

FEM Analysis of Universal Joint of Bharath Benz DE210BS4

Ananthu S Kumar¹, Rahul P John²

¹Mtech Student, Sri Vellappally Natesan College of Engineering, Mavelikara, Kerala

²Assistant Professor, Sri Vellappally Natesan College of Engineering, Mavelikara, Kerala

Abstract - The power generated in the engine as the result of combustion of fuel is transformed to wheels through the transmission system in the vehicle. This system mainly consist of gear box, propeller shafts, universal joints and differential gears. The drive shaft imparts torque from the engine. The failures that caused to the driveshaft is mainly due to vibrational stress, fatigue, bending stress or even the misalignment of the driveshaft. All these defects are mainly due to the overloading or even repeated overtime use. Our project mainly aims at testing the universal joint at different working angles and different materials so that the time to time failure of the universal joint can be reduced to some extent.

Key Words: Universal joint, ANSYS, Total deformation, Equivalent stress, Vibration.

1. INTRODUCTION

Universal joints are capable of transmitting torque and rotational motion from one shaft to another when their axes are inclined to each other by some angle, which may constantly vary under working conditions. Universal joints are incorporated in the of vehicle's transmission system to perform three basic applications: (a) Propeller shaft end joints between longitudinally front mounted gearbox and rear final drive axle. (b) Rear axle drive shaft end joints between the sprung final drive and the unsprung rear wheel stub axle. (c) Front axle drive shaft end joints between the sprung front mounted final drive and the unsprung front wheel steered stub axle. Universal joints have movement only in the vertical plane when they are used for longitudinally mounted propeller shafts and transverse rear mounted drive shafts. When these joints have been used for front outer drive shaft they have to move in both the vertical and horizontal plane to accommodate both vertical suspension deflection and the swivel pin angular movement to steer the front road wheels. The compounding of angular working movement of the outer drive shaft steering joint in two planes imposes large and varying working angles even when the torque is being transmitted to the stub axle. Due to the severe working conditions, special universal joints known as constant velocity joints are employed. These joints have been designed to absorb torque and speed fluctuations and to operate reliably with very little noise and wear having long life. The main types of universal joints are:

1.1 Cross-Type Joint

In this type of joint , the two yokes are arranged at right angles to each other and are connected by using a t union joint, which is cross shaped. Hardened steel cup is provided in each arm of the yoke. the bottom of the cup forms a contact with the end of the block, for the alignment of t union.

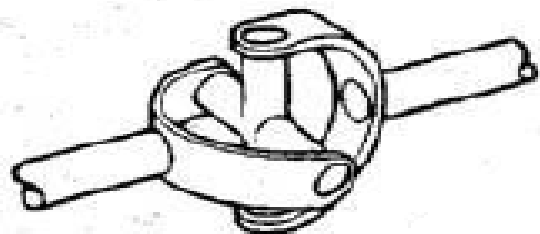


Fig 1.1 Cross-type joint

1.2 Moulton Joint

This joint is based on hooke type coupling. This joint uses the moulded type rubber coupling for the transmission of power between the coupling. No additional lubrication is required for the synthetic rubber coupling. The torsional shocks are damped due to its high flexibility and hence the drive is transmitted through the coupling.

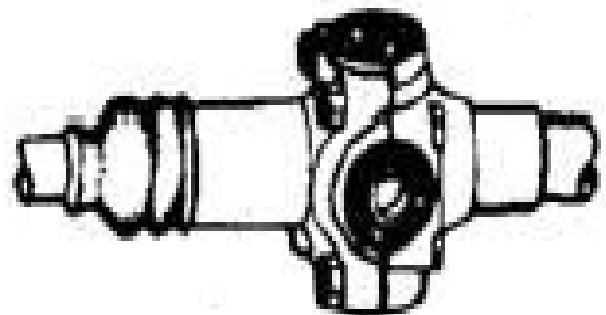


Fig.1.2 Moulton Joint

2. NEED FOR THE STUDY

The load carrying trucks are mainly subjected to overloading to meet their daily targets. The engine taken here is of Bharath benz truck having output power upto 220 hp. These

power is transmitted to wheel through a drive shaft mechanism containing a universal joint (cardan joint) in between them. The main failure occurs to these joints either by fatigue, velocity changes, etc. So in this study a cardan shaft of the engine is taken and its deformation and equivalent stress is analyzed in different operating angles and different materials and the result is analyzed for further study.

3. SCOPE AND OBJECTIVES

The scope of the present study is to perform the analysis of total deformation and to determine equivalent stress using ANSYS 19.1(workbench).

The objectives are:

- 1) To conduct a study on total deformation occurs on the cardan shaft in different working angles and different materials by applying different torques of the order of 450Nm, 1000Nm and 35000Nm.
- 2) To conduct a study on equivalent stress occurs on the cardan shaft in different working angles and different materials by applying different torques of the order of 450Nm, 1000Nm and 35000Nm.
- 3) Comparison to find out the best working material at the specified working angles.

4. METHODOLOGY

A cardan shaft which is made up of steel and is attached to the gearbox assembly of the specified engine is considered for the analysis. The shaft is allowed to perform its operation in different angles ranging from 5 degrees to 40 degrees is analysed. The different operations performed are as follows:

- 1) Selection of the model of cardan shaft to be analysed.
- 2) Application of different torques to the cardan shaft.
- 3) Application of the torques at different working angles ranging from 5 degrees to 40 degrees.
- 4) The analysis is conducted on different working materials such as Steel ,AISI 4140,Carbon fiber, SAE 1130.
- 5) Total deformation is analysed.
- 6) Equivalent stress is analysed.
- 7) Results are compared.

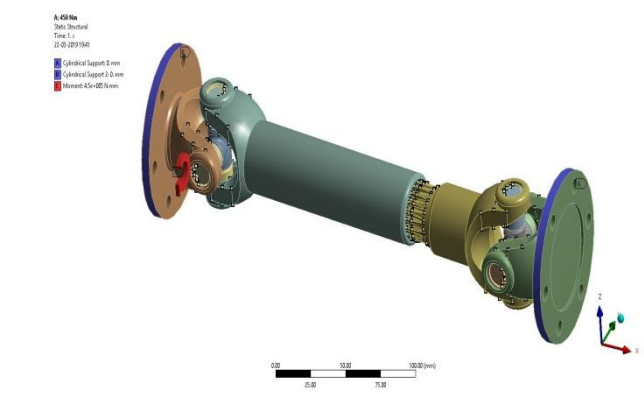


Fig.3: Model of Cardan shaft

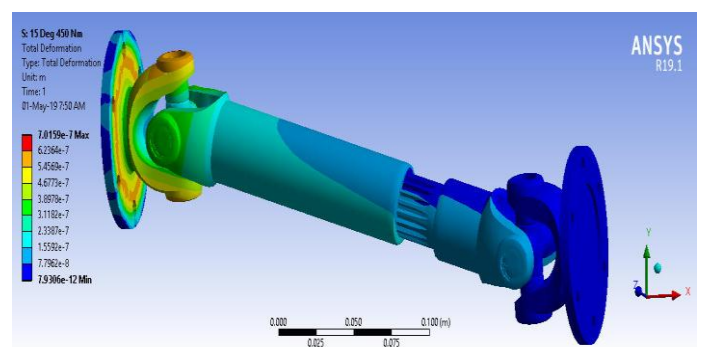


Fig. 4: Total deformation obtained on SAE 1130 with a torque of 450 Nm

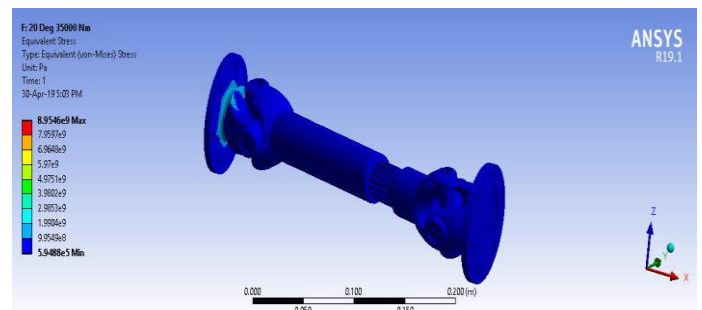


Fig 5 : Equivalent stress obtained on AISI 4140 with a torque of 35000 Nm.

Table-1: Metal properties

Metals	Young's Modulus	Poisson's Ratio
Steel	250 GPa	0.30
AISI 4140	210 Gpa	0.27
Carbon Fiber	228 GPa	0.30
SAE 1130	190 GPa	0.29

5. RESULT AND DISCUSSION

The total deformation and equivalent stress were analysed on different working angles of the cardan shaft ranging from 5 degrees to 40 degrees with an increment of 5 degrees. The analysis was also performed by using different materials of

the cardan shaft namely steel, carbon fiber, AISI 4140, SAE 1130. The analysis was conducted in ANSYS 19.1 (workbench).

5.1 Total Deformation

ANGLE(in degrees)	STEEL	AISI 4140	CARBON	SAE 1130
5	6.919e ⁻⁷	9.728e ⁻⁷	1.0668e ⁻⁵	7.2457e ⁻⁷
10	6.955e ⁻⁷	9.705e ⁻⁷	1.0548e ⁻⁵	7.2762e ⁻⁷
15	6.707e ⁻⁷	9.398e ⁻⁷	1.0891e ⁻⁵	7.0159e ⁻⁷
20	6.779e ⁻⁷	9.509e ⁻⁷	1.0965e ⁻⁵	7.0937e ⁻⁷
25	6.767e ⁻⁷	9.486e ⁻⁷	1.0957e ⁻⁵	7.079 e ⁻⁷
30	7.038e ⁻⁷	9.449e ⁻⁷	1.0699e ⁻⁵	7.0381e ⁻⁷
35	6.702e ⁻⁷	9.421e ⁻⁷	1.0966e ⁻⁵	7.0192e ⁻⁷
40	6.745e ⁻⁷	9.475e ⁻⁷	1.1092e ⁻⁵	7.0609e ⁻⁷

Fig 6 : Total deformation with a torque of 450 Nm

Figure 6 shows the total deformation obtained on different working angles namely 5,10,15,20,25,30,35,40 degrees. The comparison is also made among different materials namely steel, carbon fiber, SAE 1130, AISI 4140. The results here shows that carbon fiber material is the one that is least susceptible to the total deformation. Similar comparison is made with 10000 Nm and 35000 Nm torques.

ANGLE(in degrees)	STEEL	AISI 4140	CARBON	SAE 1130
5	0.00075846	0.0010718	0.0079399	0.00079562
10	0.00077271	0.0010922	0.0081137	0.00081063
15	0.00078614	0.0011114	0.008298	0.0008104
20	0.00079923	0.0011301	0.008409	0.00083855
25	0.0008165	0.0011478	0.0085998	0.00085164
30	0.00082258	0.0011635	0.0085998	0.00086316
35	0.00091249	0.0012919	0.009535	0.00095785
40	0.0009285	0.0013151	0.0097418	0.00097479

Fig 7 : Total deformation with a torque of 35000 Nm

Fig 7 also shows the same experiment with an imaginary torque of 35000 Nm. The result is very much similar to that of earlier one. Thus carbon fiber material is least susceptible to the total deformation in all the analysis conducted.

5.2 Equivalent Stress

ANGLE(in degrees)	STEEL	AISI 4140	CARBON	SAE 1130
5	1.585e ⁷	1.583 e ⁷	3.691 e ⁷	1.5851e ⁷
10	1.681e ⁷	1.6912e ⁷	2.865 e ⁷	1.6847e ⁷
15	1.250e ⁷	1.2521e ⁷	1.7438e ⁷	1.251 e ⁷
20	2.131e ⁷	2.1257e ⁷	4.9447e ⁷	2.130 e ⁷
25	3.584e ⁷	3.582 e ⁷	1.179 e ⁸	3.584 e ⁷
30	3.618e ⁷	3.602 e ⁷	1.473 e ⁸	3.6181e ⁷
35	1.441e ⁷	1.442 e ⁷	5.022 e ⁷	1.4414e ⁷
40	1.210e ⁷	1.2208e ⁷	1.5596e ⁷	1.2177e ⁷

Fig 8 : Equivalent stress with a torque of 450 Nm

Figure 8 shows the equivalent stress obtained on different working angles namely 5,10,15,20,25,30,35,40 degrees. The comparison is also made among different materials namely steel, carbon fibre, SAE 1130, AISI 4140. The results here shows that carbon fibre material is the one that is least susceptible to the equivalent stress. Similar comparison is made with 10000 Nm and 35000 Nm torques.

ANGLE(in degrees)	STEEL	AISI 4140	CARBON	SAE 1130
5	8.583 e ⁹	8.8801e ⁹	1.3176e ¹⁰	8.7359 e ⁹
10	8.639 e ⁹	8.8616e ⁹	1.3289e ¹⁰	8.7172 e ⁹
15	8.7688e ⁹	8.982 e ⁹	1.6127e ¹⁰	8.0891 e ⁹
20	8.7456e ⁹	8.946 e ⁹	1.5613e ¹⁰	8.8188 e ⁹
25	8.6916e ⁹	8.895 e ⁹	1.4963e ¹⁰	8.762 e ⁹
30	8.642 e ⁹	8.8368e ⁹	1.4168e ¹⁰	8.7102 e ⁹
35	8.258 e ⁹	8.846 e ⁹	1.342 e ¹⁰	8.7235 e ⁹
40	8.407 e ⁹	8.5879e ⁹	1.2048e ¹⁰	8.4696 e ⁹

Fig 9 : Equivalent stress with a torque of 35000 Nm

Fig 9 also shows the same experiment with an imaginary torque of 35000 Nm. The results very much similar to that of earlier one. Thus carbon fiber material is least susceptible to the equivalent stress in all the analysis conducted.

6. CONCLUSIONS

The cardan shaft of Bharath Benz DE210BS4 engine is considered for the study. The shaft is allowed to perform its operation in almost all the angles ranging from 5 degrees to 40 degrees with an increment of 5 degrees. The making materials of the cardan shaft was also changed and analysed in ANSYS 19.1(workbench). Different materials that were

analysed are steel, carbon fiber, AISI 4140, SAE 1130. From the analysis, the following conclusions can be made:

International Journal of Mechanical Engg. and Robotics Research, vol.3, 135-152.

[1] While comparing the total deformation in almost all the working angles, it was found that carbon fiber material has the least total deformation. Thus it can be concluded that this material can be considered for the further development of the cardan shaft in future purposes.

[2] While comparing the equivalent stress in almost all the working angles, it was found that carbon fiber material has the least value of equivalent stress. Thus it can be concluded that this material can be considered for the further development of the cardan shaft in future purposes.

7. ACKNOWLEDGEMENT

We are thankful to all the teaching and non-teaching staff of Sri Vellappally Natesan College of Engineering for supporting us throughout this work.

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