# Finite Element Analysis of Universal Joint and Propeller Shaft Assembly

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*Abstract—* **The power transmission system is the system which causes movement of vehicles by transferring the torque produced by the engines to the wheels after some modifications. The transfer and modification system of vehicles is called as power transmission system. The power transmission system of vehicles consists of several components which encounter unfortunate failures. These failures may be attributed to material faults, material processing faults, manufacturing and design faults, etc. Maintenance faults and user originated faults may also responsible. Propeller shaft and the universal joints form the important links that help in transmitting power from the engine to the wheels. In this study, analysis is being performed on the universal joint yoke and the propeller shaft. In the universal joint yoke, certain modifications are made in the existing geometry and analysed for the identical loading and boundary conditions as in the reference paper from which the problem has been taken. In case of propeller shaft a comparative study has been made between two shafts differing in their material, keeping in view the possible weight reduction that can be obtained without affecting the functionality of the shaft. Both the components are analysed in ANSYS and the results are compared.**

## I. INTRODUCTION

A universal joint or Hooke's joint is a joint or coupling in a rigid rod that allows the rod to bend in any direction, [3] and is commonly used in shafts that transmit rotary motion. It generally consists of two hinges located close together, oriented at 900 to each other, connected by a cross shaft. It is widely used in industrial applications and vehicle drivelines to connect misaligned shafts. [4]. A major problem with the use of a Hooke's joint is that it transforms a constant input speed to a periodically fluctuating one. The kinematical consequences of this property of this joint can be remedied, as long as rigid body rotations are concerned, by using two converse Hooke's joint. But if torsional vibrations of the propeller shaft are concerned, there is no way of removing the dynamical consequences of an introduced Hooke's joint in a rear wheel drive vehicle. In a widely used

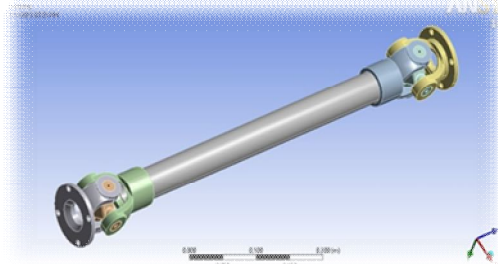single piece drive shaft, two universal joints are used [2].



Fig 1: Universal joint and drive shaft assembly

Two universal joints are preferred in order to avoid the transformation of constant input speed into a fluctuating speed which is encountered when a single universal joint is used. Now a day's more emphasis is being given on reducing the weight of the drive shaft. It is being tried to replace the existing Steel shafts with Aluminium alloy shafts as it has a higher strength to weight ratio. Attempts are also being made to replace conventional steel shafts with hybrid aluminium/composite drive shaft.

## II. METHODOLOGY

(YOKE) The problem for this analysis was taken from the paper [1]. As can be seen from the paper the model (Universal joint yoke) was analysed in ANSYS considering the component to be made up of AISI5046H, which is a material in the low alloy steel group. The typical mechanical properties are 1750Mpa as ultimate tensile strength and 1400Mpa as the yield strength. A force applied at the spider mounting location is a torsional moment of 200Nm. In addition the given torsional moment, a 500rmp speed was also considered. When AISI5046H material is considered, standard fatigue limit estimation process from the tensile stress (Half of tensile strength with size correction factor, loading correction factor) gives a value between 400 and

450 Mpa. As per the ANSYS analysis, the highest stress that occurred in the yoke was found to be 356 Mpa.

For this study the model was replicated from the above mentioned paper. The material as well as the loading conditions of torsional moment and the rotational speed is kept the same. The model was analysed in ANSYS14. However there were some changes made in the geometry of the existing model.

### III. MODIFICATIONS

The model of yoke that is analysed in the reference paper has its mounting base a plain surface with a central hole. A chamfer was provided on the inner hole at the base of the yoke where it is mounted to the power transmitting end of the gearbox.

The original model has the region joining the main body of the yoke with the base as a sharp edge structure as shown in the Fig 4.

In the modified model, a fillet of radius 5.5mm is also provided in addition to the chamfer. The model with added fillet is shown in the figure below.

The component was modelled in SolidWorks. Rest of the geometry is kept unaltered. The purpose of this analysis is to study the effects due to changed geometry on the behaviour of the component under identical loading and boundary conditions as mentioned in the above reference paper. The results obtained are discussed below
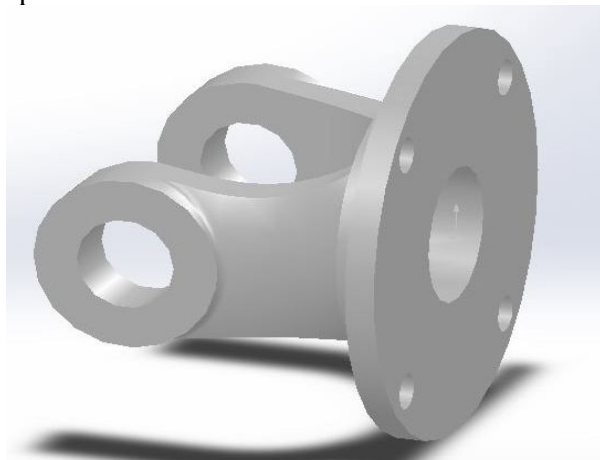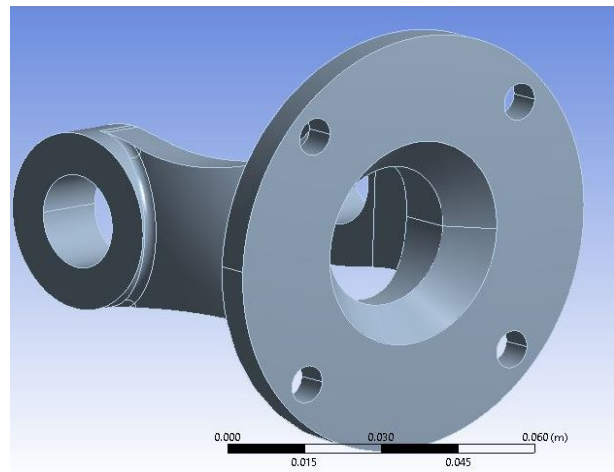


Fig 3: Yoke with chamfer at the base



Fig 4: Sharp edge type base region



Fig 2: Yoke without Chamfer at the base
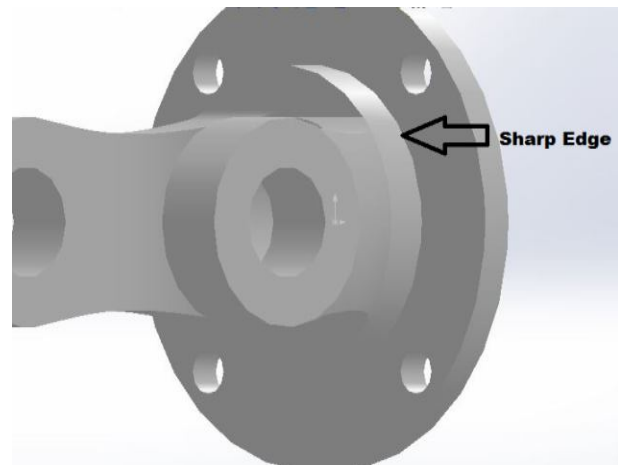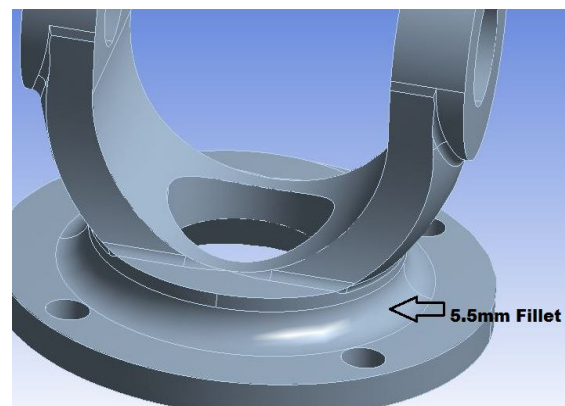


Fig 5: Model with a 5.5 mm fillet at the base

### IV. UNIVERSAL JOINT YOKE ANALYSIS

Due to the introduction of the chamfer in the geometry, the weight of the component reduces, but we have also introduced a fillet, resulting in material addition. But, the resultant effect is reduction in the overall material content of the component.

The ANSYS analysis shows that the maximum stress value obtained is 242.94 Mpa, which is within the safety limits as prescribed in the paper. Moreover, this value is also less than the maximum stress value in the original component. Thus we can conclude that the new modelled component is well within the safe limit.
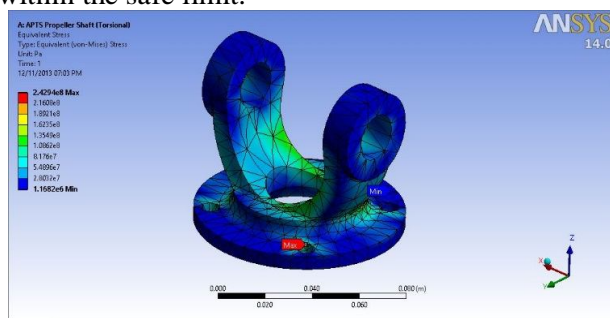

Fig 7: ANSYS analysis of yoke

Figure below shows the analysis of the universal joint yoke in ANSYS. Analysis was carried out for identical loading conditions as in the reference paper.

In design there are two main factors considered, the first being the maximum stress encountered in the component and the other being the area affected by this maximum stress. Due to introduction of the fillet the maximum stress encountered is reduced. The point of maximum stress also shifts and the region affected by the maximum stress is significantly reduced. The maximum stress affected zone reduces to a point as shown in the fig 6.The stresses in the component even out leading to reasonably uniform stress distribution. Thus the modified geometry results in a component having lesser weight and at the same time is able to respond to the applied loading conditions.
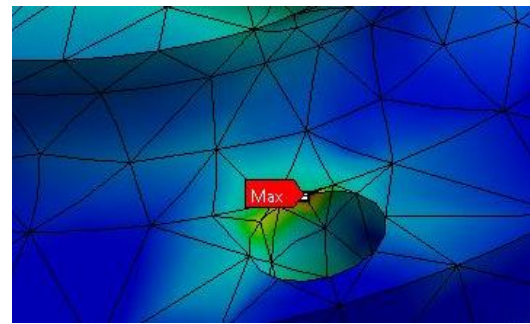

Fig 6: Maximum stressed zone reduces to a point

### V. PROPELLER SHAFT ANALYSIS

The propeller shaft was designed taking into consideration the torque to be transmitted by it. The material utilized for the shaft as per the reference paper is AISI 94B30H (Material- 2) which is a alloy steel (oil quenched and tempered) with Boron as an alloying element. The tensile strength and the yield strength values are 1700 Mpa and 1550 Mpa respectively.

Considering the given loading conditions, the max stress value in the component when calculated analytically was found to be 24.14 Mpa using Von mises stress theory.
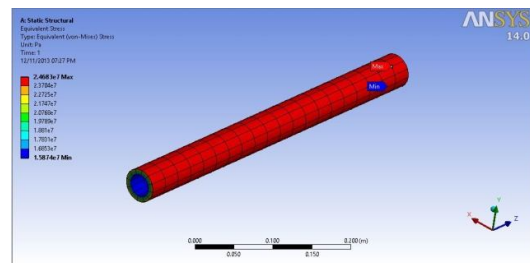

Fig 7: Stress analysis of propeller shaft (Steel)

The propeller shaft model when analysed in ANSYS using identical loading conditions as in case of the yoke, the maximum stress encountered (at the surface) using AISI 94B30H is 24.68Mpa which is approximately equal to the value calculated analytically. As far as the material strength is concerned, the maximum allowable stress in the component is found to be 428 Mpa. This allowable stress value is obtained as per the standard fatigue limit estimation process from the tensile stress (half of tensile strength with size correction factor). Thus we can conclude that the

shaft of material-2 can perform satisfactorily under the given loading conditions.
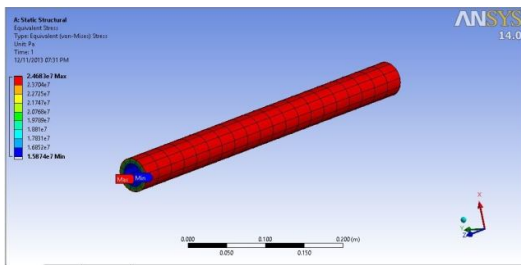

Fig 8: Stress analysis of propeller shaft (Aluminium)

However, the above mentioned class of steel has a density of 7820 Kg/m$^3$. For the design dimensions utilized in the analysis, the mass of the shaft obtained is about 3.45 Kg. Hence, emphasis is given on reducing the weight of the propeller shaft, without affecting the dimensions of the component.
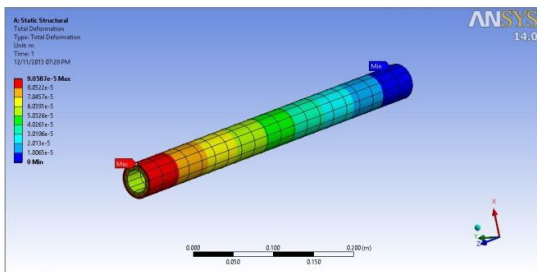

Fig 9: Displacement of propeller shaft (Steel)

In order to reduce weight, another material Al-6061-T6 alloy (Material-3) is suggested in place of material-2. The tensile strength and the yield strength values are 310 Mpa and 276 Mpa respectively. The above Aluminium alloy has a density of 2700 Kg/m$^3$.
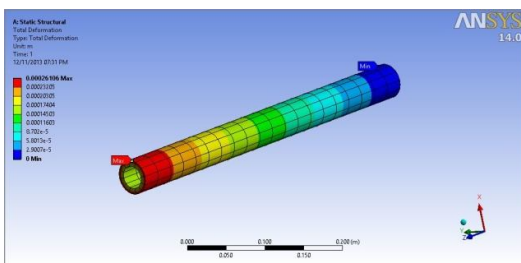

Fig: Displacement of propeller shaft (Aluminium)

When the propeller shaft made up of material-3 was analysed in ANSYS, It yielded a maximum stress value of 24.69 Mpa. This stress value is approximately same as that in the previous case utilizing material-2 (maximum allowable stress in the Al-6061-T6 alloy is 39.5 Mpa as per the analytical calculations). However the weight of the new shaft is only 1.2 Kg i.e. significantly less when compared to material-2. Hence, we are able to significantly reduce the weight of the propeller shaft without making any changes in the original dimensions.

## VI. CONCLUSIONS

- Analysis of the Yoke clearly shows that by a small modification in the existing design the strength of the part can be increased significantly.
- Also with the same changes we obtain a small amount of weight reduction in the design.
- The maximum stress values generated are significantly reduced and the stress is evenly distributed over the entire part.
- Propeller shaft can be made either from steel or from the aluminium. The stress generated are approximately the same but the deformation in aluminium shafts is higher than that of the steel shaft.

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