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Design and analysis of Three Pin Constant Velocity Joint for Parallel and Angular Power Transmission

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ABSTRACT

The basic role of couplings is to join two parts of rotating elements while permitting some degree of misalignment or end movement or both. By cautious selection, establishment and support of couplings, generous reserve funds can be made in decreased maintenance cost and downtime. Presently Oldham's coupling and Universal joints are used for parallel offset power transmission and angular offset transmission. These joints have limitations on maximum offset distance, angle & speed and result in vibrations and low efficiency (below 70%). The three pin constant velocity joint is an alteration in design that offers upto 18 mm parallel offset and 21 degree angular offset, at high speeds upto 2000 or 2500 rpm at 90% efficiency. This design lowers cost of production, space requirement and simply technology of manufacture as compared to present CVJ in market.

Keywords :- 3-pin Constant Velocity Joint, Parallel Offset, Angular Offset, Power Transmission, Von-Mises Stress

I. INTRODUCTION

In any direct mechanical drive system, there exists a need to couple the variety of driven elements that may be included. The majority of drive elements, including gear reducers, lead screws, and a host of other components, are driven by shafting that is supported by multiple bearings. This allows for shafting to be held extremely straight and rigid while rotating, avoiding any possible balancing and support problems. Because of this rigid support, it is virtually impossible to avoid slight misalignments between a driving and driven shaft when they are connected. Restoring forces that occur as the two coupled shafts compete to maintain their original positions can put unwanted strain on shaft bearings, causing them to wear out prematurely. Additional axial loads are also placed on the bearings as thermal growth occurs in shafting during operation.

The essential capacity of a force transmission coupling is to transmit torque from an info shaft to a yield shaft at a given shaft speed and, where important, to accomodate shaft misalignment. Misalignment is the after effect of numerous components including installation erors and tolerence variation. Shaft misalignment can expand the axial and radial forces applied on the coupling. In misaligned applications, undesirable side loads are usually introduced by the coupling. These side loads which are resulting from dynamic behavior of coupling, frictional loads and loads caused by flexing or compressing coupling components. The undesirable results include:

1. Torsional or angular velocity vibrations which diminish system accuracy.
2. Excessive forces and warmth on system bearings which diminish machine life.

Expanded system vibration and commotion which unfavorably influences equipment operation

Tae-Wan Ku, et.al [1] given that the external race of CV (steady speed) joints with six internal ball grooves has been con-ventionally created by the multi-arrange warm producing forms, which includes a few operations including forward expulsion, irritating, in reverse expulsions, estimating and necking, and also extra machining. There is still no decision yet to create the complex formed parts other than by this warm forging process. As an option, multi-stage cold forging process is exhibited to supplant these customary warm forging. It is demonstrated that the multi-stage chilly fashioning process in this study could be effectively connected to the large scale manufacturing of the external race of the

CV joints with the considerably decreased processing time for the machining procedure on the inward ball groove.

Chul-Hee Lee, et.al [2] In this paper, a phenomenological CV joint friction model was created to display the contact conduct of tripod CV joints by utilizing an instrumented CV joint grinding mechanical assembly with tripod-sort joint gatherings. Examinations were directed under various working states of oscillatory speeds, CV joint articulation angles, oil, and torque. The trial information and physical parameters were utilized to build up a material science based phenomenological CV joint element grinding model. It was found that the proposed friction model catches the trial information well, and the model was utilized to anticipate the external produced axial force, which is the fundamental main source of power that causes vehicle vibration issues.

K.S. Park, et.al [3] In the present study on forged outrace CV joints, a ball groove estimation system that uses the mechanical linear displacement sensor and lab view programming has been created and executed in a modern production line. The ball groove estimation system was composed particularly to gauge the measurements of the six ball grooves in the external race. The recently created system provides high measurement accuracy with a simple operational sequence.

Katsumi Watanabe, et.al [4] shown the Rzeppa constant velocity joint is composed of several sets of the ball and two circular-arc grooves. Relative motions of the ball to two circular-arc grooves is analyzed and the output angle error in a practical use which contains sinusoidal fluctuations with periods 2π, $2\pi/3$, $2\pi/6$ is simulated by the circular-arc-bar constant velocity joint.

Nishant Ramesh Wasatkar [5] Misalignment in shaft may results into undesirable strains on shaft orientation bringing. Couple of ordinary arrangements are available for misalignment issues like Oldham's coupling and widespread joint which have a few confinements. These limitations can be overcome with Thompson constant velocity (CV) coupling which offers highlights like minimizing side loads, higher misalignment capabilities, more operating speeds, improved efficiency of transmission and many more. This paper presents review on constant velocity joints/couplings configuration and advancement. In this paper the examination work of different researchers identified with transmission couplings and constant velocity joints is reviewed.

II. METHODS AND MATERIAL

The solution to the above problem is an indigenous coupling that gives constant transmission of torque and angular velocity. The main features of the coupling being;

1. Minimize or even eliminate side loads
2. Higher shaft misalignment capabilities
3. Greater drive accuracy.

Schematic showing the arrangement of test rig in three condition of testing namely:

a) Zero offset condition
b) Parallel offset condition
c) Angular offset condition

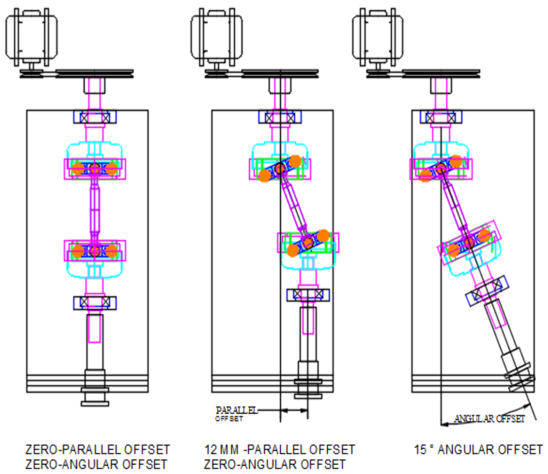

Figure 1. Arrangement of Test Rig

Work will be carried out in the following steps.

1. Design Of Input Shaft:

Analytical Approach:
Material: EN24
Ultimate Tensile Strengh: 800N/mm^2
Yield Strengh: 680N/mm^2
fs_{max} = Uts/fos = 800/2 = 400 N/mm^2
Check for torsional shear failure of shaft

$$Te = \frac{\pi}{16} fs\, d^3$$

$$fs_{act} = \frac{16\times0.25\times10^3}{\pi\times16^3}$$

fs_{act} = 0.310 N/mm^2

As; $fs_{act} < \; fs_{all}$

Input Shaft is safe under Torsional load.

Finite Element Analysis:

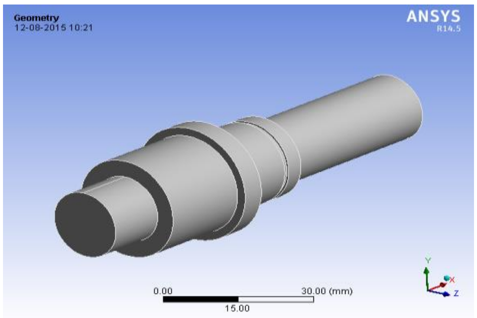

Figure 2. Modeling of Input Shaft

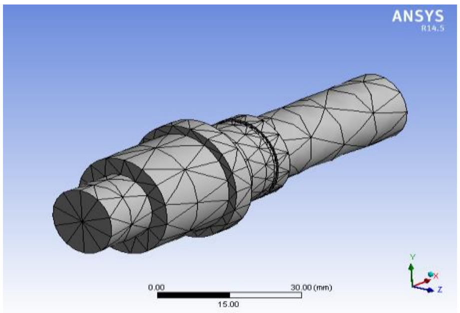

Figure 3. Meshing of Input Shaft

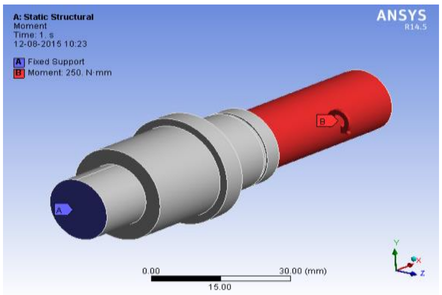

Figure 4. Static Strucure of Input Shaft with Fixed Support & Moment

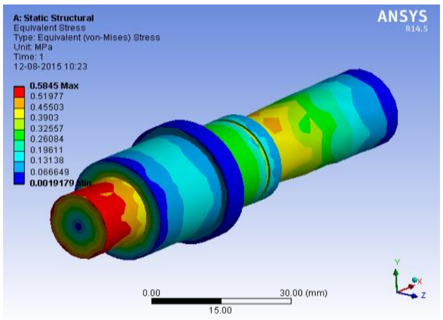

Figure 5. Von-Mises stress of Input Shaft

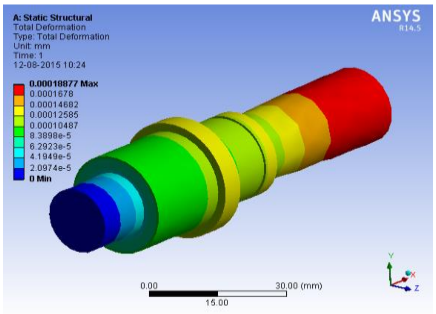

Figure 6. Total Deformation of Input shaft

2. Design of Input Coupler Body

Analytical Approach:

Material: EN24

Ultimate Tensile Strengh: 400N/mm^2

Yield Strengh: 280N/mm^2

fs_{max} = UTS/FOS =400/2 = 200N/mm^2

Check for torsional shear failure:-

$$T = \frac{\Pi \text{ x } fs_{act}}{16} \text{ x} \left(\frac{Do^4 - Di^4}{Do} \right)$$

$$0.25 \text{ x } 10^3 = \frac{\Pi \text{ x } fs_{act}}{16} \text{ x} \left(\frac{22.5^4 - 16^4}{22.5} \right)$$

fs_{act} = 0.15N/mm^2

As; $fs_{act} < fs_{all}$

Input coupler body is safe under torsional load

Finite Element Analysis:

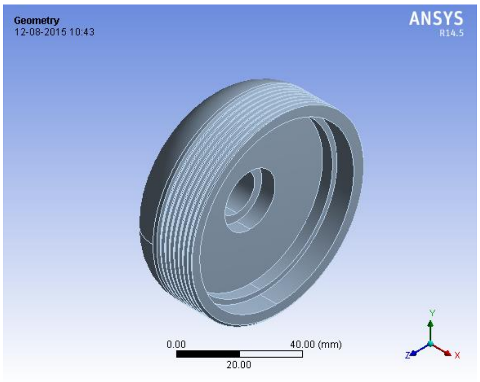

Figure 7. Modeling of Input Coupler Body

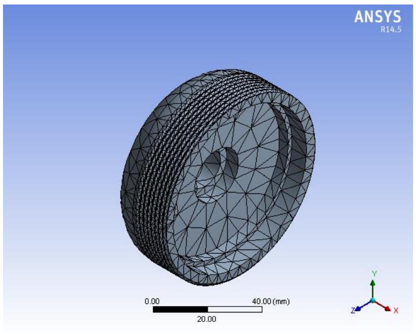

Figure 8. Meshing of Input Coupler Body

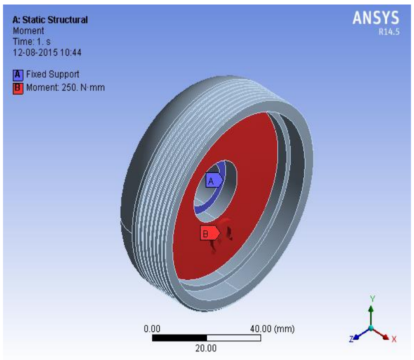

Figure 9. Static Structure of Input Coupler Body with Fixed Support & Moment

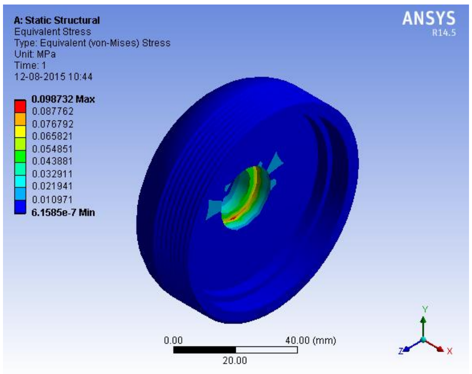

Figure 10. Von-mises stress of Input Coupler Body

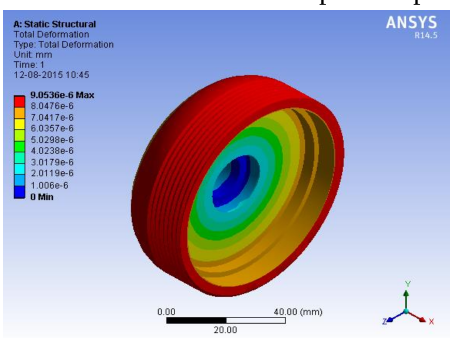

Figure 11. Total Deformation of Input Coupler Body

3. Design of Input Coupler Ring

Analytical Approach:
Material: EN24
Ultimate Tensile Strengh: $800 N/mm^2$
Yield Strengh: $680 N/mm^2$
fs_{max} = UTS/FOS =800/2 $=400 N/mm^2$

Check for torsional shear failure:-

$$T = \frac{\Pi \times fs_{act}}{16} \times \left(\frac{Do^4 - Di^4}{Do}\right)$$

$$0.25 \times 10^3 = \frac{\Pi \times fs_{act}}{16} \times \left(\frac{88^4 - 73^4}{88}\right)$$

$fs_{act} = 0.0035/mm^2$
As; $fs_{act} < fs_{all}$

Input coupler ring is safe under torsional load

Finite Element Analysis:

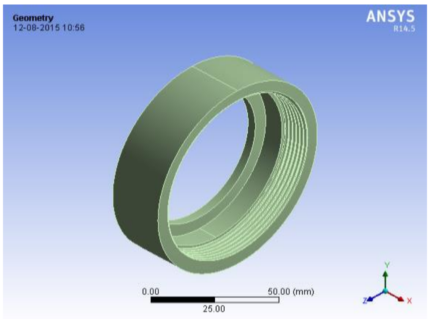

Figure 12. Modeling of Input Coupler Ring

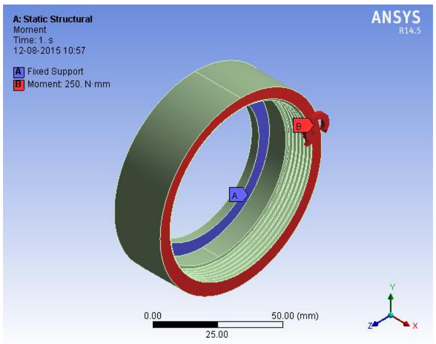

Figure 13. Static Structure of Input Coupler Ring with Fixed Support & Moment

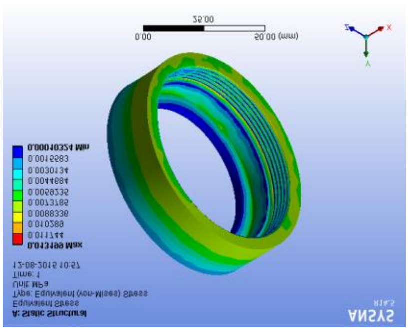

Figure 14. Von-Mises stress of Input Coupler Ring

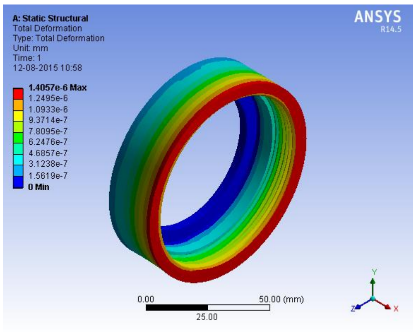

Figure 15. Total Deformation of Input Coupler Ring

4. Design Of Input Coupler Female Liner

Analytical Approach:

Material: EN24

Ultimate Tensile Strengh: 800N/mm^2

Yield Strengh: 680N/mm^2

fs_{max} = UTS/FOS =800/2 = 400N/mm^2

Check for torsional shear failure:-

$$T = \frac{\Pi}{16} \times fs_{act} \times \left(\frac{Do^4 - Di^4}{Do}\right)$$

$$0.25 \times 10^3 = \frac{\Pi}{16} \times fs_{act} \times \left(\frac{65^4 - 57^4}{65}\right)$$

$\Rightarrow fs_{act} = 0.0113\text{N/mm}^2$

As; $fs_{act} < fs_{all}$

Input Coupler Female Liner is safe under torsional load

Finite Element Analysis:

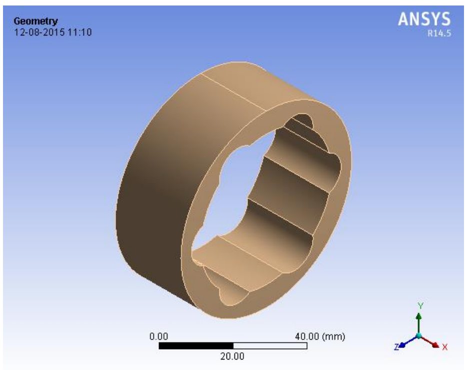

Figure 16. Geometry of Input Coupler Female Liner

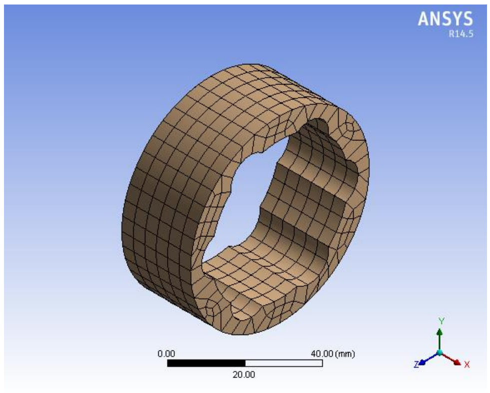

Figure 17. Meshing of Input Coupler Female Liner

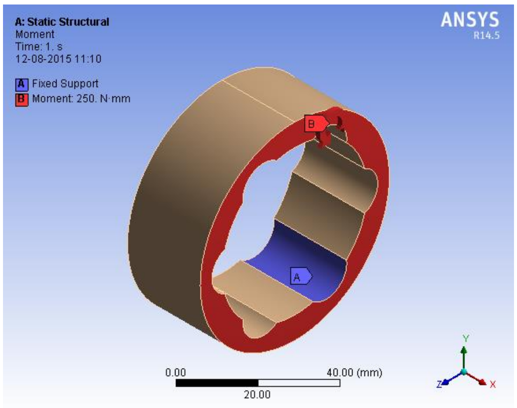

Figure 18. Static Stricture of Input Coupler Female Liner with Fixed Support & Moment

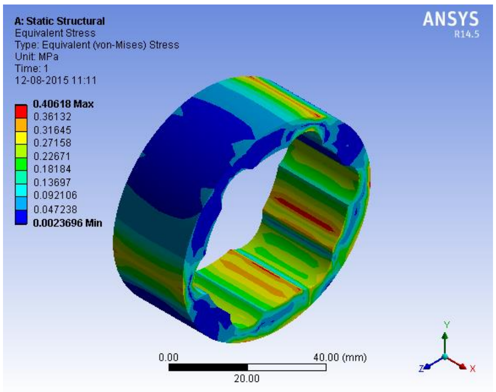

Figure 19. Von-Mises stress of Input Coupler Female Liner

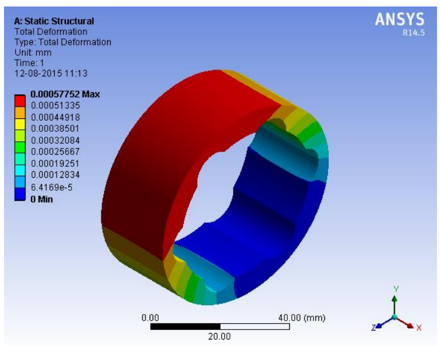

Figure 20. Total Deformation of Input Coupler Female Liner

5. Design of Coupler Pin

Analytical Approach:
Material: EN24
Ultimate Tensile Strengh: 800N/mm^2
Yield Strengh: 680N/mm^2
fs_{max} = uts/fos = 800/2 = 400 N/mm^2
This is the allowable value of shear stress that can be induced in the shaft material for safe operation.
Check for torsional shear failure of shaft

$$Te = \frac{\pi}{16} fs\ d^3$$

$$fs_{act} = \frac{16 \times 0.25 \times 10^3}{\pi \times 8^3}$$

fs_{act} = 2.4860 N/mm^2
As; $fs_{act} < fs_{all}$
Coupler pin is safe under torsional load.

Finite Element Analysis:

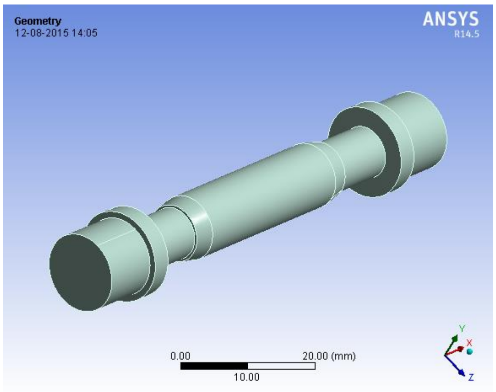

Figure 21. Modeling of Coupler Pin

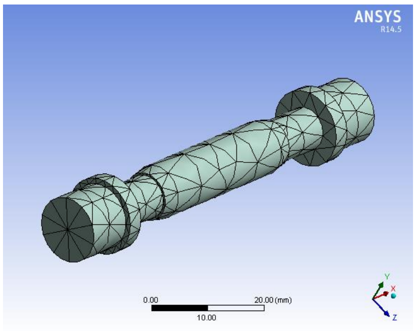

Figure 22. Meshing of Coupler Pin

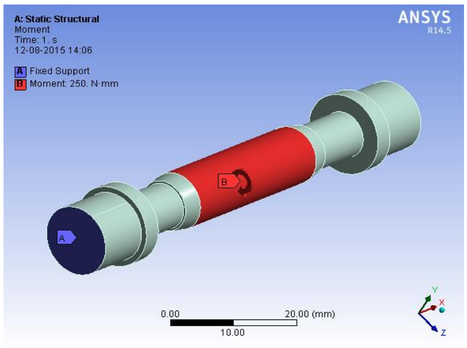

Figure 23. Static Structure of Coupler Pin with Fixed Support & Moment

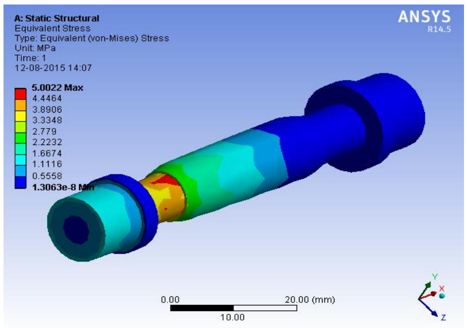

Figure 24. Von-Mises stress of Coupler Pin

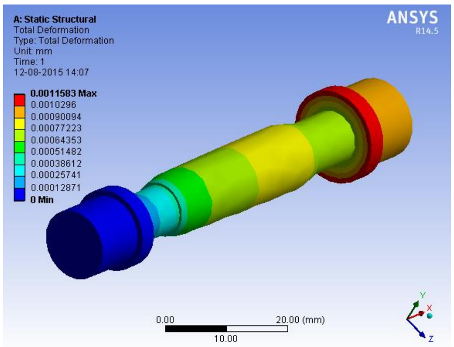

Figure 25. Total Deformation of Coupler Pin

6. Design Of Trunion Holder

Analytical Approach:
Material: EN24
Ultimate Tensile Strengh: 400N/mm^2
Yield Strengh: 280N/mm^2
fs_{max} = UTS/FOS =400/2 = 200N/mm^2

Check for torsional shear failure:

$$T = \frac{\Pi \times fs_{act} \times}{16} \left(\frac{Do^4 - Di^4}{Do} \right)$$

$$0.25 \times 10^3 = \frac{\Pi \times fs_{act} \times}{16} \left(\frac{36.4^4 - 23^4}{36.4} \right)$$

fs_{act} = 0.2 N/mm^2
As; $fs_{act} < fs_{all}$

Trunion holder is safe under torsional load

Finite Element Analysis:

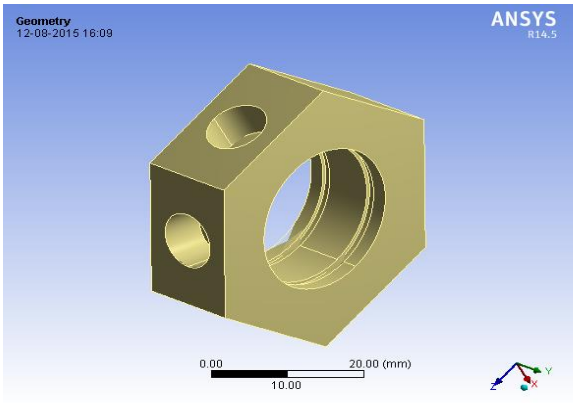

Figure 26. Modeling of Trunion Holder

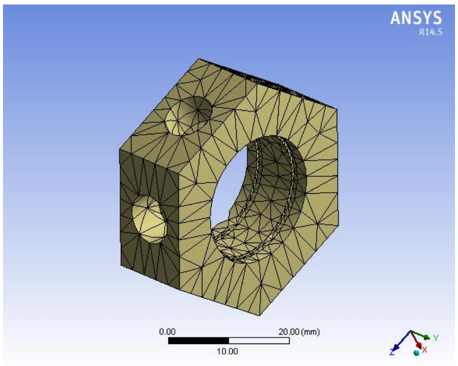

Figure 27. Meshing of Trunion Holder

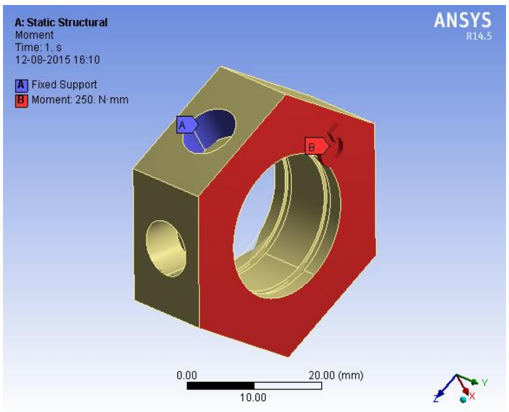

Figure 28. Static Structure of Trunion Holder with Fixed Support & Moment

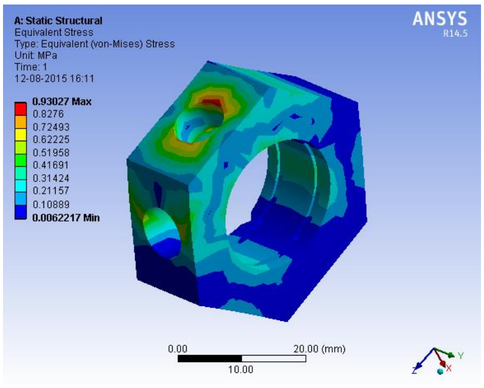

Figure 29. Von-Mises Stress of Trunion Holder

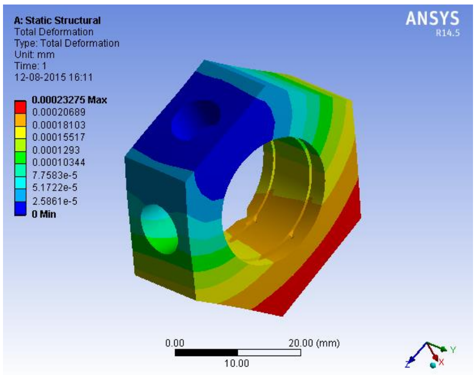

Figure 30. Total Deformation of Trunion Holder

III. RESULTS AND DISCUSSION

The Finite element analysis results of all other parts are given below

Table 1. Von Mises Stress and Total Deformation of all Design Parts

Part name	Max. shear stress N/mm^2	Actual Theoretical stress N/mm^2	Von-mises stress N/mm^2	Total deformation mm
Input Shaft	400	0.310	0.5845	0.0001887
Input Coupler Body	200	0.15	0.098	9.06E-6
Input Coupler Ring	400	0.0035	0.013	1.045E-6
Input Coupler Female Liner	400	0.0113	0.40	1.045E-6
Coupler Pin	400	2.486	5.02	0.0011
Trunion Holder	200	0.2	0.9	0.00023

IV. CONCLUSION

Theoretical Actual stress and Von-mises stress of all parts are well below the allowable limit; hence all the parts are safe. Also the value of Total deformation of all parts is very small so the deformation is neglected. So, the three pin constant velocity joint is designed and analysed successfully.

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