Design of Engine Gimbaling Mechanism for Thrust Vector Control of Liquid Propellent Rocket Engine

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Abstract*— *The purpose of this research is to design a gimbaling mechanism for thrust vector control of a space launch vehicle's first stage liquid propellent rocket engine. A complete framework has been developed where the basic propulsion design was carried out using excel, for 400 KN of thrust. The parameters obtained were used for a parametric design of a knuckle joint and support structure. Using stainless steel, the net assembly was modeled using CATIA V5 and analyzed using ANSYS Workbench. The outcomes were checked for failure using Tresca's criterion and it has been determined that the proposed design for Austenitic Stainless Steel 304L can be used for gimbaling the Liquid Propellant Rocket Engine and the results of analysis are presented.

Keywords*—*Gimbaling; TVC; LPRE; failure theories; knuckle joint, stainless steel, nozzle, clustered engines

I. INTRODUCTION

Thrust Vector Control (TVC) is a method of attitude control for rocket propelled vehicles. In Space and Upper atmosphere where air density is extremely low, attitude control mechanisms such as using control surfaces are impractical and TVC is the only feasible way. However, even below upper atmosphere many long range missiles use TVC as their foremost method of attitude control because of their large size.

Over the years several methods have been developed to accomplish thrust vector control. Jet Vanes, Secondary Fluid Injection, Gimbals or hinges and small Thrusters are a few prominent methods of attaining TVC [1].All methods have their benefits and drawbacks and naturally for a certain mission they are chosen according to the requirements and details of the mission. For reasons mentioned above an appropriate TVC mechanism is to be chosen for the Booster Stage of a Space Launch Vehicle (SLV) with Liquid Propellant Rocket Engines (LPRE). Since thrust is very high at this stage Engine Gimbaling is chosen as the method for attitude control [2, s. 102-160]. It provides an edge over other mechanisms by its location; being placed atop the engine it is away and protected from all hot gases beyond the thrust chamber [1, s.120]. Furthermore gimbaling the whole engine instead of just the nozzle, provides a higher rotation range to the nozzle and hence a higher control authority i.e. ±7º. [1, s.217-281]

Typical gimbal-mounted and hinge-mounted LPREs used in the past for vehicles such as Viking, Atlas missiles, Titan missiles and Rocketdyne SSME, have adopted various gimbals such as spherical bearings, universal joints and multiple hinges [3, s.220].

For an LPRE providing 2000 KN of thrust, the gimbal mechanism proposed in this paper is a knuckle joint because of the simplicity associated with its design and its control. Also, engine gimballing range is small which can be achieved by knuckle joint [4][5]. The model is designed and analyzed iteratively to obtain finally an optimized assembly that will successfully provide the required TVC by an LPRE without failing.

II. TVC CONCEPT

The goal of the mechanism is to obtain full thrust vector control, that is, yaw, pitch and roll. Achieving this by a single knuckle joint is impossible as a knuckle joint provides only one degree of freedom.

A. Clustered Engines

Clustered Engines have been used well enough times in the past; notable examples of vehicles using them are the Saturn V and the Space Shuttle. Not only are they used as an alternative to an extremely large engine, they are easy to manufacture, provide incremented thrust and assist in TVC as well [3, s.35-100]. For example, to provide 3 degrees of freedom by our chosen mechanism, the following clustered engines arrangement is used: 5 clustered engines with one fixed in the center, two gimbaled along X axis and the remaining two gimbaled along Y axis as shown in Figure 1. Each engine is providing 400KN of thrust.

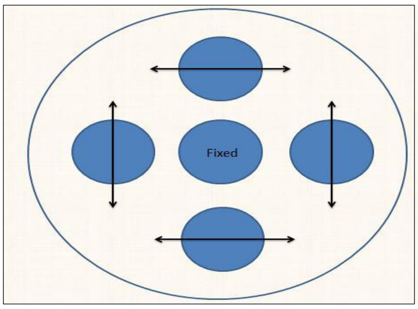

Figure 1: Clustered Engines Arrangement

III. RESEARCH METHODOLOGY

Our aim was to design a mechanism that can withstand the thrust applied not only at an upright position but also at a tilted position. Method for designing is shown in figure 2.

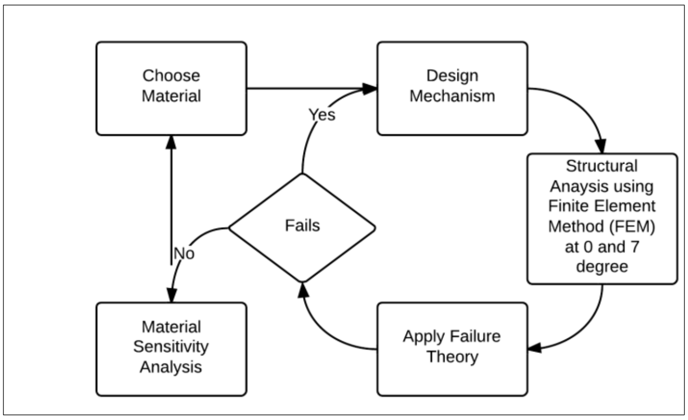

Figure 2: Research Methodology

A. Material Properties

Dimensions of the knuckle joint are dependent on the material used so to start somewhere we have selected Austenitic Steel 304L. Since Austenitic Steel's Strength-to-Weight ratio is lower than most materials, later in the design process it will be changed after performing sensitivity analysis. The essential material properties are recorded in Table 1 [6].

Table 1: Material Properties

Austenitic Steel 304L	
Compressive Strength	210.0 MPa
Yield Strength	215.0 MPa
Yield Shear Strength	124.7 MPa

B. Design

The Knuckle joint was designed for the compressive loads of 400KN using a FOS of '1' [4]. Figure 2 shows the parametric dimensions of knuckle joint where'd' is calculated by

$$\text{Compressive Strength of Material} = \text{Force} / \pi \times d^2$$

$$r = d/2$$

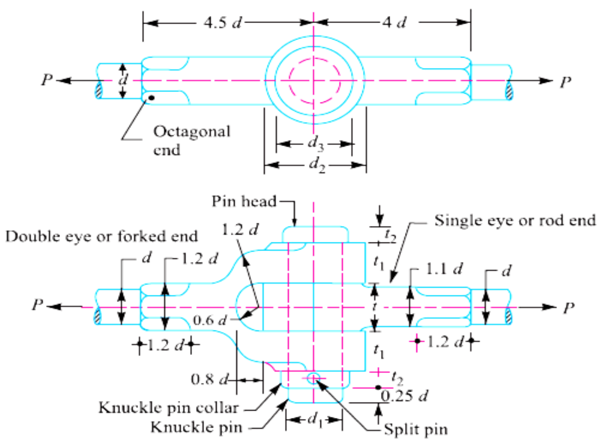

Figure 3: Knuckle Joint Dimension

However, after analyzing the joint on ANSYS, and a historical literature review we found that to design a knuckle joint

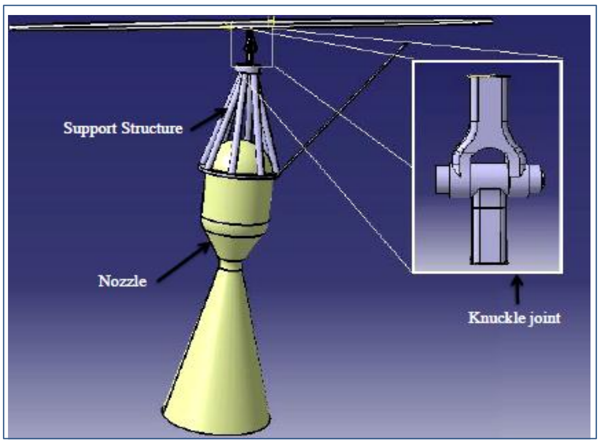

Figure 4: Complete Gimbal Assembly

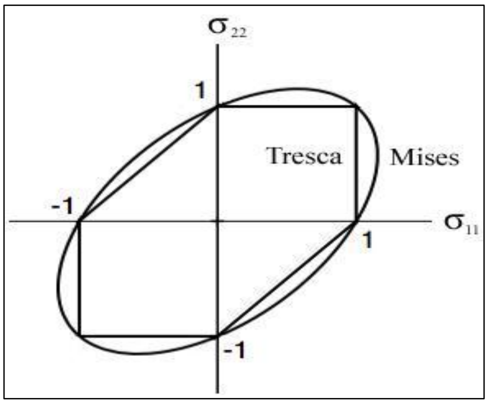

Figure 5: Validity of Tresca's Criterion

properly, we must analyze the knuckle joint combined with the engine's support structure and actuator so as to 'guide' it [3][11]. The support structure was designed based on the dimensions of thrust chamber, which were calculated using reference [1], for 400KN. Figure 4 shows the complete assembly.

C. Failure Criteria

ANSYS Workbench does not provide signed equivalent stresses experienced by the model. Therefore instead of using Von Mises' criterion, which compares Equivalent Stress to Tensile Yield Stress or Compressive Yield Stress, Tresca's criterion is used to check if structures fail or not. According to Tresca's criterion, if maximum shear strength at any point exceeds the critical value, that is, yielding shear strength of a material, then the design fails [7] [8]. Moreover, Tresca's criterion is more conservative in a sense that if a material does not fail in Tresca's criterion, it will not fail with Von Mises criterion as shown in Figure 2 [9]. Based on the above mentioned reasons, Maximum Shear Stress at any point on the body should be less than 0.58 times the yield stress [10]. In case of Stainless steel, this comes out to be 124.7 MPa.

D. Analyses Using Finite Element Method

To perform static structural analysis using finite element method ANSYS Workbench was used. Initially, only the knuckle joint assembly was analyzed, both, at zero degree and 7 degree. After that, the whole mechanism was designed together. To avoid failure, many modifications were made in the whole assembly and the design process was done step by step. Some of these modifications are discussed in this paper below:

- Modifications in knuckle joint dimensions
- Change in fillet radius of both, eye and fork end of knuckle joint
- Changes in truss structure
    1. 3 Vs truss structure
        a. Varying Truss top-plate diameter
        b. Varying Truss rod diameters
    2. Straight Inclined rods (not connected as V)
    3. 4 Vs truss structure

All truss structures were analyzed for different diameters of rods and an optimized support structure was chosen. Results for all modifications are shown in section IV. For analysis purposes, the fork-end of knuckle joint was fixed while thrust was applied on the eye or support structure end in form of pressure. Different mesh sizes were checked and around 6mm of mesh-element size was selected for knuckle joint while a default element size was used for the rest of the assembly.

IV. RESULTS & DISCUSSION

Results of knuckle joint analysis without support structure were evaluated and are shown in Figure 8 for 0 degree (upright position) and 7 degrees (full tilted position). At 0º, Maximum Shear Stress is 175.63 MPa while at 7^{o}, Maximum Shear Stress is 766.18 MPa. This increase in shear stresses in tilted position is because of the x-component of thrust that is producing pure shear and hence causing knuckle joint to fail at fork end, on the side opposite to deflection. Knuckle joints cannot be used in compression; hence guides are required [11]. In our case, this guide is the actuator rod attached to the support structure. Furthermore, we know that according to Continuum mechanics the effect of stresses in one part will increase or decrease the stresses in other parts. Based on that, we have observed different trends of Maximum Shear Stress by changing different parameters of assembly as this critical stress value is required to be lower than the maximum strength of material. These trends are explained below.

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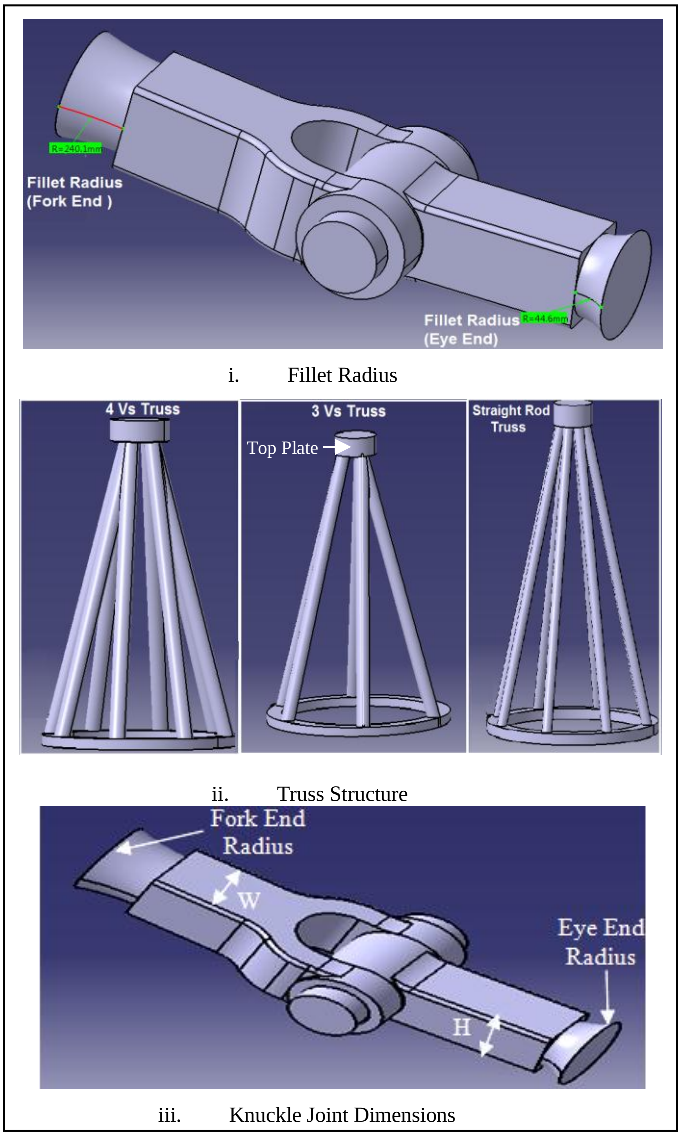

i. Fillet Radius

ii. Truss Structure

iii. Knuckle Joint Dimensions

Figure 6: Major Modification of Assembly

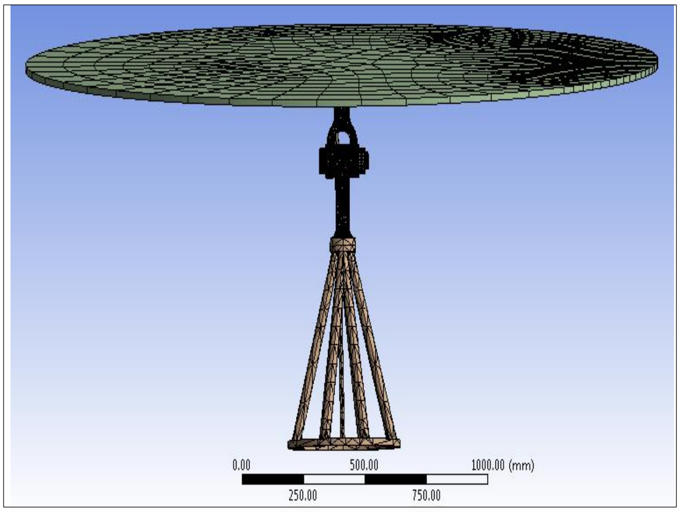

Figure 7: Mesh of Assembly

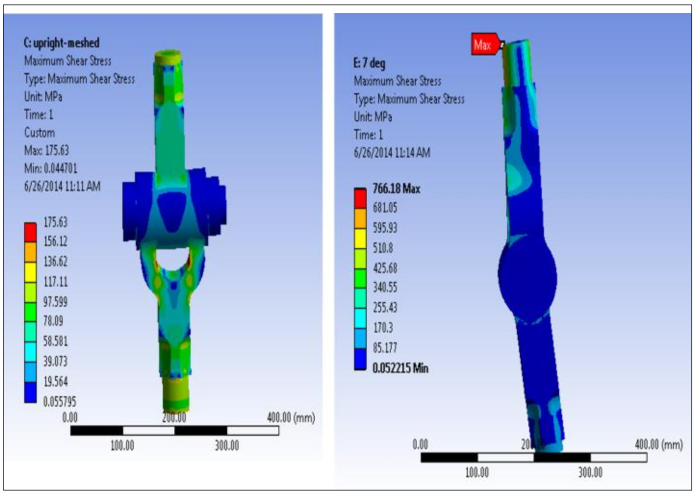

Figure 8: Analysis of Knuckle Joint at 0 and 7 degree

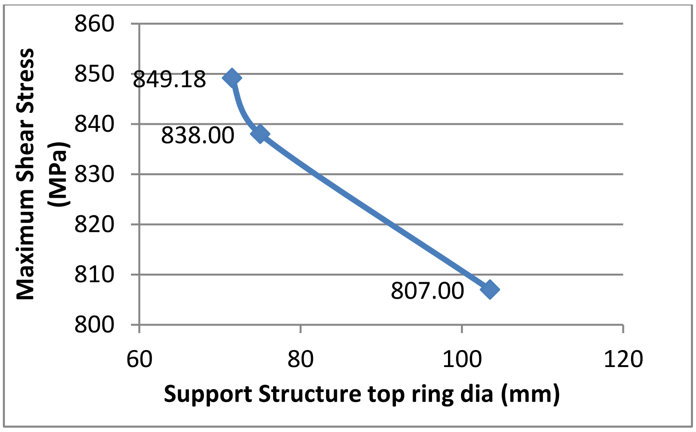

Figure 9: Effect of Increasing Top Ring Diameter of Truss

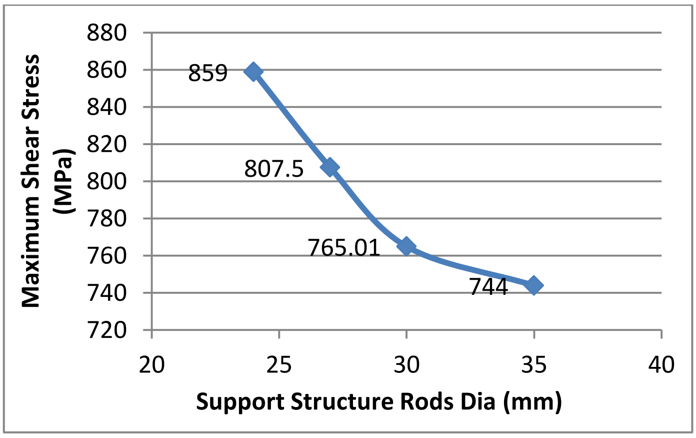

Figure 10: Effect of Changing Rod Diameter for 3Vs Truss

A. Truss Top Ring Diameter

For *3Vs support structure*, having their V-rod's diameter of 27mm, trends for top plate (or top ring) diameter were computed. Stresses were measured using ANSYS and the results are summarized in Figure 9. Eye-end's edge fillet radius is 31mm while Fork-end's edge fillet radius is 61mm. Cylindrical ends of both eye and fork have radius of 1.1r. We see that by decreasing top ring diameter, Maximum Shear Stress increases in assembly. As top ring diameter is decreased from 103.55mm to 71.5mm, stresses increase from 807MPa to 849MPa. These results are computed with constant rod angles

B. 3Vs Truss Structure (Change in Rod Diameter)

For 3Vs truss structure, diameter of rods was changed. Top ring diameter of truss for first 3 cases was 103.55mm while for the last case, it was 120mm. Edge fillets of both eye and fork end are same as before. As we increased diameter of rods from 24mm to 35mm, Maximum Shear Stress decreased from 859 MPa to 744 MPa. This decrease in critical Maximum Shear Stress is because of decrease in stresses in truss structure. The trend obtained by analysis of different diameters is shown in figure 10.

C. Straight rod Truss Structure (Change in Rod Diameter)

Straight rods are also inclined rods however unlike V-truss structure; they are not joined together at the top ring into a V. Edge fillet radius at fork end is 61.5mm while at the eye end, its 31.5mm. Results of analyses of whole assembly of rod diameter change, for this specific support structure, are plotted in figure 11. The trend is same as it is in the case of 3V truss for reasons stated above. Stresses decrease from 481MPa to 364.67MPa as diameter is increased from 15mm to 35mm.

D. 4Vs Truss Structures (Change in Rod Diameter)

Edge fillet for both ends are same as in straight rods case. Diameter of rod is increased from 24mm to 40mm and stresses decrease from 418MPa to 346MPa as shown in figure 11.

E. Fillet Radius Change at Fork and Eye End

To see the development of stresses by changing edge fillet radius at both ends, we increased the edge radius one by one and ran simulations. From results given in table 2, we see that increasing edge radius significantly reduces stresses. This is because any sudden change in geometry causes stress concentrations. Therefore, the greater the edge radii is, the gradual the change in geometry occurs, thus avoiding stress concentrations. For this analysis, the diameter of rods of 3V truss structure was constant i.e. 35mm and top diameter of ring was 103.55mm.

Table 2: Effect of Fillet Radius Change

Fork End Fillet Radius (mm)	Eye End Fillet Radius (mm)	Critical Maximum shear Stress(MPa)
61	31	744
61.5	31	402
61.5	31.5	369

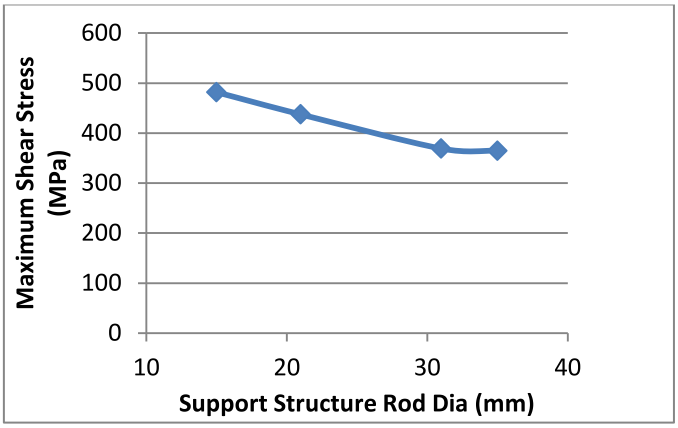

Figure 11: Effect of change of diameter for straight rod truss

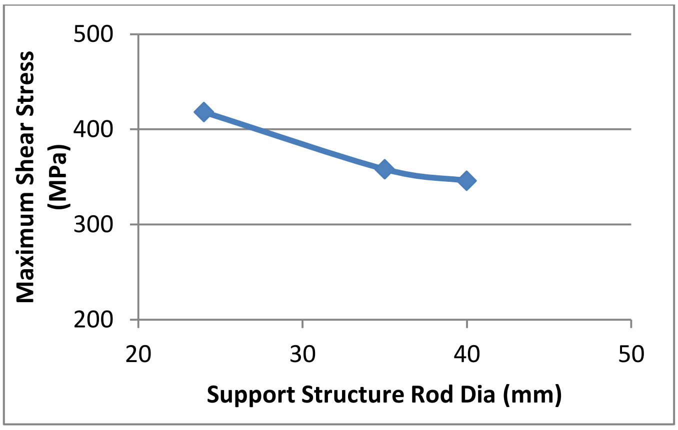

Figure 12: Effect of change of diameter for 4Vs

F. *Modiified Knuckle Joint*

Table 3 shows the changes in Maximum Shear Stress by altering the diameter of fork and eye rods. Initially these diameters were 2.2r. 4Vs support structure with rod diameter of 40mm was used.

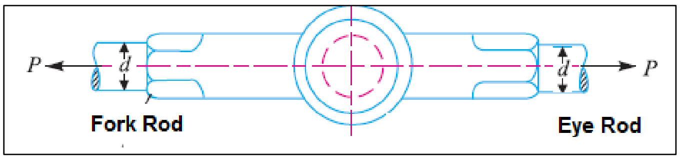

Figure 13: Modified diameter of Knuckle Joints

Table 3: Knuckle Joint Modifications

Eye Rod Diameter	Fork Rod Diameter	Critical Maximum Shear Stress
2.2r	2.5 r	284.13 MPa
2.5r	2.7r	218.78MPa
2.5r	3r	176.59MPa
3r	3.4r	133.13MPa
3.2r	3.6r	123.46MPa

Here r is the radius of the rod's end calculated by [4]. In case of steel, this r=24.63mm.

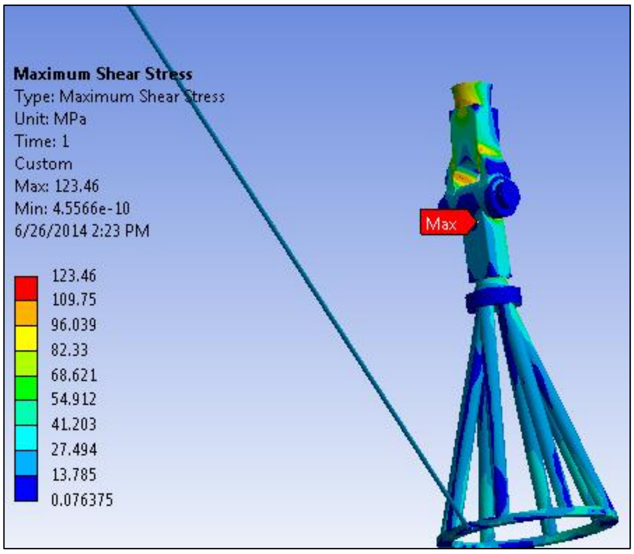

Figure 14: Final Assembly for Stainless Steel

V. Conclusion

The finalized design is shown in Figure 14. After a large number of simulations we have arrived to this final design based on the following facts:

- A 4 Vs-Support Structure results in minimum stress- as a comparison between all structures: 3Vs Structure, Straight Rods Structure and 4Vs structure with the same rod diameter (35mm), provided a minimum value of 358 MPa through 4Vs structure. Hence this 4Vs structure was further made better by increasing rod diameter to 40 mm
- Edge fillet radius should be maximum, providing a smooth gradual change in fork-rod and eye-rod diameters
- Changing knuckle joint dimensions provides drastic change in stress, as opposed to changing Support Structure dimensions. Also further increasing support structure rods diameter will simply increase weight.

VI. Future Work

Extensive analysis have been performed to test the assembly's axial stress bearing capacity, however, a rudimentary non-linear bending moment analysis is also to be performed. Furthermore, although stresses with FOS=1 are under control but due to the difference in practical and analytical work, it will be redesigned with FOS=2. Also using stainless steel gives us a large knuckle joint because its strength to weight ratio is relatively low. Hence we will perform sensitivity analysis of material and choose a material with higher strength to weight ratio, for instance, Maraging steel [12] so we can design a knuckle joint with minimum weight.

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