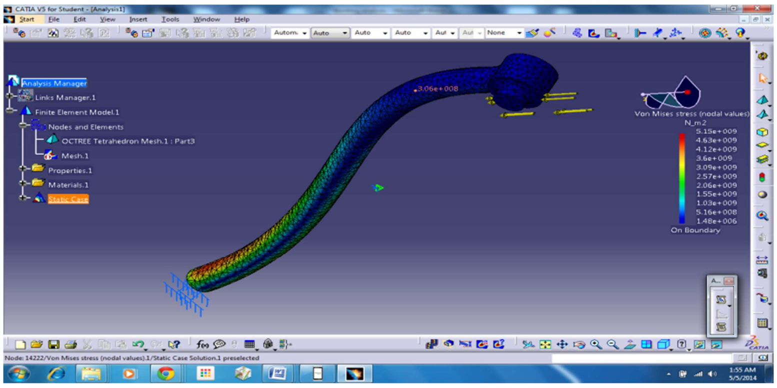

Figure 8. Analysis of Design 2.

3.2. Design 3: Application of an Arc to increase the Tie Rod Length

Design 3 is a modified version of design 2. In this design, only one arc is used instead of the multiple arcs used in design 2. The lower end of the tie rod is made of a circular rod while the upper end is made of an arc. Applying a design load of 18564N to the design gave a maximum stress of 299MPa as shown in Figure 9.

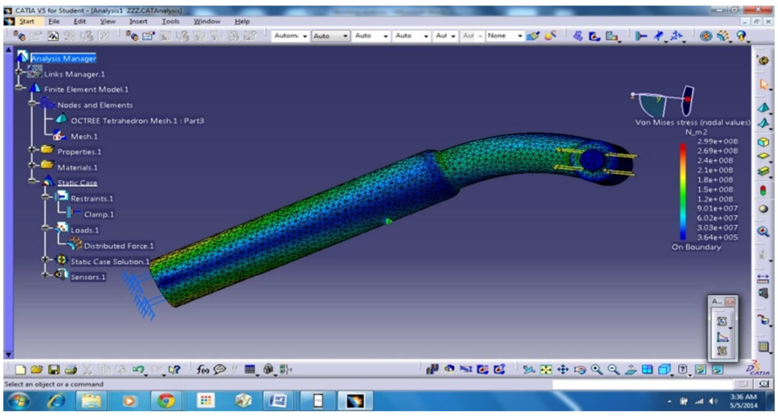

Figure 9. Analysis of Design 3.

From the buckling analysis, the first mode buckling load factor was determined as 12. This gives a critical load of 222768N. Figure 10 shows the stresses obtained when this buckling load is applied. It can be seen that the stresses above 1500MPa are found at the areas close to the constraint. The average stress which is more evenly distributed on the tie rod is about 1008MPa which is below the yield strength of the material. Taking this average value it can be concluded that the model will buckle before it yields and hence will be able to perform the required function.

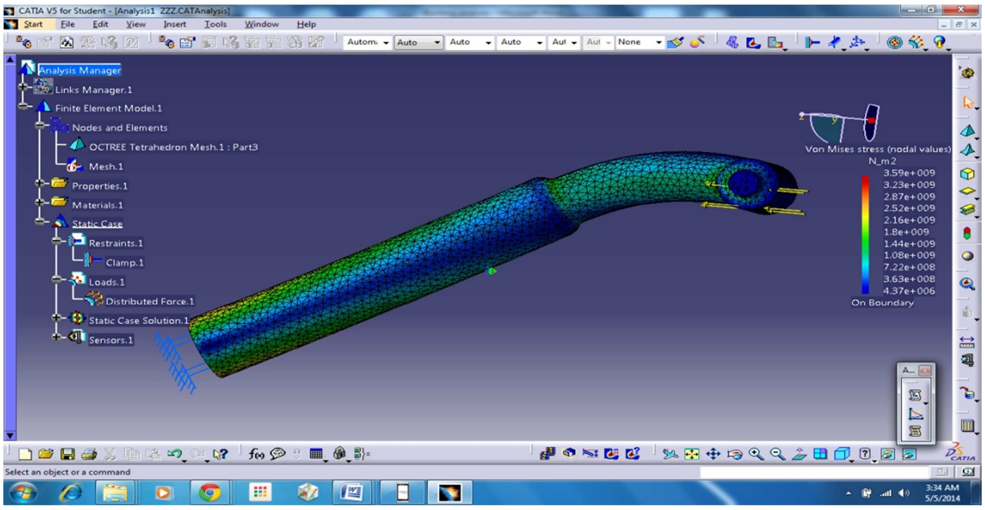

Figure 10. Analysis of Design 3 showing stresses obtained when the load is applied.

3.3. Design Optimization

To reduce the material needed to make the tie rod, it was optimized by making a hole of 25mm inside the lower part of the tie rod (ie the cylindrical part) which has a diameter of 40mm. This was done based on the knowledge that the materials closer to the centre of the cylindrical rod are not put under any stress. Second moment of area (I) was calculated to ensure it is not affected by the optimization done.

Hence

$$I = \frac{\pi}{4}\left(R^4 - r^4\right) \qquad (6)$$

Where

R is the outer radius = 20mm

r is the inner radius = 12.5mm

Hence

$$I = \frac{\pi}{4}\left(20^4 - 12^4\right)$$

$$I = 1.09 \times 10^{-7} m^4$$

This value of I is bigger than the required minimum of 1.28 * $10^{-9}m^4$. For the design load of 18564N, the maximum stress on the rod was 317MPa as shown in Figure 11 which means that the rod will not fail by yielding.

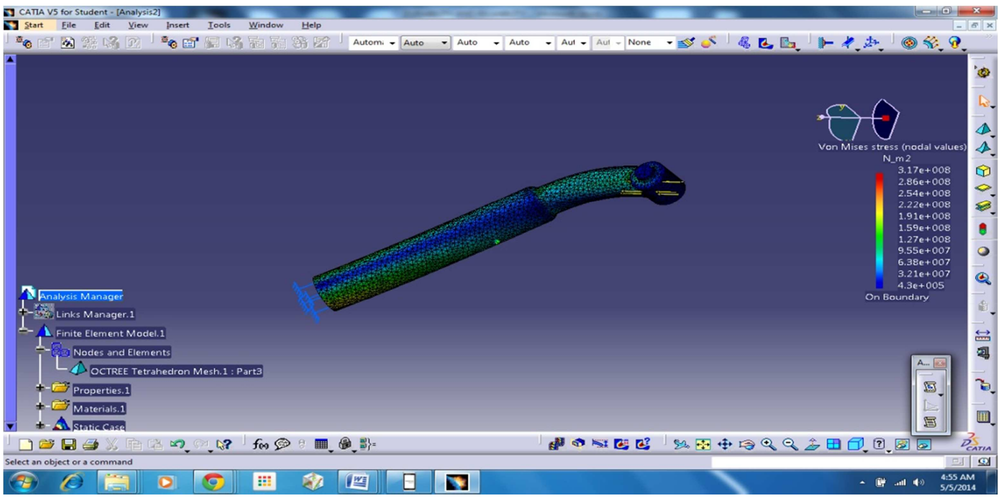

Figure 11. Result of stress analysis when the design load is applied.

When the critical buckling load of 222768N was applied stress result was applied as shown in Figure 12.

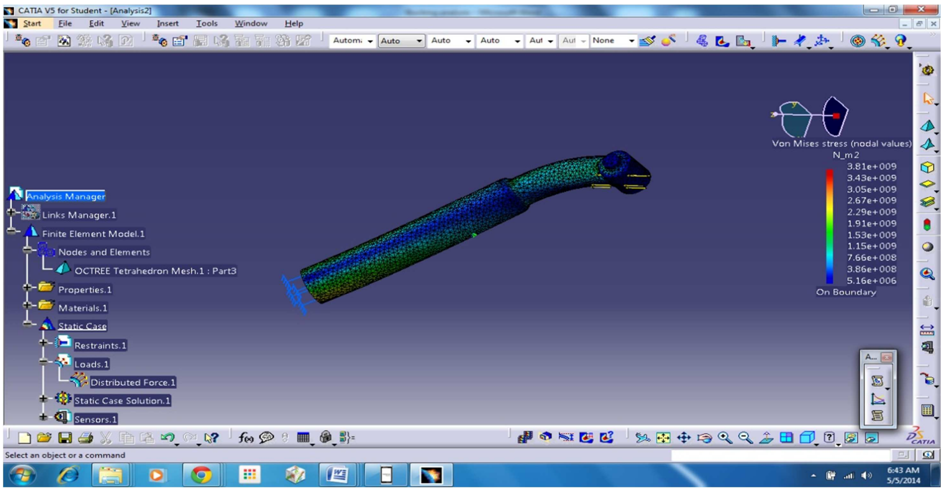

Figure 12. The critical buckling load when it is applying.

It can be seen that the average stress on the material is around 1500MPa. Since this stress value is about the yield strength of the material, the tie rod will first fail in buckling instead of yield.

Table 6. Masses of the tie rods obtained from the different designs.

Designs	Material	Mass (kg)
Design 1	Low alloy steel	4.5
Design 2	Low alloy steel	6.6
Final (solid)	Low alloy steel	5.6
Final (hollow)	Low alloy steel	4.7

3.4. Determination of the Working Frequency and Natural Frequency of Optimized Model: Determination of the First Mode Frequency

Catia FEA solver was also used to determine the first mode frequency of the tie rod. In obtaining this, the following steps were taken

i. The Catia model of the tie rod which has already been drawn and assigned steel properties was opened in the Generative Structural analysis window of Catia. On the pop-up dialog box that appears, “Frequency Analysis” is selected.

ii. The model was then meshed with a mesh size of 5mm.

iii. Restraint was applied to the appropriate end of the tie rod and the analysis run. See the Figure 13

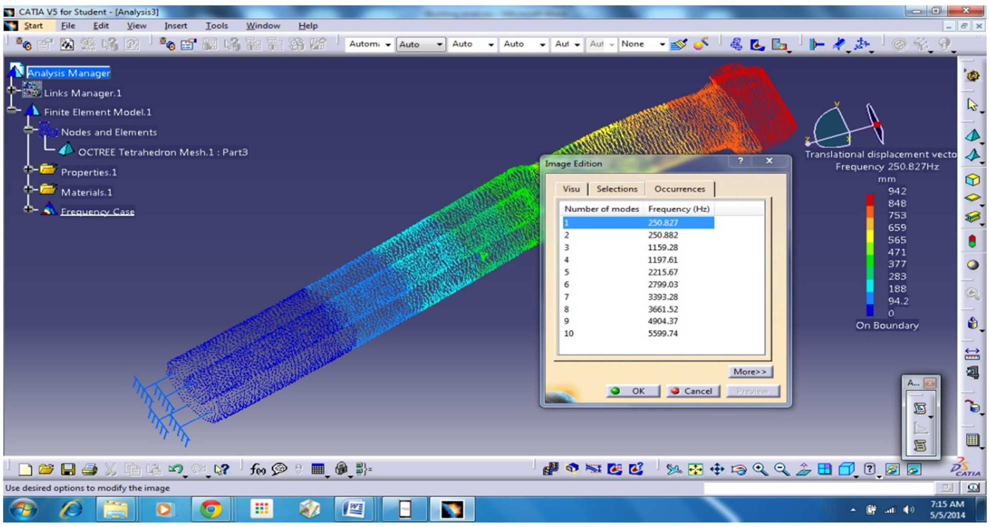

Figure 13. Catia FEA of first mode frequency.

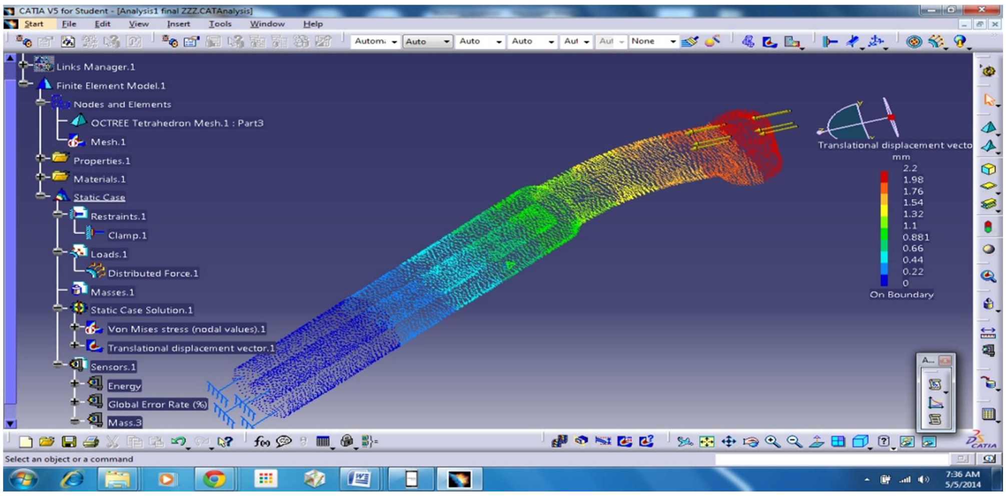

Figure 14. Results of displacement.

From equation 3, the fundamental frequency can then be written as;

$$f(\mathrm{Hz}) = \frac{1}{2\pi}\sqrt{\frac{g}{e}} \tag{7}$$

Where

g is the acceleration due to gravity which is 9.81m/s^2

e is the displacement produced.

From Figure 14, the value of the displacement is given as 2.2mm.

Hence

$$f(\mathrm{Hz}) = \frac{1}{2\pi}\sqrt{\frac{9.81\,m/s^2}{0.0022m}}$$

$$f = 10.63 \text{ Hz}$$

$$f = 11 \text{ Hz}$$

The estimated frequency of the final design is 11Hz compared to its natural frequency obtained while using Catia FEA solver which gave a value of 250.837Hz. Therefore, the natural frequency of the tie rod is high enough that the normal working frequency of the tie rod will not excite the tie rod into resonance.

4. Discussion

The Mechanical Fuse By making the upper part of the final design in form of an arc with a smaller diameter, the intention is to introduce a weak point through which the tie begins to buckle when loading exceeds the maximum design load of 18,563N. Since in buckling, the column follows the weakest path [15, 12], designing the upper end of the tie rod to be curved and to have less area than the lower end means that the tie rod would most probably follow that part during bucking. In this way, the fail safe purpose of the tie rod is achieved.

4.1. Reduction of Stress Concentrations

Stresses are normally high at bends or places where geometries change their shape. In the models procedure, fillets were used on sharp edges to reduce the localized stresses at those points and in effect increasing the durability of the tie rod as shown in Figure 15

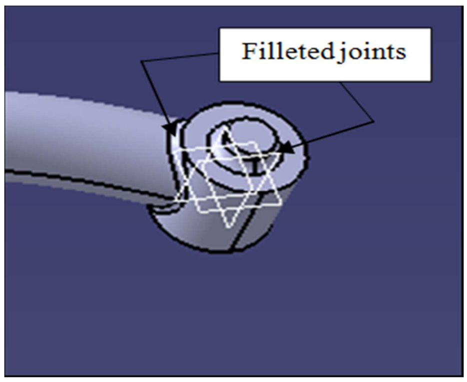

Figure 15. Filleted edges to reduce stress concentrations.

4.2. Justification of the Final Design Model

The tie rod produced at the end of the theoretical and numerical analysis is a simple model with an arc at the top part which is designed to be the point from which buckling will start when the design load is exceeded. The material used for the tie rod is low alloy steel which has a very minimal cost. The total mass of the optimised tie rod is about 4.7 kg which is not very different with what is obtained in real life. In terms of manufacturability of the tie rod component, many manufacturing processes can be used. The simplest would be to produce the hollow part via indirect metal extrusion process while the curved part could be made through hot forging process. The two parts can then be joined by welding and then heat treated to optimize the mechanical and physical properties.

5. Conclusion

In this paper, the performance of a vehicle tie rods was examined by using both numerical and theoretical calculation based on buckling load. The result of the data which was obtained from ADAMS software to arrive at various load case scenarios was used as a main source to find the maximum compressive and tensile stress on the tie rod. The low steel alloy which was selected by CES EduPack 2013 database level 2 when compared with Aluminium alloy based on all attributes was used as a material for manufacturing car tie rod. The hollow tie rod which was designed by CATIA software and after designing and comparing the three (3) tie rod designs (design 1, 2, and 3), the third design was chosen as the optimum design.

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