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Static Analysis on a Vehicle Tie Rod to Determine the Resulting Buckling Displacement

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Abstract: A vehicle tie rod manufactured with materials that can resist the vertical, lateral and horizontal forces acting on the suspension system when the car is in operation may last for a longer period of time provided the operating condition of the vehicle is such that, the tie rod material does not exceed its elastic limit. CES EduPack 2013 database level 2 was used in material selection of the tie rod which showed possible materials such as Nickel alloys, titanium alloy, aluminium alloy, low alloy steel etc. but low alloy steel was chosen based on the low cost, stiffness and yield strength. In terms of material properties, a tie rod requires high value of modulus of elasticity for stiffness, high fracture toughness against cracks and wear, and high yield strength against fatigue, and these properties were found in low alloy steel which conventional tie rods are manufactured from. The tie rod was designed using SOLIDWORKS 2012 version and static analysis was carried out to determine buckling displacements of a vehicle tie rod with a force of 18,563.7102N acting from each ends under pinned-pinned and fixed-pinned condition. Under the influence of this force, the tie rod in pinned-pinned position gave a maximum buckling displacement of 0.0156133mm whereas, tie rod under the influence of the same force buckled with a maximum displacement of 27.5852mm. This is because the pinned ends of the tie rod were sliding and exhibiting instability when the load (18,563.7102N) was applied from one end whereas, the fixed-pinned ends behavior of the tie rod was fixed and stable under the applied force from the pinned-end. The 18,563.7102N force load is the bump, braking and cornering force generated when a vehicle is in motion and was obtained from ADAMS simulation model of McPherson Subaru suspension system. Hence, the tie rod ends should be taken into consideration during manufacturing and installation, as buckling with low deflection can still carry more loads before the critical load is reached.

Keywords: Tie Rod, Vehicle, Suspension System, Buckling Displacement, Forces, Failure

1. Introduction

It is generally observed that aside the bad road (uneven roads) condition; drivers making use of road with smooth surface are still subjected to minimal degree of bounce and sway during motion, whereas, a vehicle driving on an uneven road surface characterised by pot holes and speed bumps is prone to higher degree of bounce and sway motion which do not only provide uncomfortable driving condition but also induce vibrational stresses on the suspension system which results in failure in failure of certain load bearing components in the suspension system [14] The suspension system which is a vibration dampening device comprises a coil spring, shock absorbers, wheels and linkages that connect the car to each of the wheels for relative motion. When the vehicle is exposed to bounce and pitch conditions while driving through uneven road, the linkages and other component of the suspension system undergo lateral and longitudinal load distribution which may affect the turning angle of the car wheel and the overall car performance in the long run [5]

Longitudinal acceleration force acts at centre of gravity causing a pitch torque which in turn increases the load of rear axle and lessens the load of the front axle. Also, lateral acceleration which is the cause of roll torque in a vehicle causes substantial increase depending on the severity [4, 5] and these loading effects has a direct influence on performance of the car linkages such as the tie rod. The varying load distribution as a result of lateral and longitudinal acceleration in a cause of journey can be expressed by vertical forces acting on tangentially on the tie rod. Most conventional cars operate with a steering system known as rack and pinion, which incorporates tie rods to aid in the movement of the wheels [6]. Tie rods are slender structural member that can be used as tie and only capable of carrying tensile loads. In automobile applications, tie rods are part of the steering mechanism that operates in both tension and compression, and because the ratio of a typical tie rod length to its cross section is very large, it would buckle under the influence of compressive load [9]. In other words, the working strength of a tie rod is the product of the allowable working stress and the minimum cross sectional area of the tie rod. Tie rods are coupled on the two ends of a steering rack and as the pinion rotates over the slotted rack, they function by pushing and pulling the front wheels as the steering turns either to left or right hand side of the car. In addition, a linkage relates primarily to assembly of links in which an output motion is transferred in response to a certain input motion in a specified direction [11]. In a typical front wheel suspension system, motion is transferred to the steering system through the drive shaft, connected to the track rod to allow motion of the tie rod which is linked to the lower control arm arm for rotation of the wheels and tyres [6, 8] as shown in Figure 1.

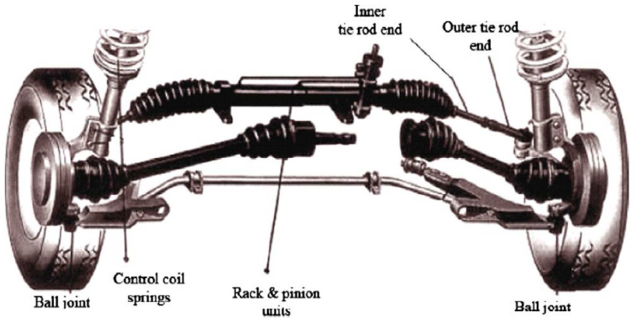

Figure 1. *Typical Steering System showing the Inner and Outer end of a Tie Rod [9].*

A typical tie rod in automotive system is made up of an inner and outer end which transmits motion from the steering centre link or rack gear to the steering knuckle which in turn caused the wheels to rotate as shown in Figure 1. The outer tie rod end is connected to an adjusting sleeve which allows the length of the tie rod to be adjusted when the car is in operation. This adjustment is useful while trying to set a car's alignment angle [8, 9, 13]. A sound tie rod plays a significant role in a car steering system performance as well as driver and passenger's safety, but the uncertainty about a typical tie rod is based on its longevity. Like most automobile components, tie rod does not easily go bad on their own, but depending on the loading conditions they are exposed, may result in wear and tear [4]. However, tie rods can last for relatively long period of time and most drivers may rarely replace them at all, but their longevity can be hampered as a result of certain driving conditions such as potholes, speed bumps or minor accident which may affect the tie rod performance. Due to its relevance in automobile systems and its high usage, routine inspection is recommended to prevent the occurrence of unforeseen accidents, and if replacement is required at the end of each inspection, both left and right set as installed on the front suspension of a vehicle is recommended and a complete four wheel balancing and alignment should be carried out at the same time to ensure safety of the driver and passenger (s) on highway [1]. Failure to carry out routine checks on the tie rod may result in certain warning signs such as incidents were the car pulls to one side while driving or when brake is applied. Also, bad tie rods can negatively affect a car's front and end alignment which may result in uneven wear of the inner and outer edge of a vehicle tire. Moreover, as tie rod ends are subjected to wear, it can slacken and loose gradually thereby, resulting in failure which may affect the drivers ability to steer and control the car [10, 13]. One of the most easily observed signs of a faulty tie rod can be squeaking noises or knocking sound from the front end of the car when the steering wheel is turned to a certain angle. Finally if the car has been exposed to any unsafe road conditions, or unusual contact with the front wheels such as driving across a big stone it may be ideal to subject the tie rod to inspection as faulty tie rods can also be responsible for instability of the steering wheel, vibration, looseness or a wandering or erratic feel on the steering wheel.

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

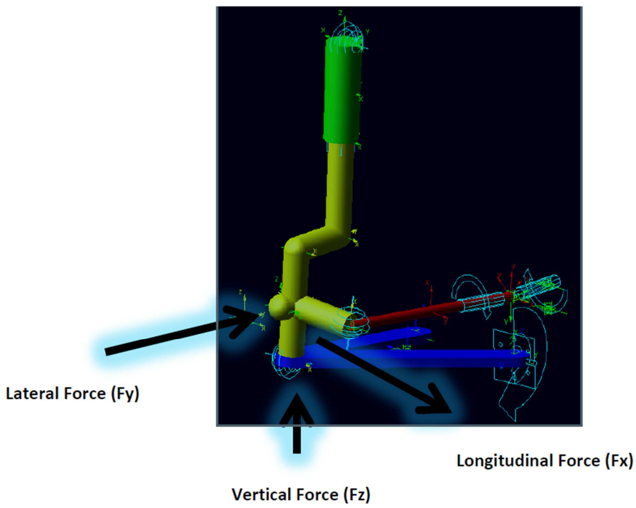

Figure 2. *McPherson Subaru ADAMS model.*

While driving through uneven road surface, the suspension system is subjected to several conditions that result in wear and tear of other components that makes up its frame work. However, the tie rod which is one of such components is prone to suffer from the vertical, horizontal and lateral forces acting on the suspension system when the road surface is filled with potholes and other obstructing conditions that will induce such forces on the vehicle suspension system. The in-service loading condition of a vehicle tie rod exposed to such terrain will gradually wear off, slacken and fail in the long run if appropriate inspection is not carried out. The load acting on the suspension assembly can be classified with respect to a co-ordinate system. As per the loads acting on the suspension system, it can be said that the tie rod will be subjected to loads from all the directions as shown in Figure 2. The loading conditions required for the tie rod analysis condition is with respect to the co-ordinate system of the tie rod alone and not of the whole suspension system. The tie rod co-ordinate system is as follows;

a. X-direction is radially outward horizontally
b. Y-direction is radially outward vertically
c. Z-direction is along the length of the rod

However, the tie rod is subjected to loading conditions which varies in direction, usually normal and tangential directions [3] known as normal stress (σ) and tangential shear stress (τ), expressed in Newton per square meter (N/m^2).

Performance of tie rods is measured by their ability to resist the operating forces acting on them during vehicle acceleration, braking, cornering, etc. Table 1 represent the simulation from ADAMS software, showing the different load cases acting on tie rods when the vehicle is in operation and the resulting loading effects on the tie rods (inner and outer) well calculated as shown on the Fz column of the Table 1.

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 determine the various loading effects on the tie rod and the result is presented in the last two columns at the right hand corner of Table 1. The basic reason for ADAMS software is to examine the load distribution on the suspension system and determines the load with the highest magnitude. Each loading scenarios was carried out using ADAMS software and limited at 1 second time, and 500 unit steps through static analysis in order to avoid transient response and spring oscillation. Table 1 shows results of the simulation using ADAMS software to arrive at the Fz (N) (Vertical load) (Max) N. From the results presented in the last two columns at the right hand corner of Table 1, the positive values represents compressive load while the negative values represents tensile load. From Table 1, it can be observed the 1G static loading gave a minimum compressive force of 231.1326N, while 4G Ditch Hook produced a minimum tensile load of about 2,641.5033N. Furthermore, it was also observed that braking load had the maximum effects on the tie rods due to tension and compression. Therefore in this report only the two cases which are Reverse Brake and Bump loading condition for compression and Cornering and Bump loading condition for tension will be considered due to their maximum values which (18,563.7102N and 13,652.4787N) can result in failure of the tie rod. Hence, the maximum results (18,563.7102N) of the load cases were used as the critical load in this paper.

2.1. Materials Selection and Consideration for Tie Rod

From CES EduPack 2014 software, a number of materials including ceramics, metal alloy and composites can be selected for tie rod application. By using CES database level 2, a graph of young's modulus was plotted against density. As shown in Figure 3; 17 materials out of 100 met the requirements.

Using CES to compare the material properties obtained in Figure 3, it can be observed that Titanium alloys and Aluminium alloys have very low density, while the density of nickel based alloys is twice that of titanium (8150-8350kg/m^3). In tie rod application, light weight is not as essential as stiffness due to the the loading condition of a tie rod during operation. However, the young's modulus for steel and nickel based alloys became outstanding and suitable for the application. Ideally, titanium and nickel alloys are difficult to use as material for tie rod because they are too expensive. The density of low alloy steel is moderate with high value of young's modulus as shown in Table 2. Comparing the prices of these materials as shown in Figure 4; it is obvious that low alloy steel and aluminium are the most suitable materials for the tie rod application. Although cast iron is less expensive, it is too brittle to be used as tie rod material.

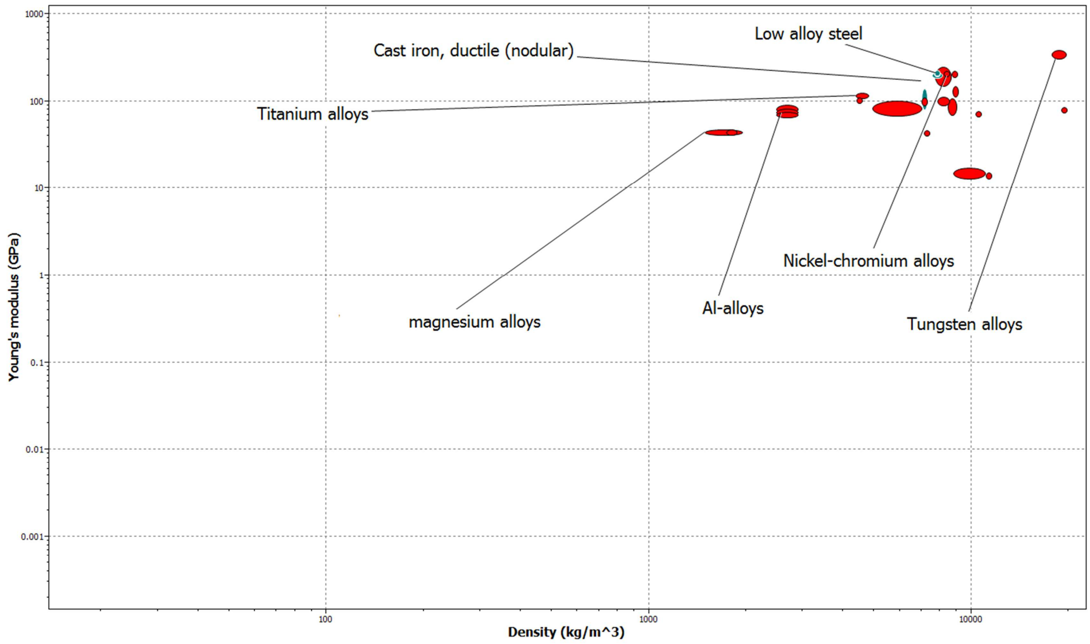

Figure 3. *List of materials which is based on young's modulus and density (CES EduPack, 2014).*

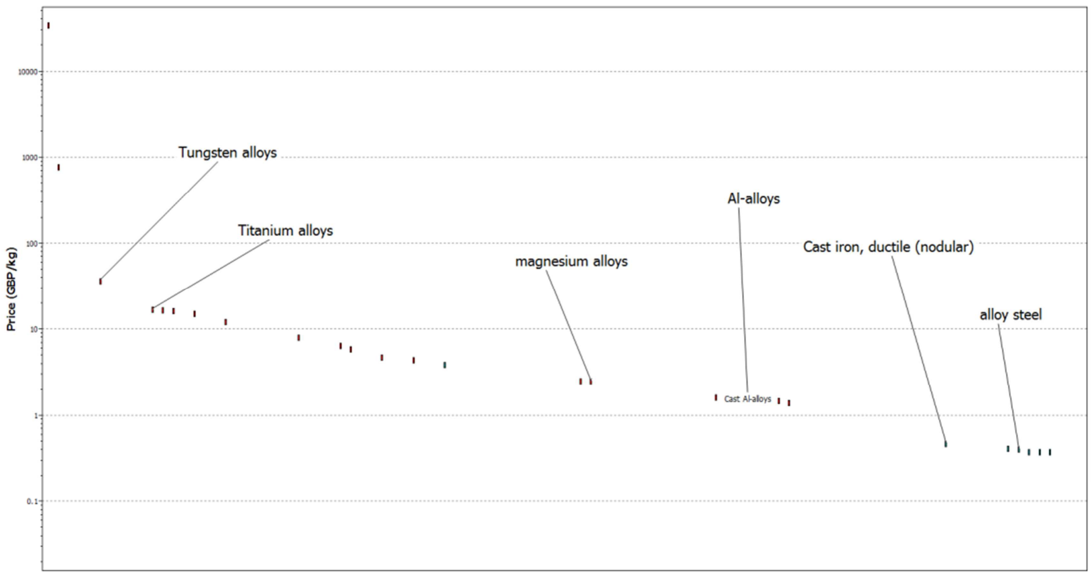

Figure 4. *List of materials which is based on price (CES EduPack, 2014).*

Table 2. *Comparing the low steel alloys and Aluminium alloy properties (CES EduPack, 2014).*

Attribute	Low Steel alloy	Aluminium alloys
General properties		
Density (kg/m^3)	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. m 0.5)	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

From the following material evaluations and comparisons, low alloy steel in terms of the price and modulus of elasticity proved to be suitable for tie rod application. Hence the design will be carried out with a low alloy steel material with properties shown in Table 2.

2.2. Operating Criteria for the Tie Rod

2.2.1. Natural Frequency

A member (tie rod) vibrates while operating under a time varying continuous loads, but the vibrations subsides after some times and this is known as natural frequency [12]. It is important for a component's working frequency to operate below its natural frequency in order not to be excited into resonance. In this paper, the working frequency of a tie rod can be obtained from equation 1 [2].

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

Where K is the stiffness constant and is given as

$$K = \frac{AE}{l} \quad (2)$$

Where
f is the frequency in Hertz
m is the mass (load) in Kg
E is the modulus of elasticity
l is the length of the tie rod
A is the cross sectional area of the tie rod

2.2.2. Fracture Toughness

The fracture toughness (K_{1c}) can be obtained from equation 3

$$K_{1c} = \frac{wl}{A}(\pi c)^{0.5} \quad (3)$$

From equation 3, A is given as

$$A = \frac{wl}{K_{1c}}(\pi c)^{0.5} \quad (4)$$

Where
K_{1c} is the fracture toughness
σ is the stress induced on the tie rod
c is a very small crack
wl is the total load on the tie rod

2.2.3. Fatigue Strength

Considering the fatigue strength of the tie rod,

It is important that the Fatigue strength endurance limit (σ_e) be as high as possible to enhance longevity of the tie rod during operation. Therefore, equation 5 and 6 should be taken into consideration

$$\frac{wl}{A} \leq \sigma_e \quad (5)$$

$$A \geq \frac{wl}{\sigma_e} \quad (6)$$

2.2.4. Buckling Load

If a slender bar (such as tie rod) of constant cross section is pined at each end, the applied compressive load P_{cr} (P_{cr} is also known as the Euler buckling load) that will result in buckling is expressed as

$$P_{cr} = \frac{\pi^2 EI}{l^2} \quad (7)$$

Where,
I is the minimum moment of inertia of cross-sectional area about an axis through the centroid.
l is the length of the tie rod

To apply equation 7 to other end conditions, it can be rewritten as

$$P_{cr} = \frac{C\pi^2 EI}{l^2} \quad (8)$$

In this case, the end condition constant (*C*) depends on the end conditions of a column which can be pinned-pinned, fixed-free, fixed-pinned and fixed-fixed of which a tie rod end may fall into one of this categories.

Using the relation,

$$I = Ak^2 \quad (9)$$

Equation 8 can be rewritten as

$$\frac{P_{cr}}{A} = \frac{C\pi^2 E}{\left(\frac{l}{k}\right)^2} \quad (10)$$

Where
$\frac{l}{k}$ is the slenderness ratio
k is the radius of gyration
$\frac{P_{cr}}{A}$ is the critical unit load (This is the load per unit area required to subject the column to unstable equilibrium).

Equation 10 shows that the critical unit load is mainly depended on the modulus of elasticity and the slenderness ratio. Hence, a column obeying the Euler formula manufactured with high strength alloy steel is not stronger than the one manufactured with low carbon steel provided *E* is the same for both [3].

2.2.5. Bending

Bending is one of the failure modes associated to a straight material subjected to varying load conditions of which a vehicle tie rod is one of such materials due to its in-service condition. Bending can be calculated from the generic beam bending equation given as;

$$\frac{M}{I} = \frac{\sigma}{y} \quad (11)$$

Where,
σ is the stress at the distance y from the neutral axis of the tie rod
M is the bending moment of the tie rod
y is the distance from the neutral axis
I is the second moment of area

In cases where the cross section of the tie rod design is rectangular, the second moment of area can be expressed as;

$$I = \frac{bh^3}{12} \quad (12)$$

However, the cross section of the tie rod design in this paper is cylindrical; therefore, the second moment of area can be expressed as;

$$I = \frac{\pi d^4}{64} \quad (13)$$

Where,
d is the diameter of the tie rod
h is the height of the tie rod
b is the width of the tie rod

3. Design of a Vehicle Tie Rod

A low alloy steel material was chosen for the tie rod design because of its properties as shown in Table 2. The design of the Tie rod was simple and the cross section was designed such that the manufacturing would not be difficult. The tie rod design details are as follows;

a. Object Name: mac _ front _ corner. Toe _ link. Toe _ link _ graphic
b. Object Type: Cylinder
c. Parent Type: Part
d. Cylinder Angle: 360.0deg
e. Cylinder Length: 383.2965079413mm
f. Number of Sides: 20
g. Cylinder Radius: 10.0mm

The tie rod design has circular rod, threaded part, outer and inner end which will be subjected to static analysis to determine the buckling displacement. One end of the tie rod will be fixed and the other end will be pinned and will be subjected to compressive force of 18,563.7102N. Also, both ends of the tie rod will be pinned and subjected to compressive force of 18,563.7102N. As shown in Table 1, this force is the bumping force and steering force obtained from ADAMS model simulation of McPherson Subaru suspension system shown in Figure 2. The design was carried out using solid works 2012 version as shown in Figure 5 and 6.

Figure 5. *Design of a Vehicle Tie Rod Using SOLIDWORKS 2012 Version.*

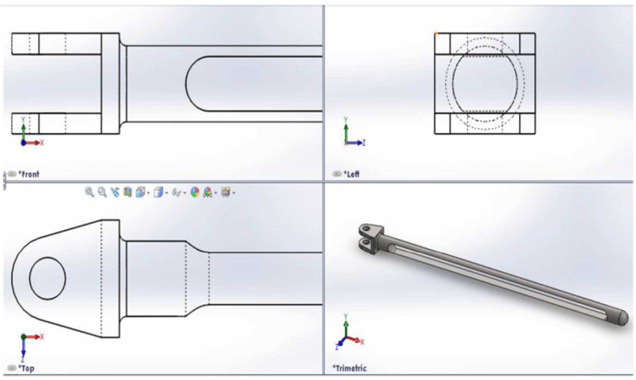

Figure 6. *3rd Angle Projection of the Tie Rod Design.*

3.1. FEA Analysis of the Tie Rod Design

A vertical slender member was subjected to axial compressive loading known as a column. If such a member is relatively short, it will maintain the straight nature under the influence of continuous axial load, and will eventually fail by direct compression prior to buckling whereas, a relatively long column undergoing the same loading condition will fail by buckling (bending). In other words, when the compressive load acting on the material reaches a maximum point known as "critical load" the long column will be subjected to bending effect in which the lateral deflection will become very large with slight increase in load. This phenomenon is known as "buckling" and can take place even when the maximum stress in the column is less than the yield stress of the material. In this case, a vehicle tie rod was assumed to be a column due to its straight slender nature. The loading conditions in which a column will buckle is influenced by the column length, material properties, and cross section and end conditions. Finite element analysis (FEA) is one of the most effective and popularly used engineering analysis tools for Non-linear problems.

3.2. Finite Element Meshing

Finite element meshing is one of the geometric input requirements while carrying an FEA and it involves discretization of a domain existing in one, two or three dimensions. In other words, Meshing is a process in FEA in which the model to be analyzed is divided into smaller discrete elements. The mesh sizes used greatly affect the solutions obtained in finite element analysis. Generally, discretization error reduces as the mesh is made finer and hence a more accurate result is obtained. However, the finer the mesh sizes, the more the computation time is required. As the mesh sizes are refined, the solutions obtained keeps changing until a point is reached when further refinement of the mesh produces little or no change in the solutions obtained after taking a long time for the solver to generate results. When further reduction in the mesh size has little or no effect on the result obtained, the mesh is said to have converged and the result at this point is taken as the solution to the problem [7]. Mesh generation in FEA can be achieved directly from a solid model for detailed part design in a three-dimensional (3D) CAD analysis model. Depending on the computational techniques the detailed solid model may affect the computational time or become too complex to analyse properly, some simplification with an appropriate idealization process such as reducing mesh size in the FE model was necessitated in order to reduce the long computation time. As shown in Figure 7, the mesh type used for the modelling was solid mesh and the mesher was standard mesh with absolute sag of 1mm. The Von Misses stresses obtained when different mesh sizes were used for the tie rod design was used to get the mesh convergence size. As shown in Table 3, 5mm was found as the mesh convergence point.

Table 3. *Mesh Sizes for the Tie Rod Design.*

Mesh Size	Von-Misses (MPa)	Displacement (mm)
50	504	13
25	500	13
10	476	13
5	514	13
3	520	13

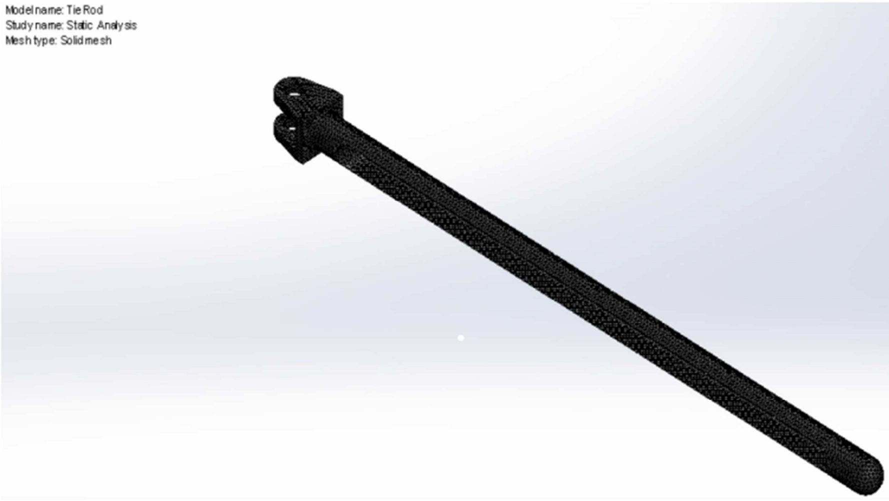

Figure 7. *Mesh model of a steering system Tie rod in SOLIDWORKS.*

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement	0 mm Node: 115	0.0156133 mm Node: 49766

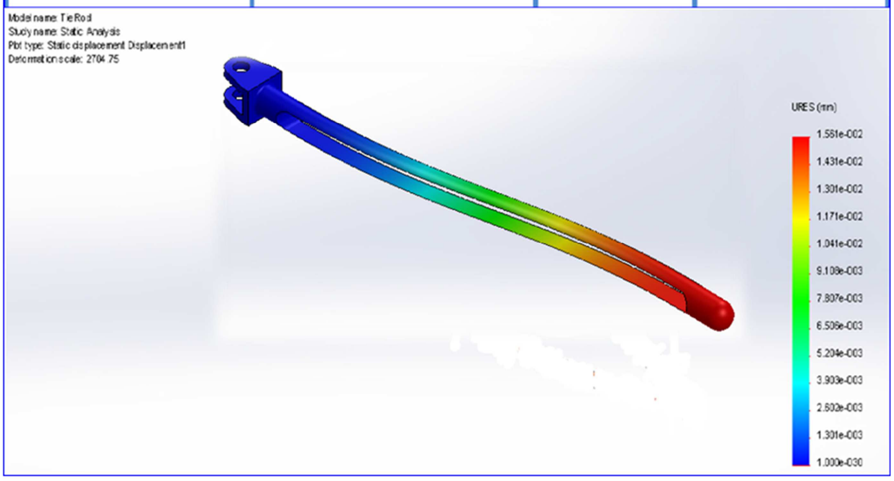

Figure 8. *Static Displacement of a Pinned-Pinned Vehicle Tie Rod Ends.*

Name	Type	Min	Max
Displacement1	URES: Resultant Displacement Plot	0 mm Node: 103	27.5852 mm Node: 320

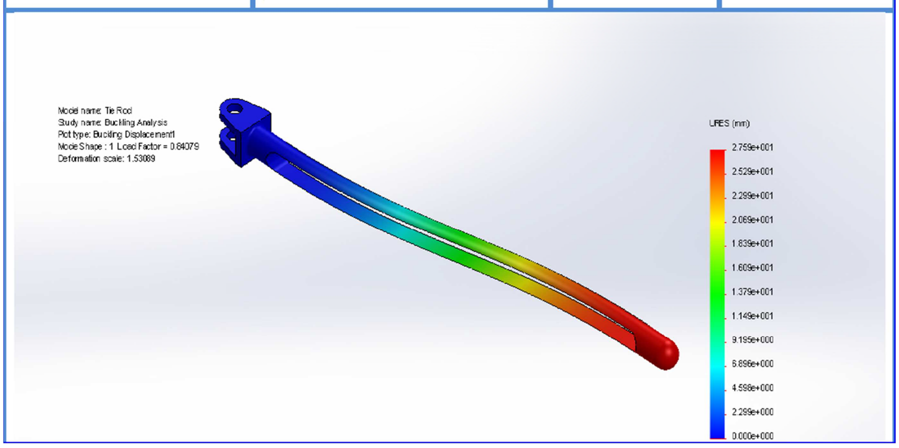

Figure 9. *Static Displacement of a Fixed-Pinned Vehicle Tie Rod Ends.*

4. Result

Result obtained from the static analysis is tabulated in Table 4.

Table 4. *Result obtained from the Static Analysis.*

Pinned-Pinned Displacement (mm)			Fixed-Pinned Displacement (mm)	
S/no	**Minimum**	**Maximum**	**Minimum**	**Maximum**
1	0	0.0156133	0	27.5852
Node			**Node**	
1	115	49766	103	320

5. Discussion

Mechanical failures in automobile are often attributed to wear and tear of two moving parts of a vehicle in contact, poor design or manufacturing defects, lack of proper inspection or maintenance etc. A typical example of vehicle parts where wear can easily occur is the tie rod ends of the steering linkage which wears and loosens with time due to the forward movement its impacts on the steering rack to enable rotation the front wheel of a vehicle rotate. The danger associated with incidents where tie rod ends fail and pull apart is that the wheel on the side of disengaged tie rod ends can abruptly be pushed as far back as possible, causing the car to veer instantly in the direction of the same wheel. If tie rod located on the right hand side of the suspension system fails, that can possibly pull the vehicle across the highway centreline and into the direction of incoming traffic. This can happen when the tie rod socket is badly worn and the ball stud loosens as a result of bumpy road condition, potholes or too many speed bumps on the road which can distribute the lateral, horizontal and the vertical forces (loads) acting on the suspension to other parts of the vehicle including the tie rod. In summary, this can result in failure of the tie rod when the maximum load (critical load) capacity is reached and the tie rod can no longer carry any further load. A vehicle tie rod is always in a static condition under the influence of various forces, particularly tensile or compressive acting from different directions. On normal operating condition, the behaviour of a tie rod with one end fixed and the other end pinned is different from condition where both ends are pinned (like in the case of most modern vehicles) and their failure mode which is usually buckling, due to compressive load will have different buckling displacement values as shown in Figure 8 and 9 and Table 4 respectively. As shown in Figure 8, result of the static analysis in pinned-pinned end condition of a vehicle tie rod subjected to maximum vehicle suspension load of 18,563.7102N acting on one end gave a buckling displacement value of 0.0156133mm. However, static analysis in the fixed-pinned end condition of a tie rod subjected to maximum vehicle suspension load of 18,563.7102N acting from the pinned end, gave a buckling displacement value of 27.5852mm. Comparatively, the pinned-pinned end produced minimal buckling displacement compared to the fixed-pinned end. This implies that buckling displacement with minimal value can still withstand subsequent loading effects, provided the critical load value or yielding point of the material is not exceeded unlike buckling displacement with higher values which in some cases may be at the point of failure or may fail suddenly. This property relates to the ultimate strength of the material which is the maximum stress that a material can withstand before it breaks or weakens [4]. In other words, failure will occur on the tie rod if the magnitude of the maximum shear stress in the part exceeds the shear strength of the material and this expression correlates with the analytical results obtained by Manik et al. [9], and the findings of Falah [4] in a failure investigation on a tie rod end of an automobile steering system. The result of the fixed-pinned end of the tie rod showed a maximum buckling displacement of 27.5852mm due to 18,563.7102N load acting on the suspension system. Considering this scenario, it is important to note that this load is not generated instantly, but rather increases gradually till a point is reached where the tie rod and other linkages connecting the wheel and the steering mechanisms in the front suspension can no longer carry applied loads and therefore fail. Kim et al. [8] attributed this phenomenon to failure caused by repeated loading on one end of the tie rod pinned, with the other end fixed. As mentioned earlier, load acting on one end pinned with the other end also on the pinned position will cause a sliding movement in response to the loading effect which will end up in minimal buckling displacement. However, it is important to note that repeated loading often initiates brittle cracks, which grow and result in failure in the long run. The cracks always start at areas with high stress concentrations, particularly changes in cross-section of the component, near holes and notches at nominal stress levels far lower than those specified for the strength of the material.

6. Conclusion

Result of the ADAMS simulation model was obtained from ADAMS software and was used as the primary data to finding the maximum compressive and tensile stresses on the tie rod. Low steel alloy was selected from CES EduPack 2013 database level 2 and when compared to aluminium alloy based on few salient attributes, low steel alloy was chosen as desired material for manufacturing the vehicle tie rod. The tie rod was designed with SOLIDWORKS 2012 version and static analysis was carried out on the tie rod design which showed that a vehicle tie rod can still carry more load provided the yielding point of the material is not exceeded. This analysis correlated with hooks law on elastic materials.

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