ILLINOIS INSTITUTE OF TECHNOLOGY

MMAE597: SPECIAL TOPICS

STEERING SYSTEM DESIGN FOR AN FSAE CAR

Hugo Aristegui Garcia

ILLINOIS INSTITUTE OF TECHNOLOGY

Contents

LIST OF FIGURES/ TABLES/ DRAWINGS 4
Figures 4
Tables 4
Drawings 5
INTRODUCTION 6
CONTEXT 6
Formula SAE 6
Static Events 7
Dynamic Events 7
Rules 9
SCOPE 10
ANALYSIS OF ALTERNATIVES 11
Location of the rack-pinion 11
Connection of the steering wheel to the rack-pinion 11
Shaft section 14
METHODOLOGY 16
Force calculation 16
Double universal joint 17
Shaft-Union 17
Mesh 19
Boundary Conditions 19
Von Mises Stress 19
Inserts 20
Mesh 21
Boundary Conditions 21
Von Mises Stress 22
Deformation 23
Mesh 24
Boundary Conditions 24
Von Mises Stress 25
Deformation 25

Shaft 26

Mesh 27

Boundary Conditions 27

Von Mises Stress 27

Deformation 28

Attachment tubes 28

Mesh 29

Case 1 29

Case 2 30

Case 3 31

BUDGET 33

CONCLUSIONS 34

BIBLIOGRAPHY 35

APPENDIX I: DRAWINGS 36

APPENDIX II: TEMPLATES 44

ILLINOIS INSTITUTE OF TECHNOLOGY

LIST OF FIGURES/ TABLES/ DRAWINGS

Figures

Figure 1. IIT's FSAE car 7
Figure 2. SkidPad Layout 8
Figure 3. T3.11.14 9
Figure 4. IIT2016 car sketch 12
Figure 5. 95th Percentile Template 12
Figure 6. Alpha angle 13
Figure 7. Forces in the Upright 16
Figure 8. Double Universal Joint 17
Figure 9. Shaft-union First Design 17
Figure 10. Shaft-union Final Design 18
Figure 11. Shaft-union mesh 19
Figure 12. Shaft-union stresses 19
Figure 13. Shaft-union deformation 20
Figure 14. Insert 1 mesh 21
Figure 15. Insert 1 Two bolts stresses 22
Figure 16. Insert 1 One bolt stresses 22
Figure 17. Insert 1 Two bolts deformation 23
Figure 18. Insert 1 One bolt deformation 23
Figure 19. Insert 2 mesh 24
Figure 20. Insert 2 stresses 25
Figure 21. Insert 2 deformation 25
Figure 22. Shaft mesh 27
Figure 23. Shaft stresses 27
Figure 24. Shaft deformation 28
Figure 25. Auto-lubricated plastic bushing 28
Figure 26. Attachment mesh 29
Figure 27. Attachment stresses case 1 29
Figure 28. Attachment deformation case 1 30
Figure 29. Attachment stresses case 2 30
Figure 30. Attachment deformation case 2 31
Figure 31. Attachment stresses case 3 31
Figure 32. Attachment deformation case 3 32
Figure 33. Steering System CAD 34

Tables

Table 1. Competition Points 9
Table 2. Different alternatives 11
Table 3. Alpha angles 13
Table 4. Hollow sections 14
Table 5. Lay orientation 26

Table 6. Components budget 33
Table 7. Manufacturing budget 33
Table 8. Double Universal Joint measures 43

Drawings

Drawing 1. Attachment 36
Drawing 2. Bushing 37
Drawing 3. Insert 1 38
Drawing 4. Insert 2 39
Drawing 5. Shaft 40
Drawing 6. Shaft-union 41
Drawing 7. Double Universal Joint 42
Drawing 8. Template 1 44
Drawing 9. Template 2 45

INTRODUCTION

In this document it is presented the design of a new steering system for the IIT FSAE race car developed each year at the Illinois Institute of Technology.

In order to design a good racing car, the conditions which the car will be exposed at has to be taken into account. At the same time the steering system is able to withstand the external requirements, the design of it has to deal with the rules and the designs of other parts of the car.

The objective is to improve the actual steering system, which is a direct transmission with a steel tube. There are several things to improve of that design, such as ergonomics, security, weight… It is really important for the driver to be comfortable with the steering system because in some dynamic events he may have to drive the car during half an hour.

Once the different alternatives have been contemplated and chose the best one, the whole system will be designed in CAD and calculated in ABAQUS. This will let us choose the materials we will use in order to optimize the weight.

This document also includes a budget of the whole steering system including the price of the different parts as well as the price of the process needed to manufacture its components.

CONTEXT

Formula SAE

Formula SAE is a student design competition where the students have to design, build and test a Formula-style car to compete with it later. The competition started in 1978 in USA but now it is present in many countries in Europe as well. As the time goes on, the level present in the competition has increased greatly, in fact, the world record of acceleration of an electric car is hold by a Swiss team (from 0 to 62 mph in 1.5 sec).

Figure 1. IIT's FSAE car

The competition has many different events and they are divided in two major categories; static and dynamic.

Static Events

Design Event

The students have to justify their designs to a group of judges who usually are people closely related to Formula 1. The maximum points that a team can achieved in this event are 150. The judges are not only looking for a good design, but for a lot of testing as well.

Cost Event

It would be easy for a team with a budget of millions to develop a much better car than one who has a tight budget. For this reason, this event consists in defending in front of the judges the Bill of Materials of the car.

Business Case Event

Finally, to complete the statics events it has to be some marketing. In the Business Case Event one team member has to try to sell the car as the team was a fictional company and the judges were investors.

Dynamic Events

Acceleration Event

The opening of the dynamic events, is the one which is less related with the driver's skills. It is a 75 meters long straight and the car has to do it as fast as it can.

SkidPad Event

This event is meant to test the cornering behavior of the car. First it has to do two laps in the right circle and then other two laps in the left one. The time in this event is the average time of the four laps.

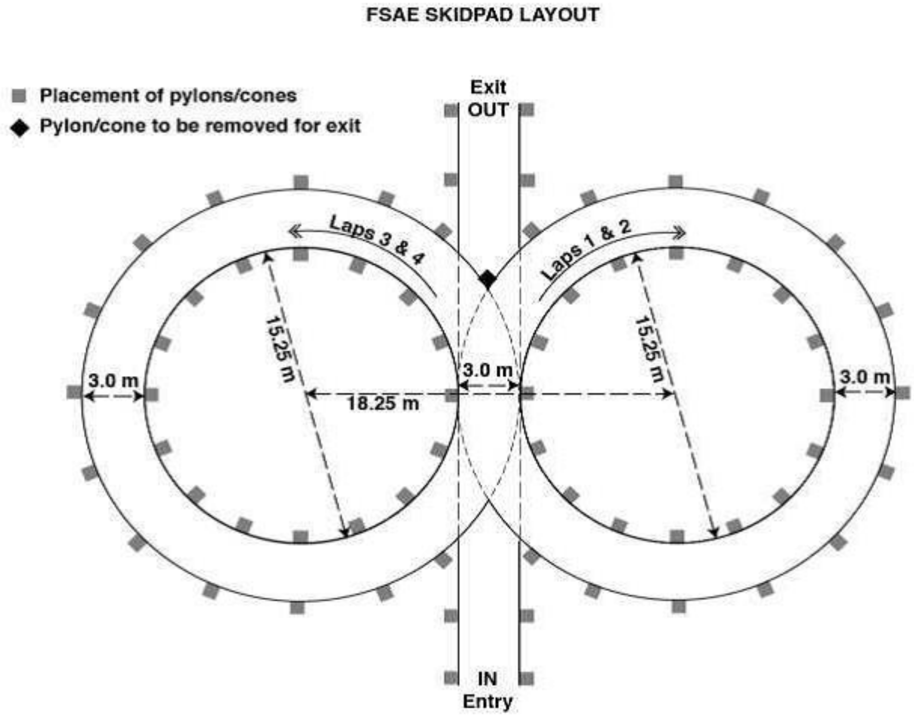


Figure 2. SkidPad Layout

Autocross Event

The Autocross event consists in a race circuit with many closed curves which gives more importance to acceleration than to top speed. Each driver has to chances and the fastest time is the one counted towards the classification.

Endurance Event

This is the greatest event and the hardest one. Each year few cars are able to finish it. The Endurance event consists of running in the same circuit of the Autocross event but this time it is about laps which makes a total distance of 22 km.

Fuel Economy Event

During the Endurance event the fuel consumption is recorded. At the end of the event the consumption of all cars that managed to finish the Endurance are ranked.

Event	Points
Design Event	150
Cost Event	100
Business Case Event	75
Acceleration Event	75
SkidPad Event	50
Autocross Event	150

Endurance Event	300
Fuel Economy Event	100
Total	1,000

Table 1. Competition Points

Rules

The design of the whole FSAE car is regulated by the FSAE rules that exists to maximize the safety of the vehicle not only for the driver but for the pedestrians as well. The rules that affect directly the steering system are:

- **T3.11.14** The top-most surface of the Front Hoop must be no lower than the top of the steering wheel in any angular position.

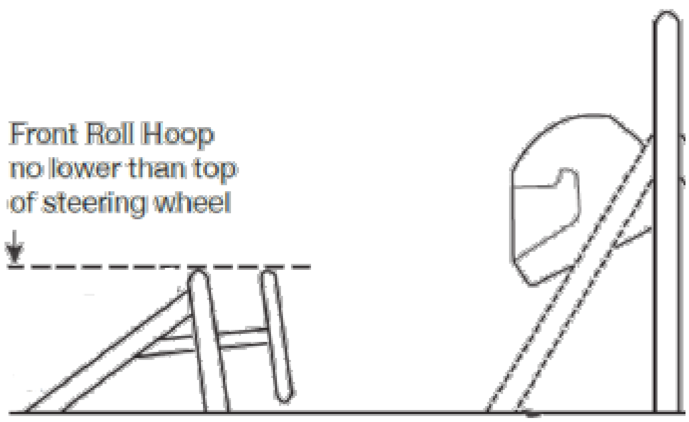


Figure 3. T3.11.14

- **T3.11.15** The Front Hoop must be no more than 250 mm forward of the steering wheel. This distance shall be measured horizontally, on the vehicle centerline, from the rear surface of the Front Hoop to the forward most surface of the steering wheel rim with the steering in the straight-ahead position.
- **T6.5.1** The steering wheel must be mechanically connected to the wheels, i.e. “steer-by-wire” or electrically actuated steering is prohibited.
- **T6.5.2** The steering system must have positive steering stops that prevent the steering linkages from locking up (the inversion of a four-bar linkage at one of the pivots). The stops may be placed on the uprights or on the rack and must prevent the tires from contacting suspension, body, or frame members during the track events.
- **T6.5.3** Allowable steering system free play is limited to seven degrees (7°) total measured at the steering wheel.
- **T6.5.4** The steering wheel must be attached to the column with a quick disconnect. The driver must be able to operate the quick disconnect while in the normal driving position with gloves on.

- **T6.5.5** Rear wheel steering, which can be electrically actuated, is permitted but only if mechanical stops limit the range of angular movement of the rear wheels to a maximum of six degrees (6°). This must be demonstrated with a driver in the car and the team must provide the facility for the steering angle range to be verified at Technical Inspection.
- **T6.5.6** The steering wheel must have a continuous perimeter that is near circular or near oval, i.e. the outer perimeter profile can have some straight sections, but no concave sections. “H”, “Figure 8”, or cutout wheels are not allowed.
- **T6.5.7** In any angular position, the top of the steering wheel must be no higher than the top-most surface of the Front Hoop.
- **T6.5.8** Steering systems using cables for actuation are not prohibited by **T6.5.1** but additional documentation must be submitted. The team must submit a failure modes and effects analysis report with design details of the proposed system as part of the structural equivalency spreadsheet (SES). The report must outline the analysis that was done to show the steering system will function properly, potential failure modes and the effects of each failure mode and finally failure mitigation strategies used by the team. The organizing committee will review the submission and advise the team if the design is approved. If not approved, a non-cable based steering system must be used instead.
- **T6.5.9** The steering rack must be mechanically attached to the frame. If fasteners are used, they must be compliant with **ARTICLE T11**.
- **T6.5.10** Joints between all components attaching the steering wheel to the steering rack must be mechanical and be visible at Technical Inspection. Bonded joints without a mechanical backup are not permitted.
- **T11.1.1** All threaded fasteners utilized in the driver’s cell structure, and the steering, braking, driver’s harness and suspension systems must meet or exceed, SAE Grade 5, Metric Grade 8.8 and/or AN/MS specifications.
- **T11.1.2** The use of button head cap, pan head, flat head, round head or countersunk screws or bolts in ANY location in the following systems is prohibited: (d) steering system
- **T11.2.1** All critical bolt, nuts, and other fasteners on the steering, braking, driver’s harness, and suspension must be secured from unintentional loosening by the use of positive locking mechanisms
- **T11.2.3** All spherical rod ends and spherical bearings on the steering or suspension must be in double shear or captured by having a screw/bolt head or washer with an O.D. that is larger than spherical bearing housing I.D.

SCOPE

The objective of this document is to explain the design of the steering system of a FSAE race car, optimizing the parts and choosing the best manufacturing method.

Firstly, there will be done an analysis of the different alternatives for the steering system and justifying the choice.

Secondly, the solution adopted will be explained in detail. The components of the steering system will be calculated using ABAQUS to check if they fulfill the mechanical requirements and optimize them.

Finally, the manufacturing methods will be explained as well as the method of assembly.

ANALYSIS OF ALTERNATIVES

Before being able to run, the car has to pass a technical inspection where the judges check that the car fulfill all the rules. Besides the rules just listed, there are two templates used to measure some minimum dimensions in the cockpit and from the steering wheel to the pedals. At the end of this document there is attached the drawings of this templates.

Until this point it is defined all the requirements of the rules that the steering system must fulfill, the next step is to decide the type of the steering system. These are the possible configurations of the steering system:

	Option 1	**Option 2**	**Option 3**
Location of the rack-pinion	Over the legs	Under the legs	
Connection steering wheel to rack-pinion	Direct	Universal joints	Gearbox
Shaft section	Round	Square	Hexagonal
Shaft material	Steel	Aluminum	Carbon fiber
Rack-pinion	Commercial	Homemade	

Table 2. Different alternatives

Location of the rack-pinion

The first step is to think where to place the rack-pinion and the advantages of locating it over or under the driver's legs. The conclusion was that locating it over the driver's legs would increase the height of the center of gravity, besides it would be more difficult to fulfill with the rule of the second template. So the decision is to place the rack-pinion under the driver's legs attached to the chassis.

Connection of the steering wheel to the rack-pinion

In order to make a decision about this, it is necessary to have some dimensions of the car and calculate which would be the angle of the steering wheel. This is really important because the ergonomics is a fundamental part of the car and not taking them into account would be a big mistake.

The following image is a basic drawing of the side-view of the car:

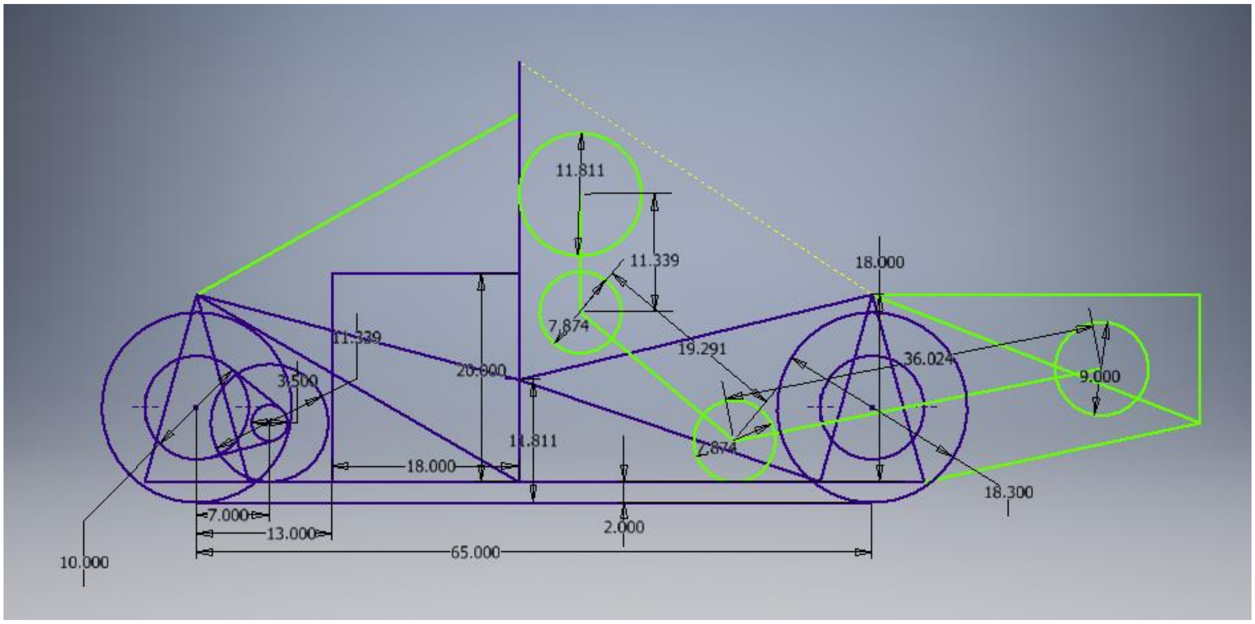


Figure 4. IIT2016 car sketch

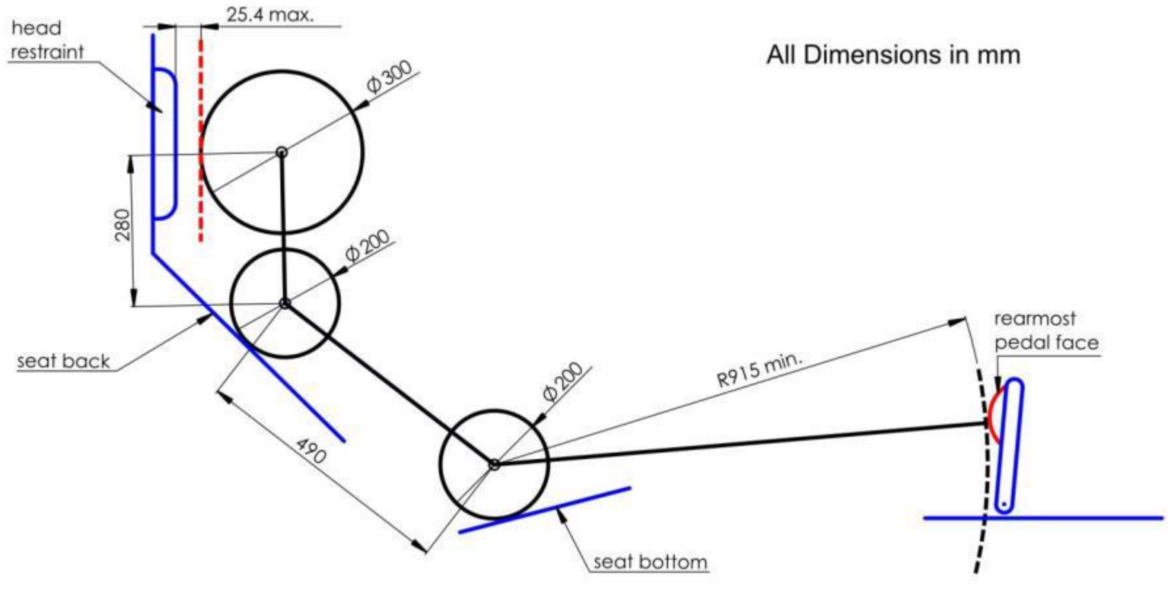


Figure 5. 95th Percentile Template

The second image includes another template which represents the measures of the driver, this is because the car must be designed for drivers whose stature varies from a 5th percentile female to a 95th percentile male.

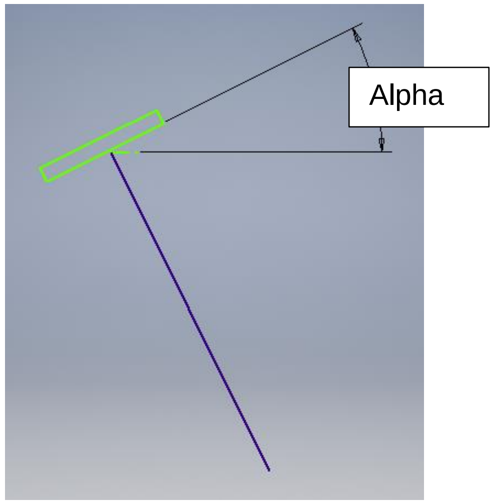


Figure 6. Alpha angle

Alpha is the angle between the steering wheel and the horizontal plane while in the street cars it has an angle of about 60-70 degrees, due to the position of the driver in the FSAE car this angle should be between 70 and 90 degrees.

With basic trigonometry the angle alpha can easily be obtained for the different steering systems:

Column connection	**Alpha**
Direct column	23 degrees
One cardan joint	35 degrees
Two cardan joints	43 degrees
Double universal joint	80 degrees
Gear box	Any angle

Table 3. Alpha angles

The first systems to be discarded is the direct column and the cardan joints because the angles obtained are far from the desired. Now a decision has to be made between the double universal joint and the gear box.

Double universal joint

Pros:

- It is commercial and cheaper
- Probably it would be lighter
- It is easy to assemble

Cons:

- The maximum angle is 80 degrees

Gearbox

Pros:

- We can design for the desired angle

Cons:

- It is harder to assemble
- It is more likely to have manufacture delays and tolerance mistakes

The most comfortable angle alpha is located between 70 and 90 degrees and it should be test in an ergonomic bench and choose the one who adapt better to the majority of the drivers. The double universal joint let us work with alpha angles between 70 and 80 with is really good considering its low price comparing with the gearbox.

Shaft section

The section of the column is independent of the system chosen to connect the steering wheel and the rack-pinion, therefore different sections have been analyzed to find how well they withstand the shear stress compared one to each other.

To compare the sections, it has been calculated the maximum shear stress of each section with the constrains of an area of 23 mm^2 and a maximum outdoor diameter of 15 mm. All sections are hollow sections.

	Round section	**Square section**	**Hexagonal section**	**Octagonal section**
Dout/Lout (mm)	15	10.64	7.5	5.8
Din/Lin (mm)	13.99	9.49	6.88	5.37
Thickness (mm)	0.505	0.571	0.533	0.515
σ max (MPa)	479.92	690.89	558.42	515.76

Table 4. Hollow sections

It can be seen that the section that best withstand the shear stress is the round section and as we change to a section more different from this geometry, the maximum shear stress increases, however, the most noticeable change of the stress is between the hexagonal section and the square section. It increases a 20%.

The material is something critical to define the geometry of the shaft. A steel tube can be welded to the inserts at both ends, however, if the shaft is made of carbon fiber, the union with the inserts is crucial. There are two ways to join the insert and the shaft if it is made of carbon fiber. The first one is to make a hole in the shaft and use a bolt to join it with the insert. The problem of this is that the shaft properties would be compromised because a there would appear stress concentration near the hole. The second one is to use glue. This option has the problem that it is difficult to predict its failure. If the section is round the insert could unstick due to the torque, however, if the section is square, hexagonal or octagonal the problem where the insert can be unstuck from the shaft disappears.

Therefore, the best choice is to use a carbon fiber hexagonal shaft.

METHODOLOGY

Force calculation

In order to dimension the forces that the steering system suffers, it is needed to know the acceleration that the car suffers. The maximum accelerations suffered by an FSAE car during the race is 1.5 G while running in a corner with a radius of 3 meters (minimum radius of the competition). Knowing that, some calculations can be done:

W: weight of the car and the driver

W = 260 kg + 70 kg = 330 kg =3,234 N

Fc = 1.5 G = 4,851 N

In the least favorable case we would have a weight distribution between the front axle and the rear axle of 50%. This means that have of the centrifuge force would be applied on the front wheels.

Fcf = 0.5 x 4,851 = 2,425.5 N

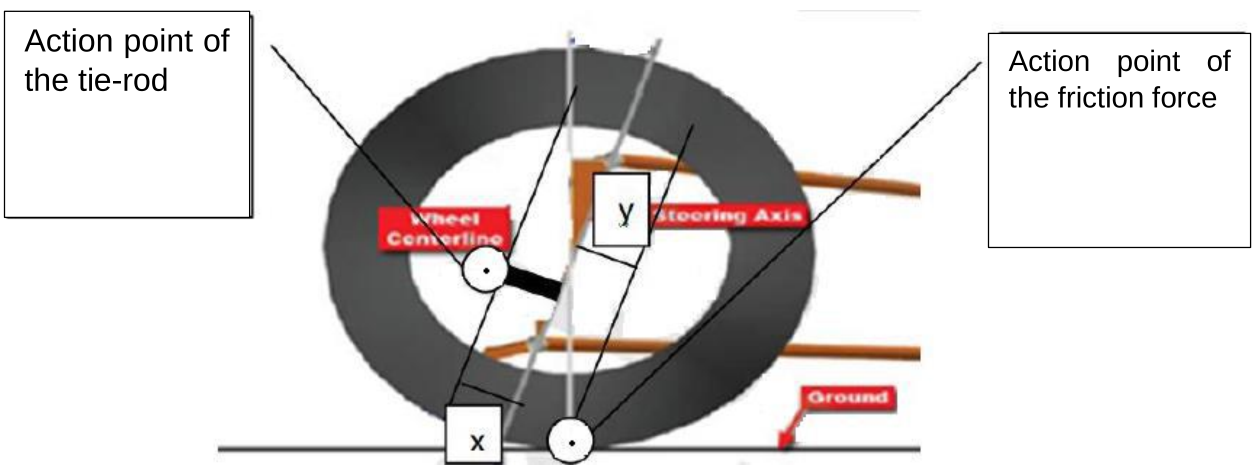


Figure 7. Forces in the Upright

The values for the upright of the car are x=60 mm and y=50 mm.

50 x 2,425.5 = Ft x 60

Ft = 2,021.25 N

With this force and the radius of the pinion of the rack the torque at the shaft can be calculated. In order to solve this, a radius chosen of 30 mm which is a safe assumption. This gives a total torque of 66 Nm. However, as the steering system is one of the most critical components of the car, the FSAE organization encourages the teams to design the shaft for a minimum torque of 80 Nm.

The first numerical calculus using that torque have shown that if the weight is optimized the carbon fiber is considerably the best choice.

Double universal joint

The double universal joints that are closer to our requirements are the shown in the following image with an operating angle of 80 degrees and a torque rating of 110 Nm.

Figure 8. Double Universal Joint

Shaft-Union

This component function is to connect the shaft of the quick-release to the double universal joint. The first idea was to connect it to the double universal joint with a turnkey, however it was not possible due to the high stresses that appeared.

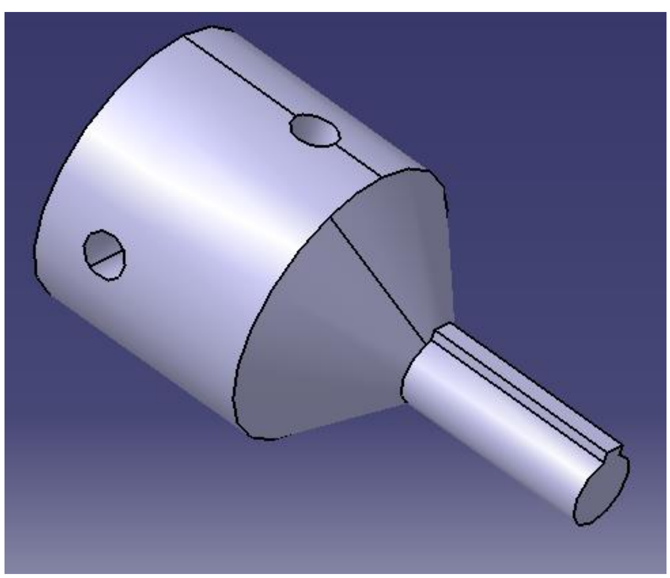


Figure 9. Shaft-union First Design

As it was not possible the double universal joint with turnkey was changed for a solid one.

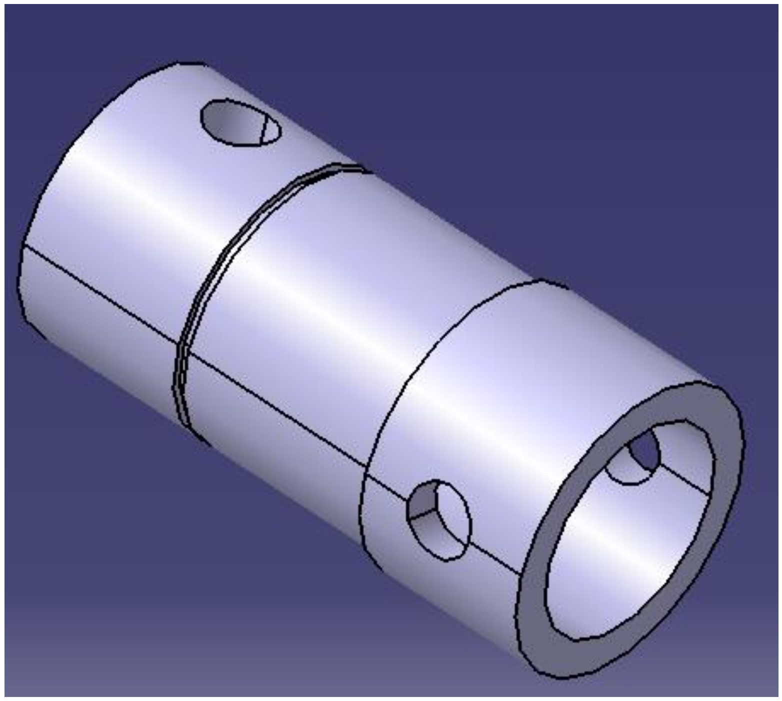
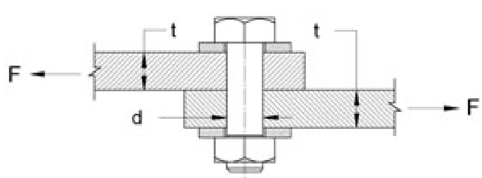


Figure 10. Shaft-union Final Design

Another change was made, instead of having four holes for two passing bolts on each side, now it has just two holes for one bolt. This is because of the following calculations:



$$Shear\ stress\ ave. = \frac{4F}{\pi\, d^2}$$

$$Bearing\ Area\ Stress = \frac{F}{t\, d}$$

$$Allowable\ Stress = \frac{Ultimate\ stress}{Factor\ of\ Safety}$$

Bolt or Pin Single Shear Stress	
Applied Force F (N, lbs) =	5000.00
Bolt/Pin Diameter d (mm, in) =	6.70
Plate Thickness t (mm, in) =	3.50
Ultimate (Yield Min.) Stress (N/mm^2, lbs/in^2) =	586.00
Factor of Safety =	2.50
Results	
Section Area of Bolt/Pin (mm^2, in^2) =	35.256
Shear Stress ave Bolt/Pin (N/mm^2, lbs/in^2) =	141.82
Bearing Area Stress B_t (N/mm^2, lbs/in^2 =	213.22
Allowable Stress (N/mm^2, lbs/in^2) =	234.40

The ultimate yield stress is the one corresponding to a 5 grade bolt.

The results show that one bolt is enough to transmit the torque. The objective of only using one bolt is to shorten this piece. If we use two bolts on each side it would be quite longer and the longer this piece is, the bigger the angle the double universal joint must work.

Mesh

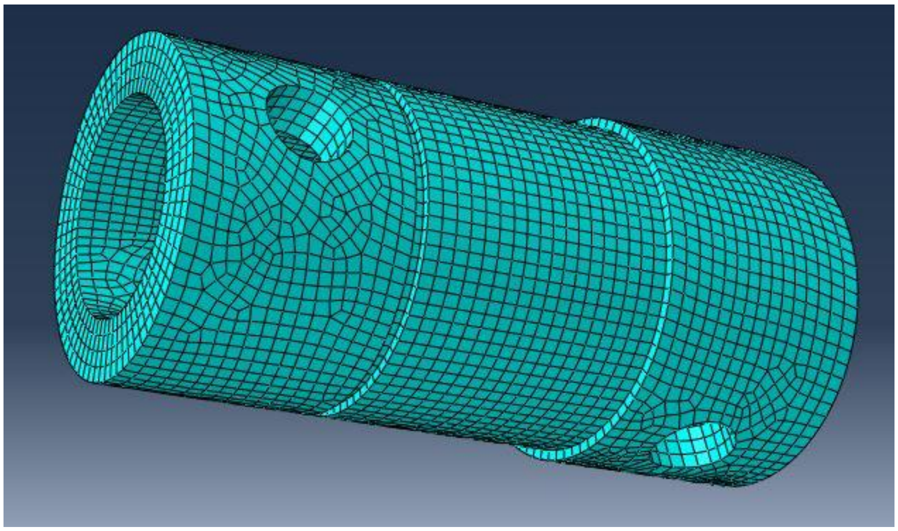


Figure 11. Shaft-union mesh

Boundary Conditions

Two holes are fixed and there is applied an 80 Nm torque in the other two.

Von Mises Stress

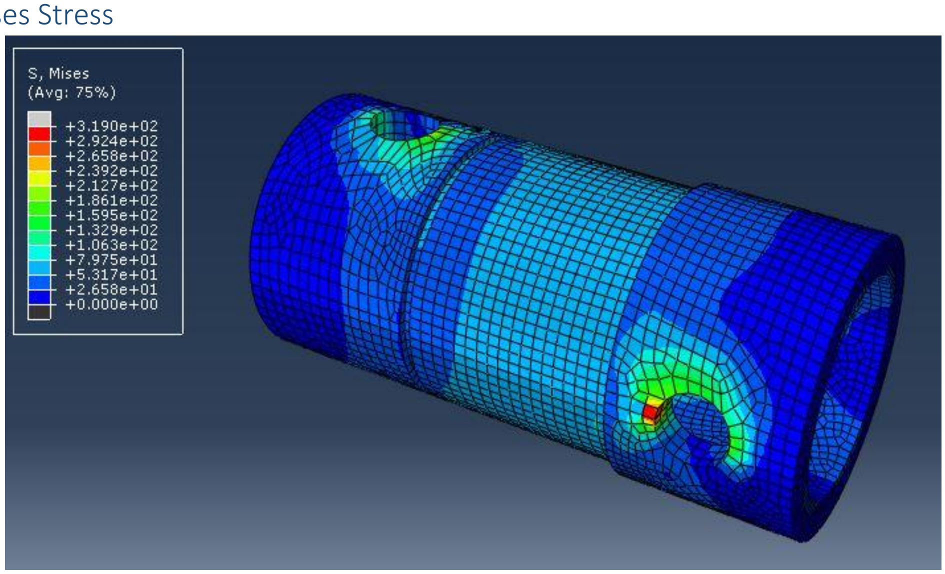


Figure 12. Shaft-union stresses

There is some minor stress concentration in the holes and all the component is under the yield stress of the Aluminum 7075 (500 MPa).

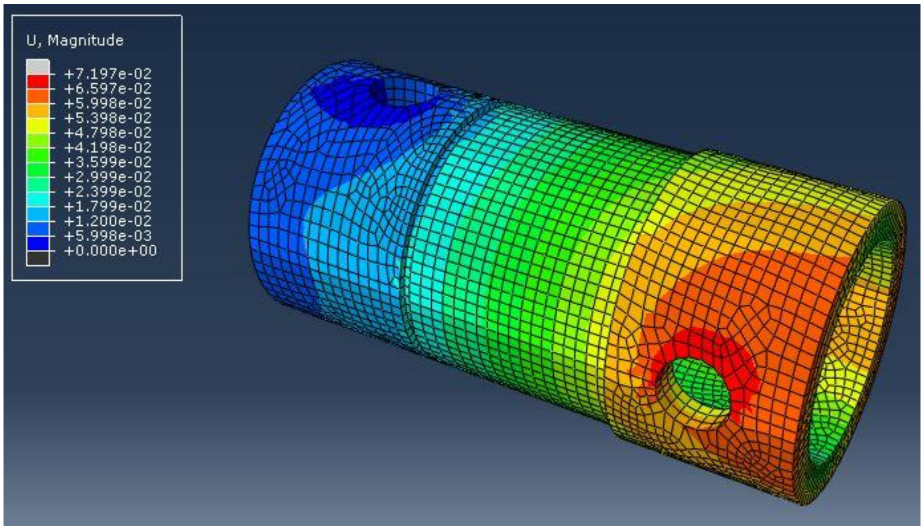


Figure 13. Shaft-union deformation

The maximum deformation is 0.072 mm so it is no significant.

Inserts

The inserts of the shaft are a key part of the design because the precision while designing and manufacturing are determinant for the good behavior of the steering system. This first insert is the one which connects the double universal joint with the shaft. There will be calculated with one and two bolts. Besides, although there haven't been included, there have been done several design of them with different diameters and changing the length of the part that goes inside the shaft. Thanks to this, the weight has been optimized maintaining a good safety coefficient.

Mesh

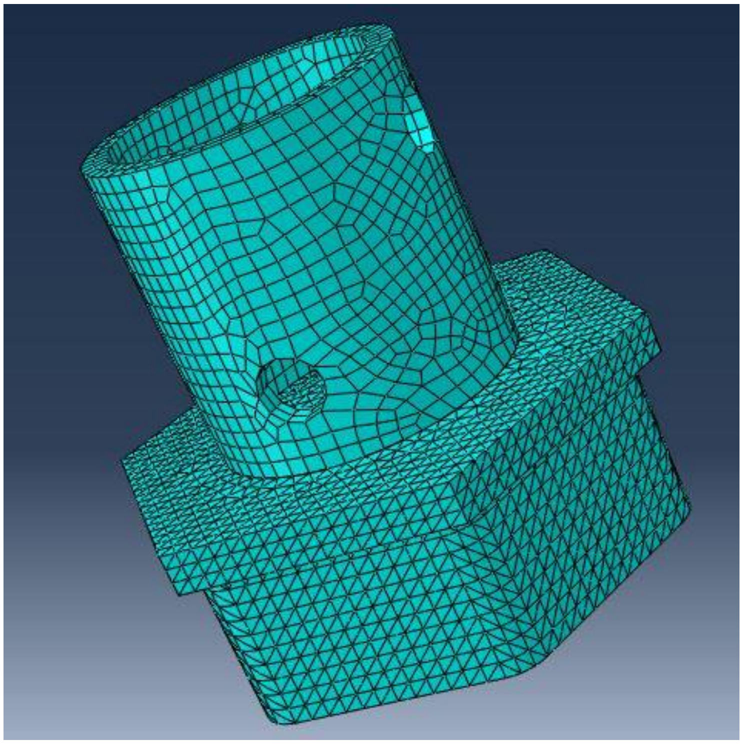


Figure 14. Insert 1 mesh

The cylindrical part is meshed with hexagons while the rest is meshed with tetrahedrons due to its complex geometry.

Boundary Conditions

The exterior faces of the hexagon are fixed, while an 80 Nm torque is applied in the faces of the holes.

Von Mises Stress

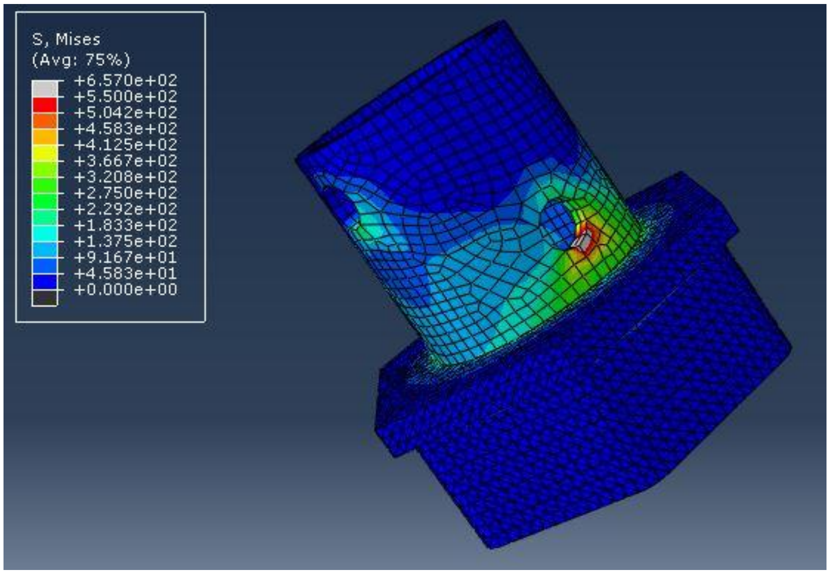


Figure 15. Insert 1 Two bolts stresses

There is a small stress concentration in the holes (657 MPa) but it will yield distributing the stress around. The stresses in the rest of the part are under the yield stress (500 MPa).

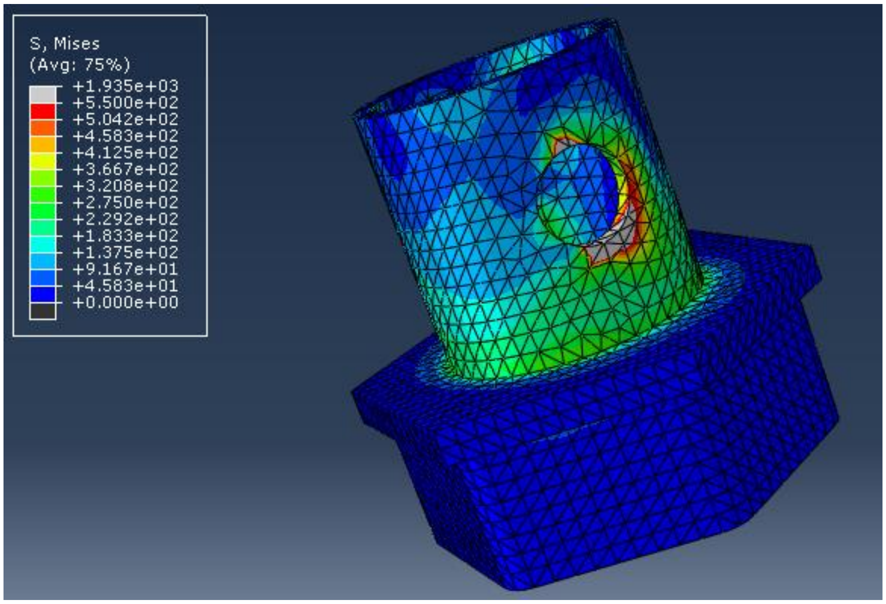


Figure 16. Insert 1 One bolt stresses

In the same component but this time with only two holes (one bolt), the area with stress above the yield stress has increased considerably. Therefore, the best design is the one with four holes (two bolts).

Deformation

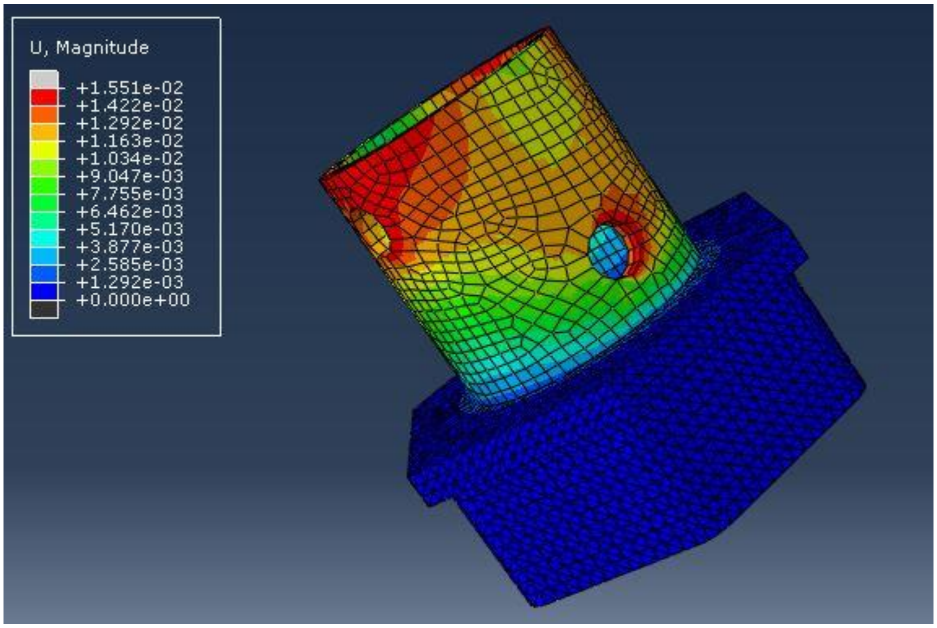


Figure 17. Insert 1 Two bolts deformation

The maximum deformation is 0.0155 mm so it is no significant.

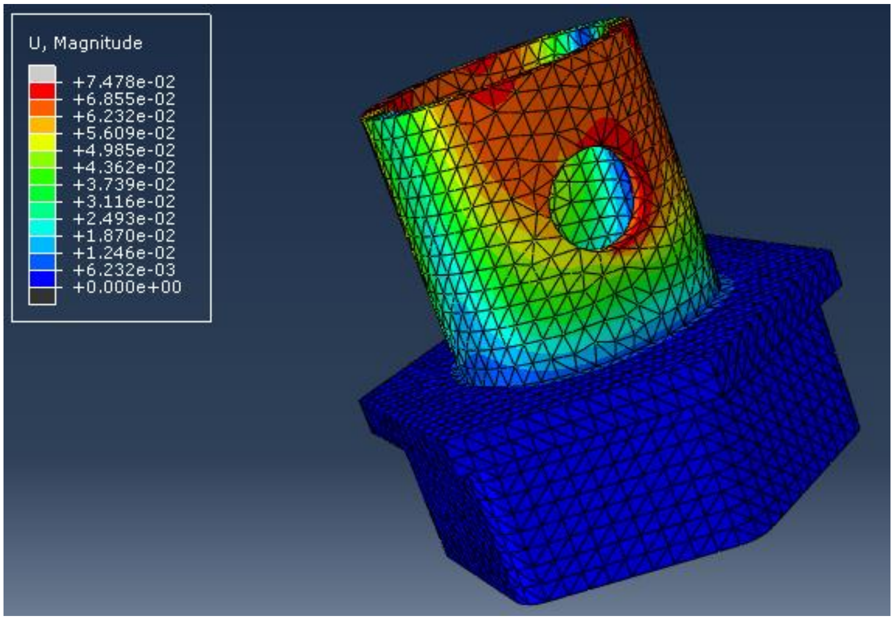


Figure 18. Insert 1 One bolt deformation

Again, with two holes the maximum deformation increases, it is still no significant but the stiffness of the part has decreased considerably.

The second insert is the one corresponding to the lower part of the shaft which connects it with the rack. In this component the torque is transmitted through the internal tooth to the spline shaft of the rack. The hole is for a small bolt to avoid the axial movement of the spline shaft, not to transmit any torque.

Mesh

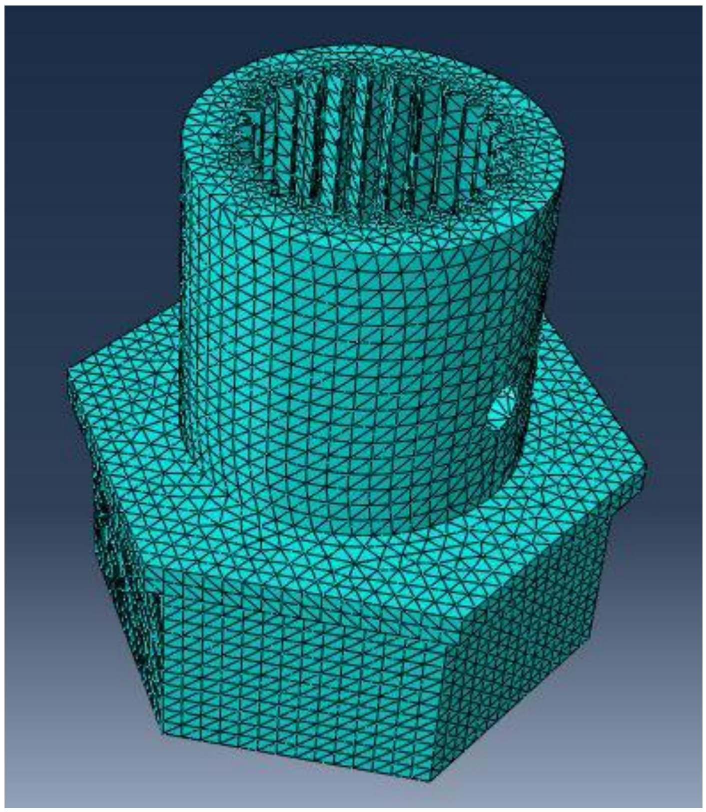


Figure 19. Insert 2 mesh

The mesh is done only with tetrahedrons because the complex geometry of the whole part.

Boundary Conditions

The faces of this component that are in contact with the carbon fiber shaft are fixed, then, an 80 Nm torque is applied in the inside faces of the insert that will contact the spline shaft.

Von Mises Stress

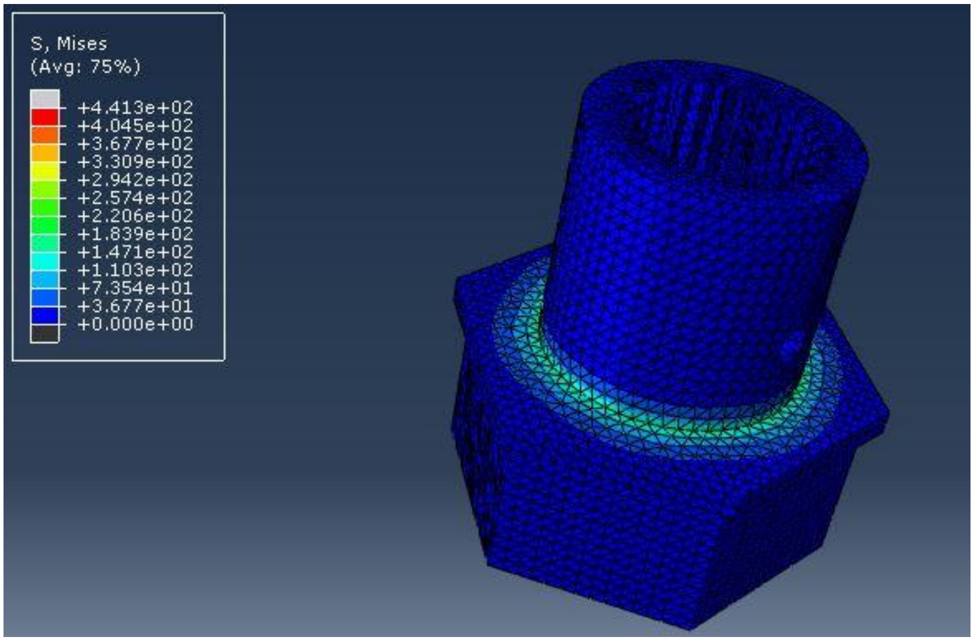


Figure 20. Insert 2 stresses

The whole component is under the yield stress.

Deformation

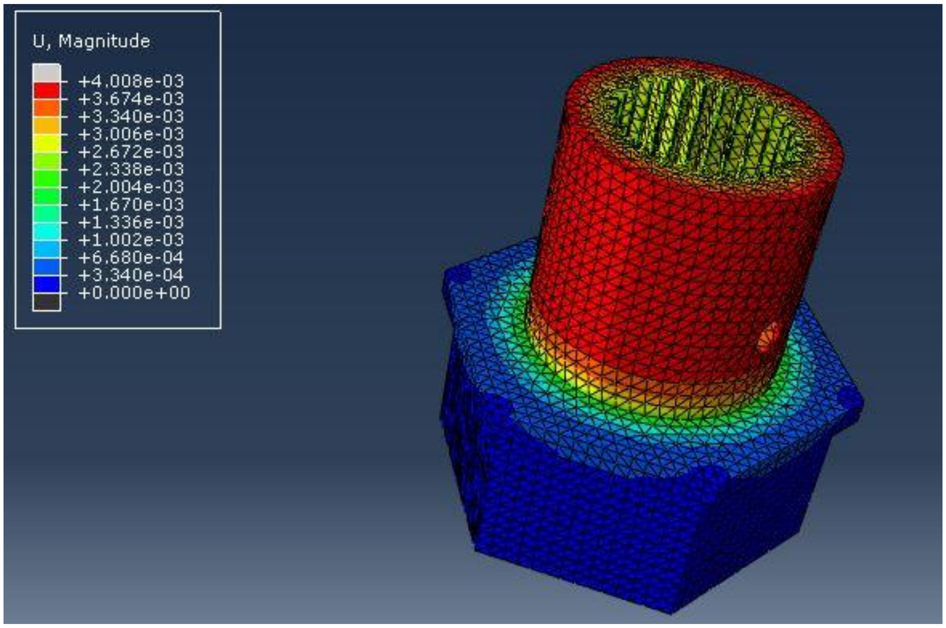


Figure 21. Insert 2 deformation

The maximum deformation is 0.004 mm so it is no significant.

Shaft

The shaft is the component of the design which will let reduced weight of the overall system. As the double cardan joint and the inserts have increased the weight comparing with the last year's system, the improvement of the shaft is such that can overcome the increases of the other components and reduce the overall weight. The weight reduction is not because of a geometry change but because of a material change.

Last year's steering shaft was a steel tube. Its diameter was not too big but as the density of the steel is 4 times greater than the density of the carbon fiber it is easy to reduce the weight.

First calculations were made with excel to get some approximations and then, the size obtained was adapted to the closet one that was available commercially.

The layer distribution of the shaft is:

SKU: 25600 Rev D		
Ply #	**Orientation**	**Location**
1	0	Inside
2	0	↓
3	45	
4	-45	
5	45	
6	-45	
7	0/90	Outside

Table 5. Lay orientation

This layer distribution is a great combination because the four layers at $\pm 45°$ maximizes the torsional stiffness while the $0°$ and the $90°$ maximizes the bending and crushing stiffness respectively.

Mesh

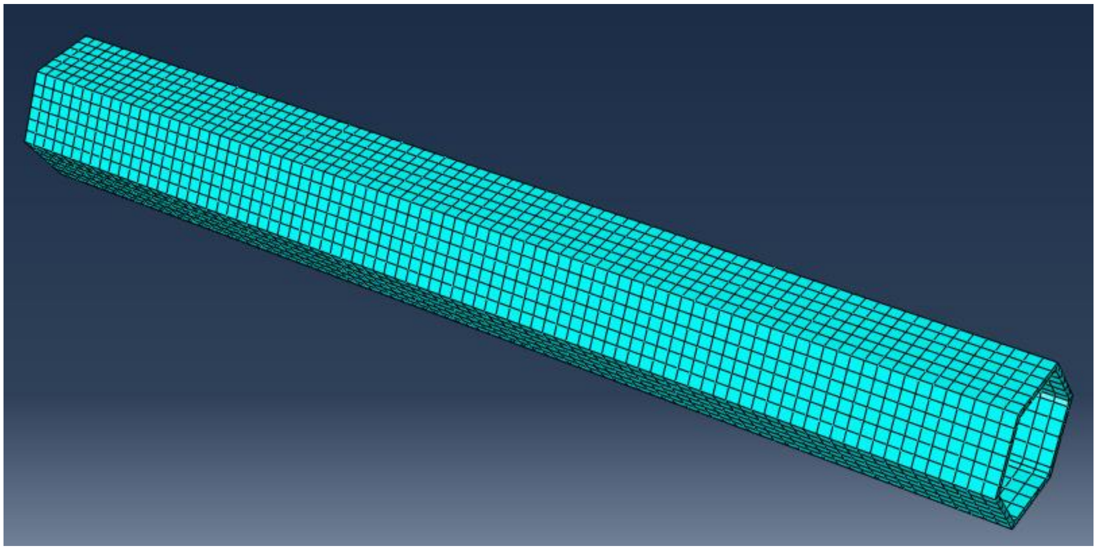


Figure 22. Shaft mesh

The simple geometry of the shaft is meshed with hexahedrons.

Boundary Conditions

In order to emulate the solicitations that the shaft will suffer during its worst conditions, one end is fixed while the other end has the insert tied and an 80 Nm load is applied to the holes of the insert.

Von Mises Stress

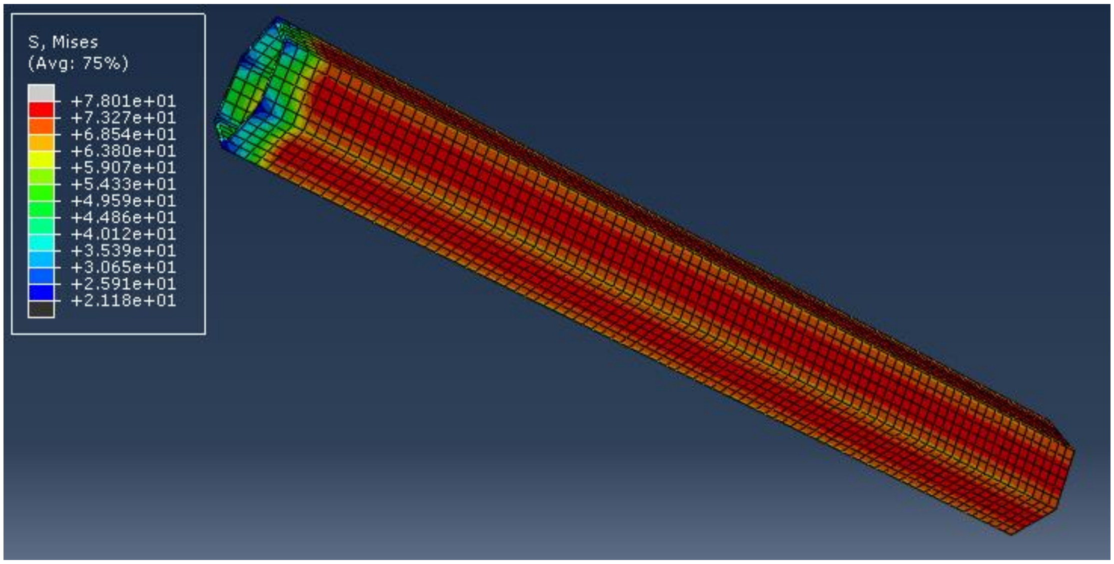


Figure 23. Shaft stresses

The maximum stress that suffers the shaft is 73.27 MPa which are low enough to not be a problem.

Deformation

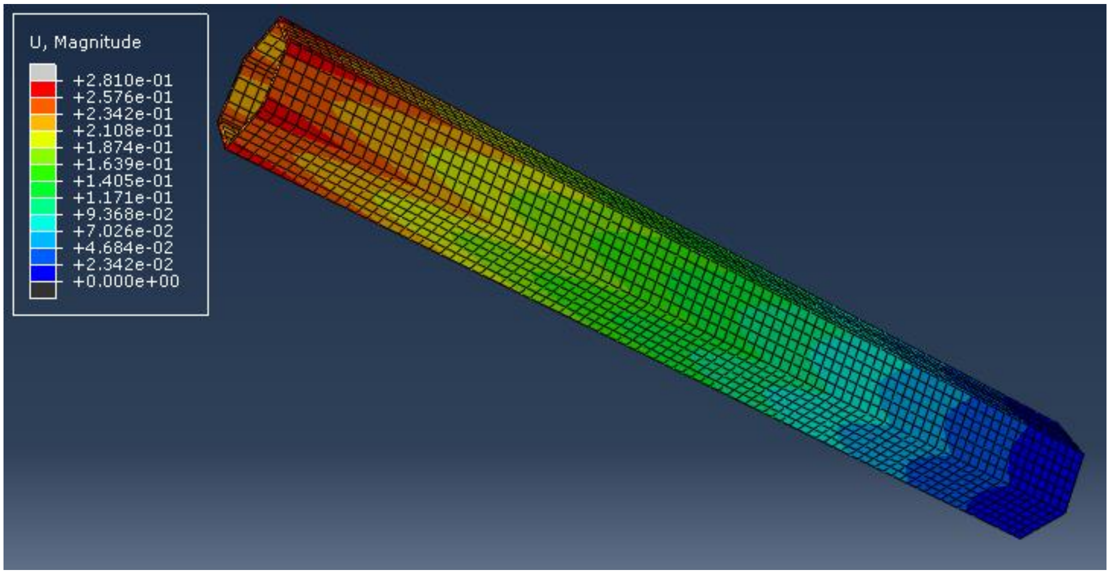


Figure 24. Shaft deformation

The maximum deformation is 0.281 mm, which means that the shaft has a really good stiffness.

Attachment tubes

Another thing to take into account is whether use a bearing or a bushing in the attachment of the shaft-union to the chassis. As the rotation speed of this component is low, the best choice for its price, weight and ease of assembly is a plastic bushing. A common material for the bushing is copper, however there are a new kind of plastic bushing that are auto-lubricated.

Figure 25. Auto-lubricated plastic bushing

In order to fix the position of this plastic bushing in the shaft-union, there will be used a circlip. The bushing goes inside a steel tube welded to two tubes that are welded to the chassis. These steel tubes are made of AISI-4130.

The axial and lateral forces used to calculate the attachment are the ones recommended by Steven Fox, a FSAE judge chief.

Mesh

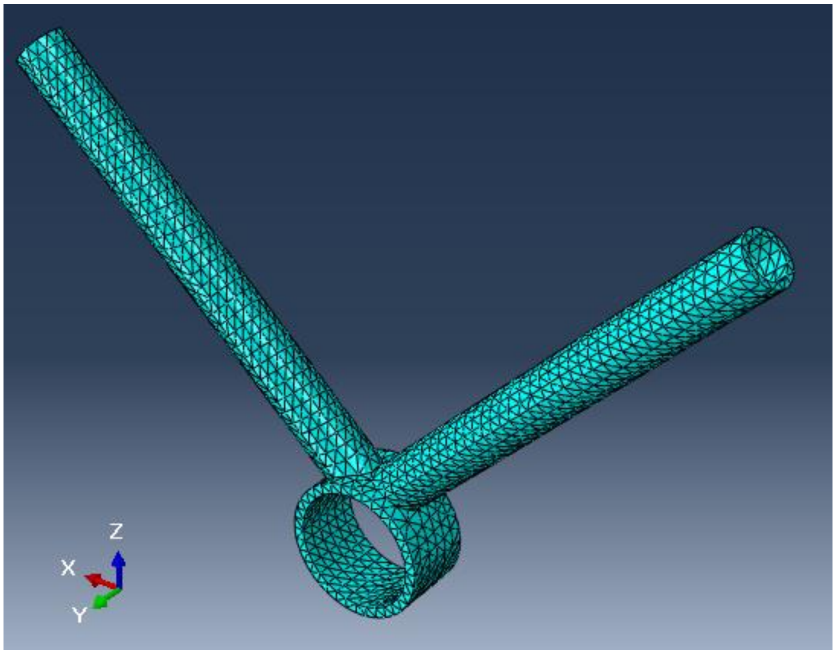


Figure 26. Attachment mesh

Case 1

Attachment fixed at the ends and a 660 N load applied in the direction of the Y-axis to the shaft joint.

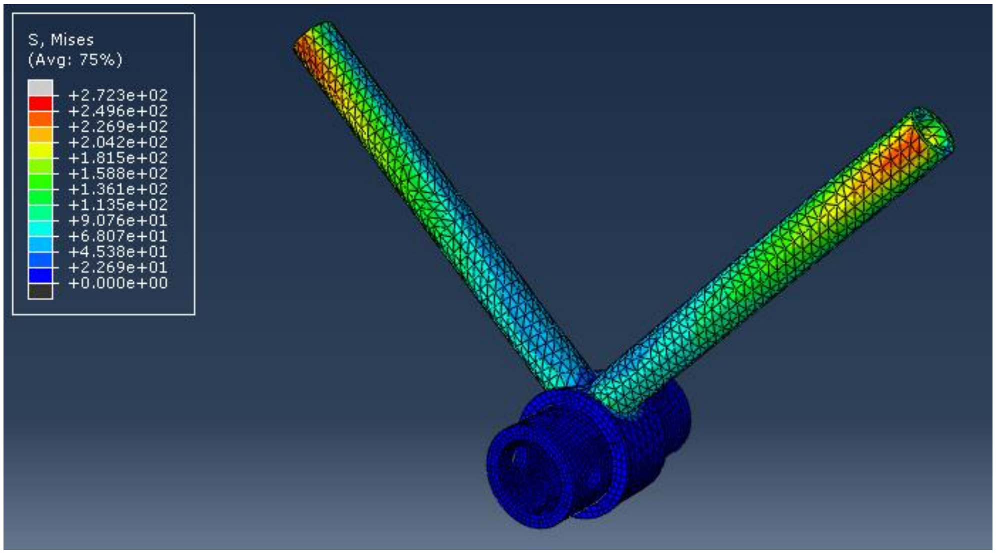


Figure 27. Attachment stresses case 1

With a maximum stress of 273 MPa, all the stresses are lower than the yield stress for the AISI-4130 (460MPa).

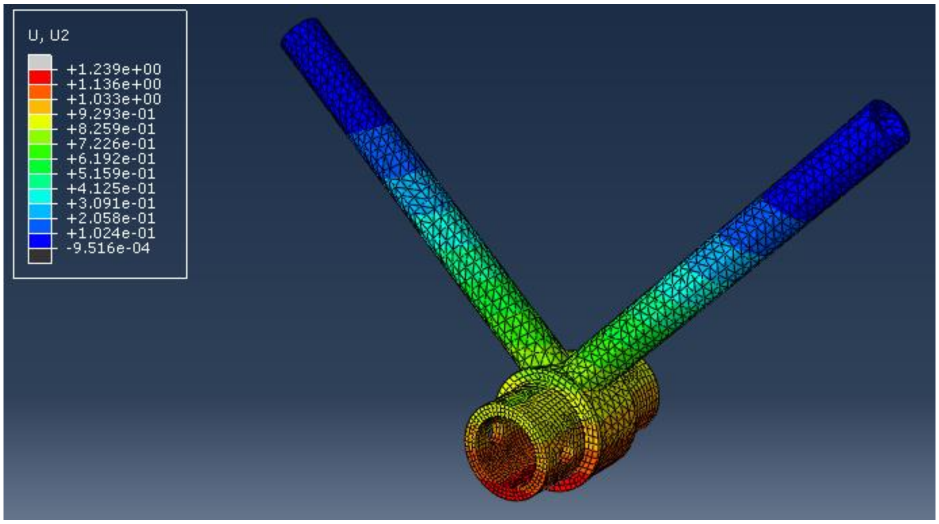


Figure 28. Attachment deformation case 1

The maximum displacement in the Y direction is 1.24 mm. This is significant but not enough to compromise the well performance of the steering system.

Case 2

Attachment fixed at the ends and a 660 N load applied in the direction of the X-axis.

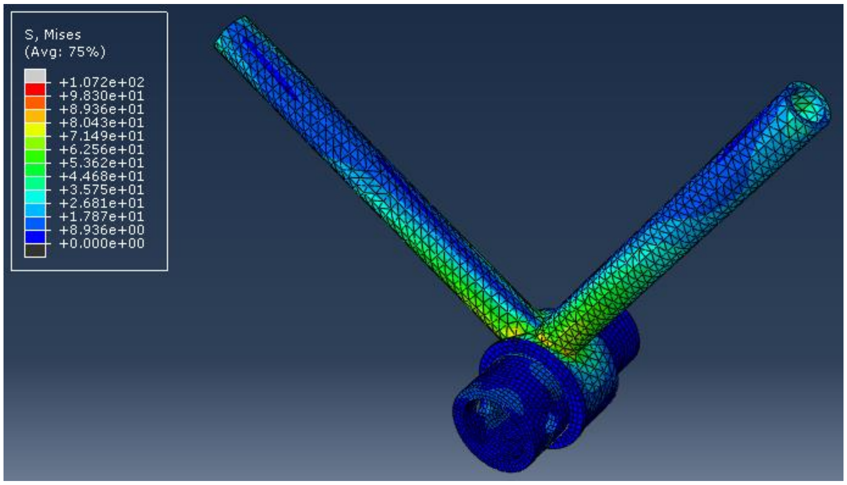


Figure 29. Attachment stresses case 2

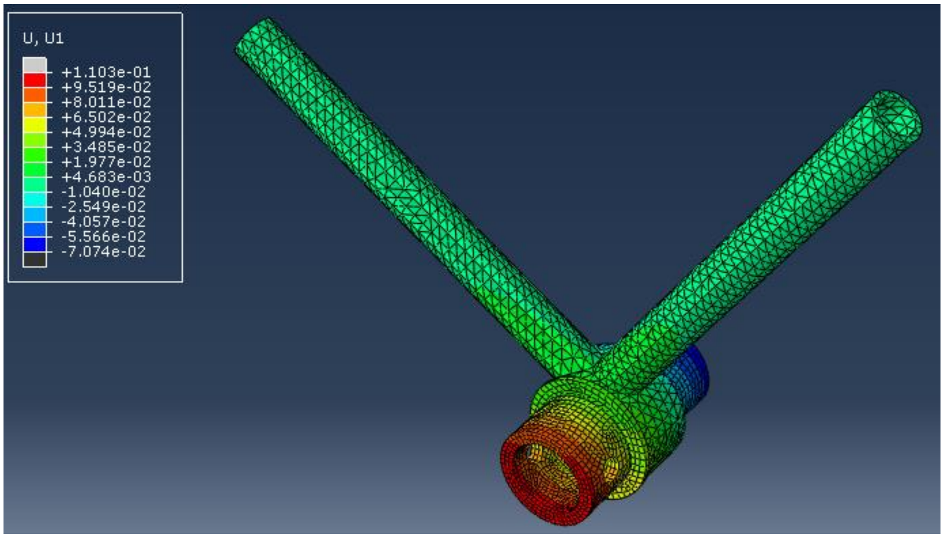


Figure 30. Attachment deformation case 2

Case 3

Attachment fixed at the ends and a 660 N load applied in the direction of the Z-axis.

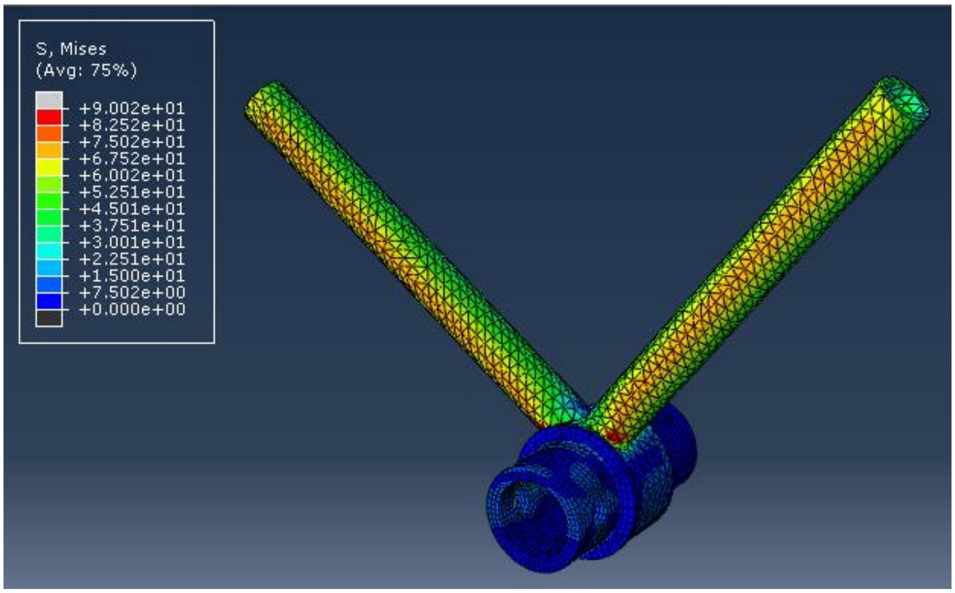


Figure 31. Attachment stresses case 3

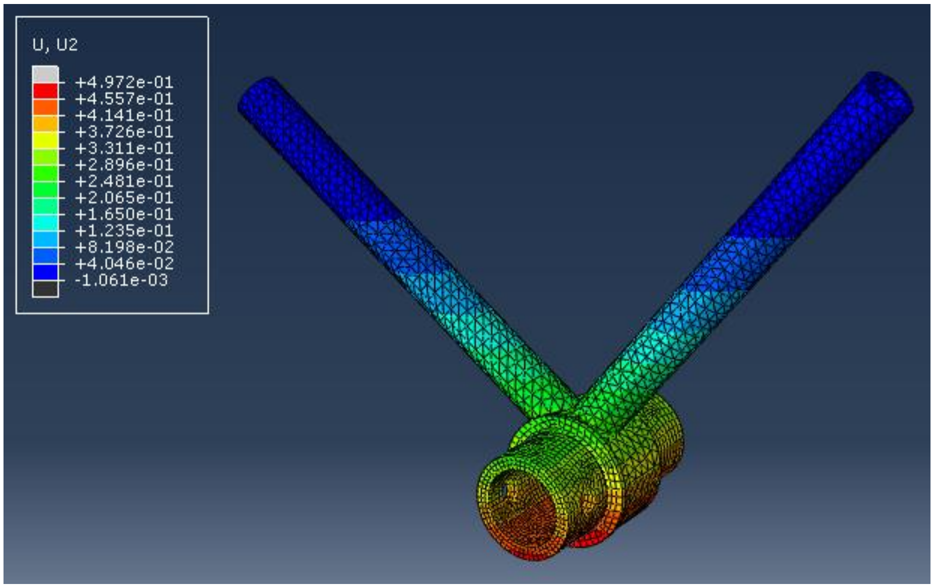


Figure 32. Attachment deformation case 3

In conclusion, the worst case is when a 660 N force is applied in the Y-direction, however, the steering system design is able to withstand the displacements produced.

The same way that the other components has been optimized, the attachment tubes has been optimized. The low stress suffered by the tubes makes think that the tube section could be reduced, however, if this is done, the displacements increase to a point that the correct performance of the system could be compromised.

BUDGET

Concept	Price
Steel tube	75 $
Aluminum	250 $
Carbon fiber	100 $
Steering wheel	250 $
Plastic bushing	5 $
Circlip	5 $
Rack-pinion	600 $
Double universal joint	75 $
Quick-release	60 $
Glue	20 $
Bolts and nuts	20 $
Total	1460 $

Table 6. Components budget

Concept	Price
Machining	1000 $
Welding	100 $
Total	1100 $

Table 7. Manufacturing budget

The total cost for the manufacturing and assemble of the steering system will be two thousand five hundred and sixty dollars. However, several components such as the rack-pinion, quick-release, and the steering wheel could be reused from the last year's car. Besides, the manufacturing could be done in the facilities of the Illinois Institute of Technology to reduce more the price. With these reductions the final budget will be around six hundred and fifty dollars.

Another thing to highlight is that in this budget there have been included only material, components and manufacturing cost which are the most important thing to take into account for the team. There are other costs that have not been included, these are the engineering hours and the price of the licenses of the software used as well as the amortization of the computer used.

CONCLUSIONS

The objectives have been met because not only the steering wheel position is the most comfortable for the pilot but the overall weight of the system has been reduced 20%. Besides, these two objectives have been met using a tight budget.

This steering system fulfills completely the competition rules, mandatory requisite to compete with the car.

The realization of this project has served to put in practice the knowledge obtained during the course MMAE 451 Finite Element Methods among others.

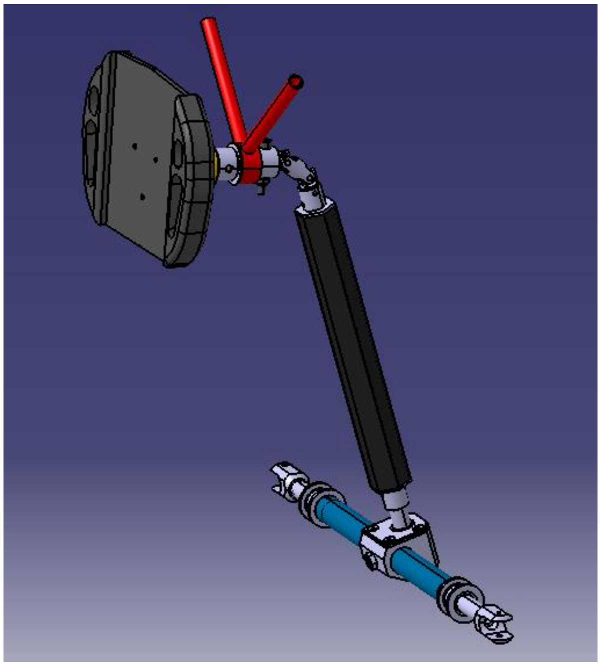


Figure 33. Steering System CAD

BIBLIOGRAPHY

FSAE (2016). 2016 FSAE Rules. Michigan USA: SAE International

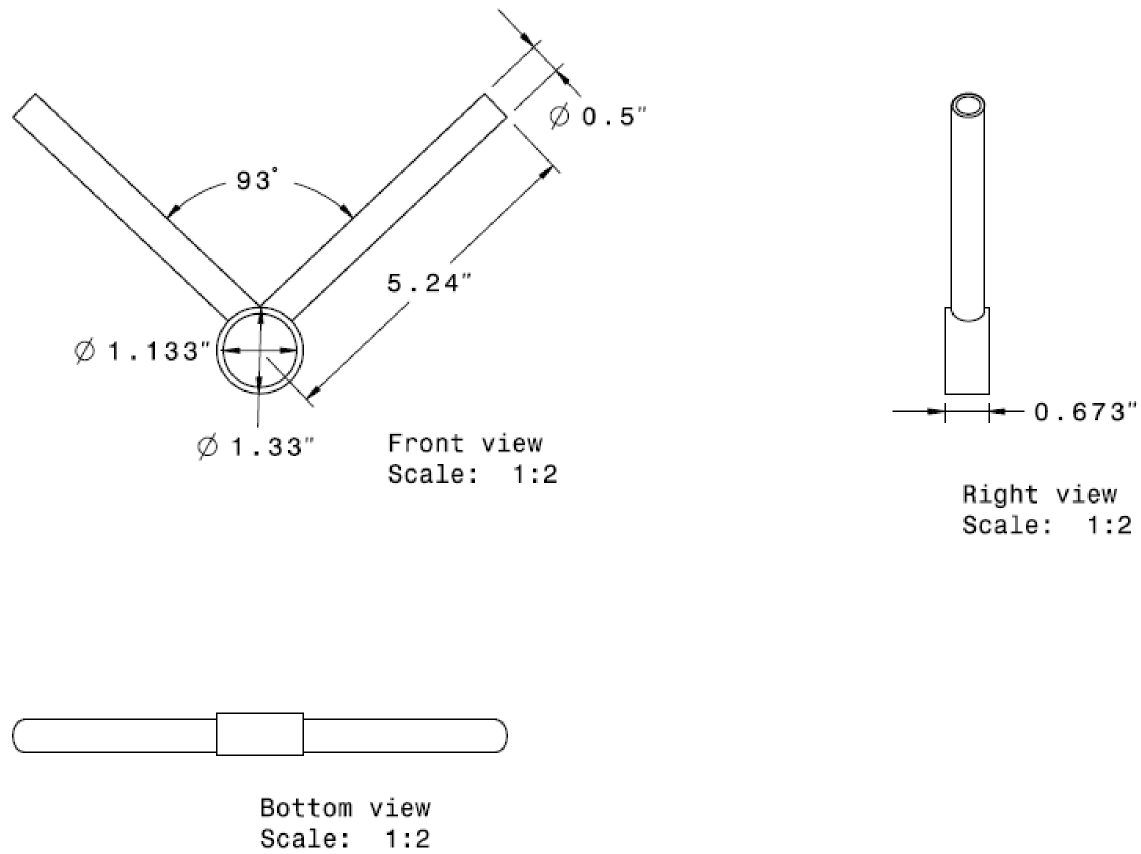
FSAE (2016). 2016 FSAE Hybrid Rules. Michigan USA: SAE International

Steven Fox (2010). Cockpit Control Forces

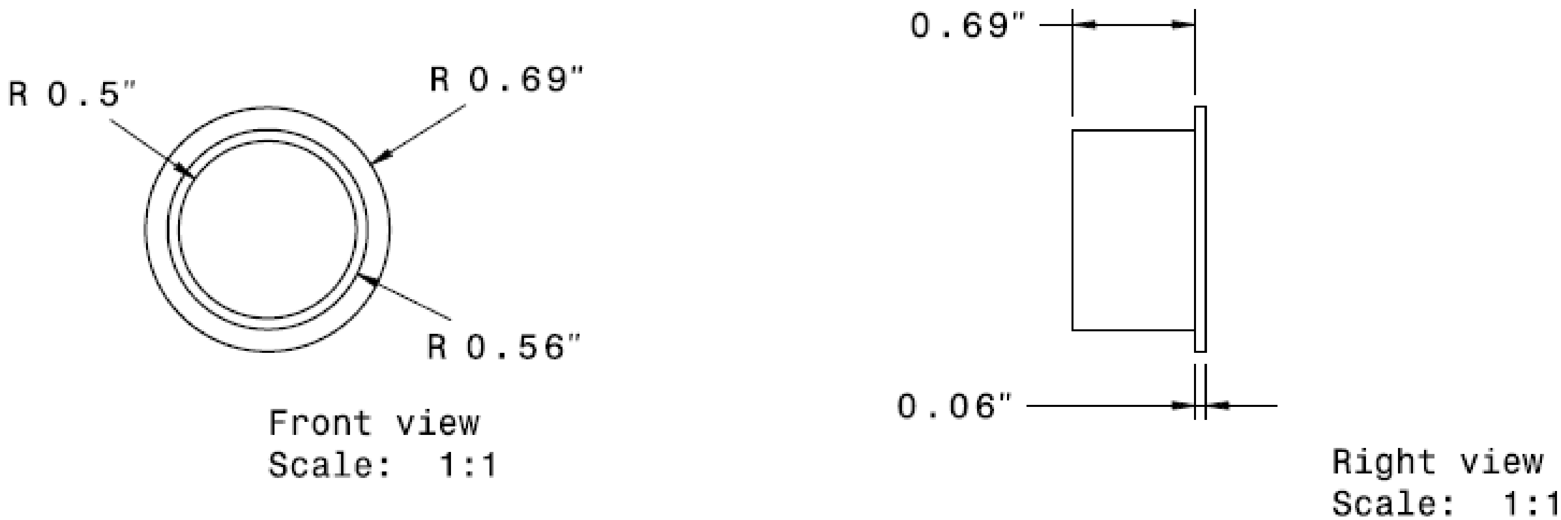
Carrol Smith (1978). Tune to win

ABAQUS User guide

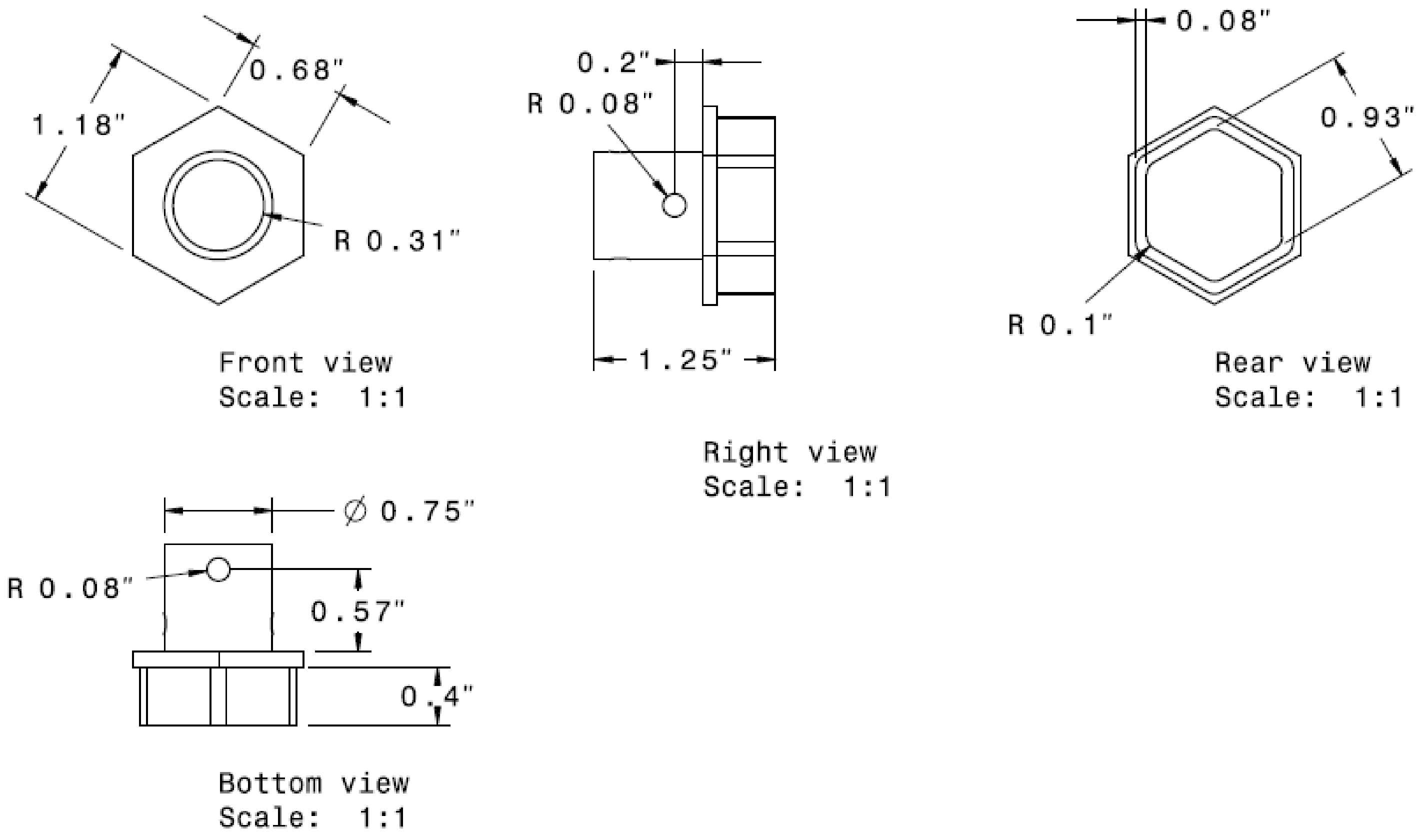
APPENDIX I: DRAWINGS



Drawing 1. Attachment

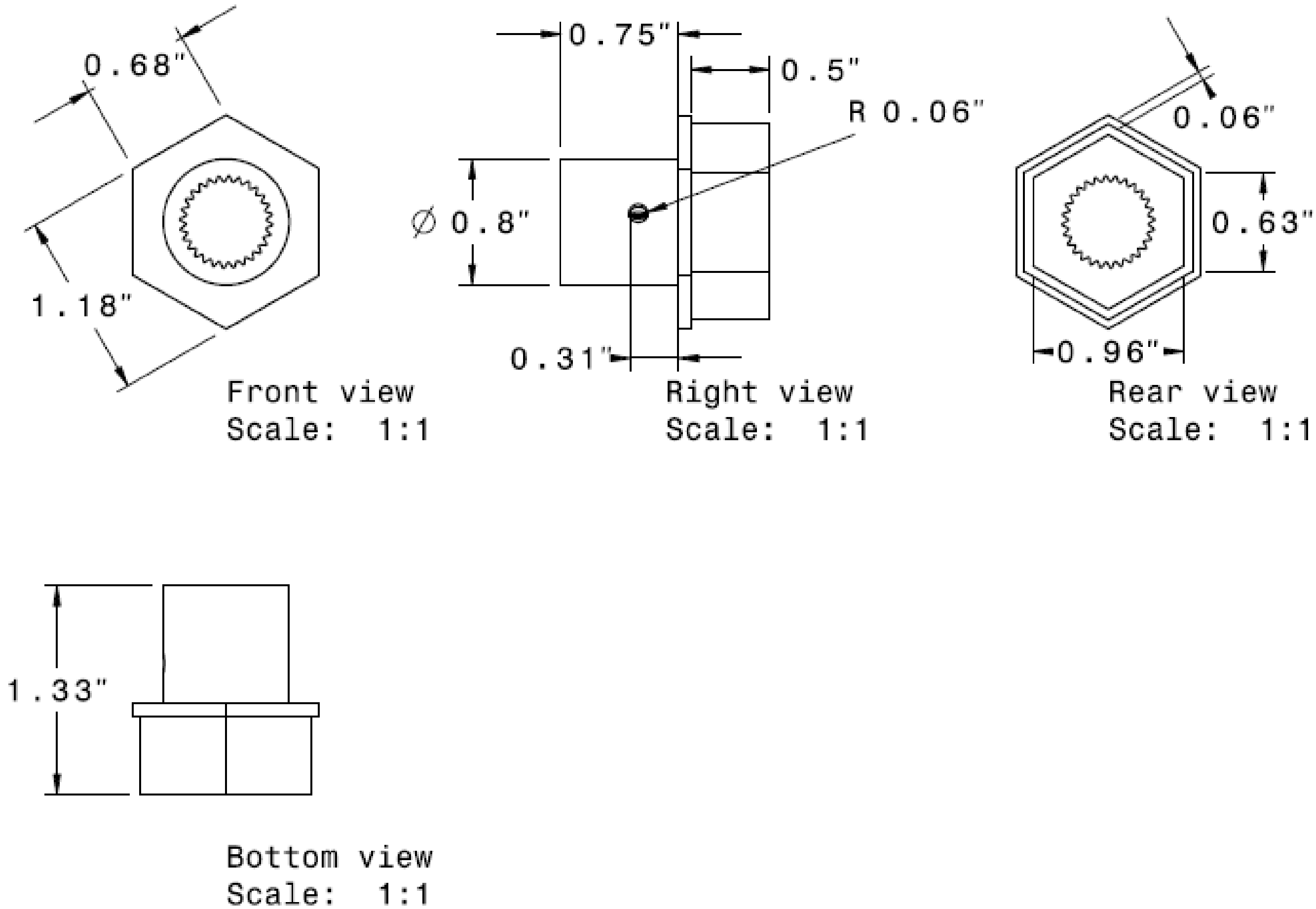


Drawing 2. Bushing

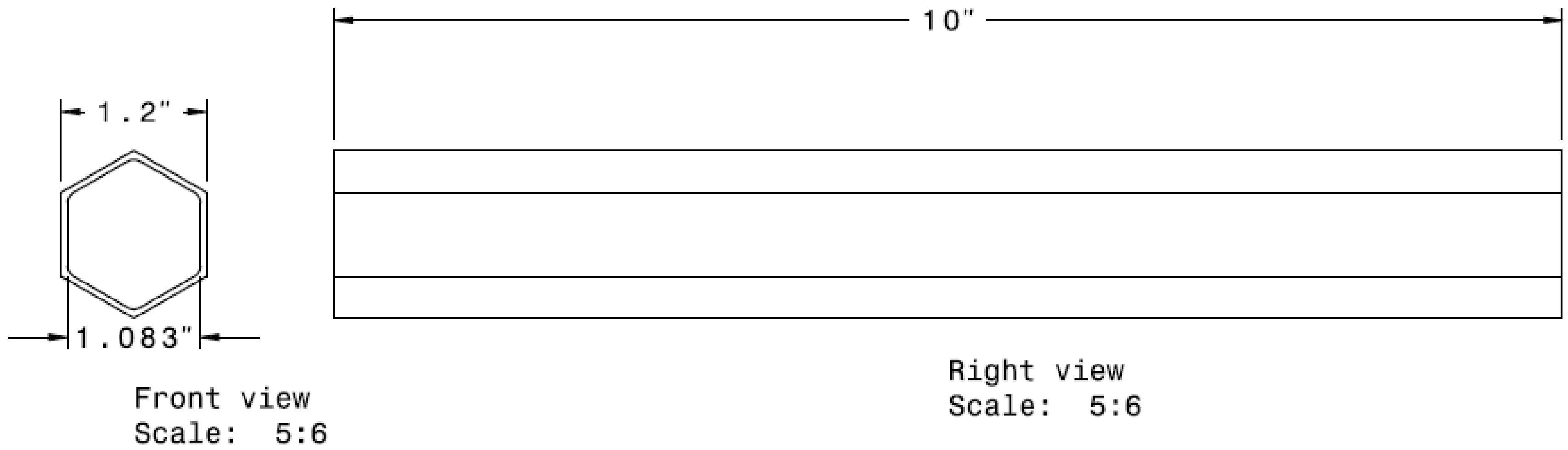


Drawing 3. Insert 1

ILLINOIS INSTITUTE OF TECHNOLOGY

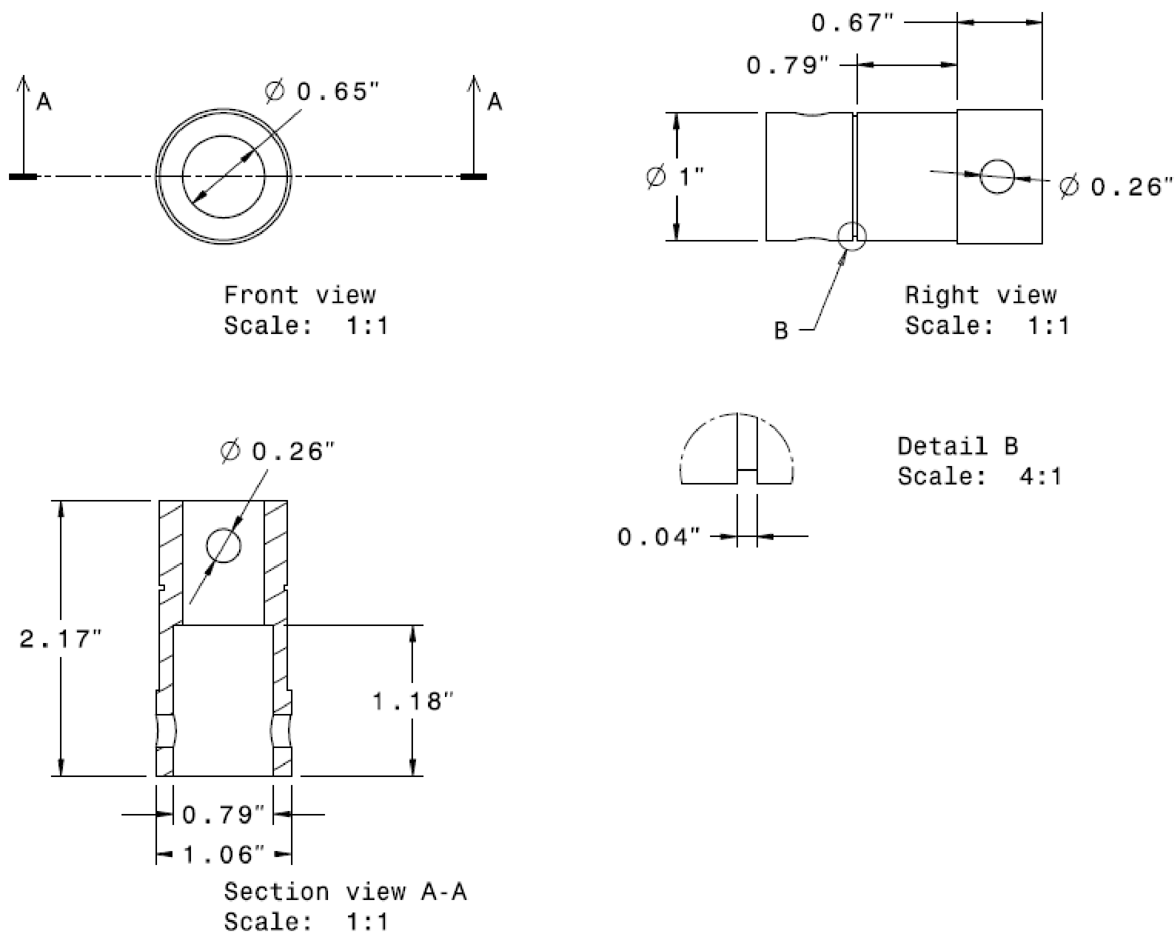


Drawing 4. Insert 2

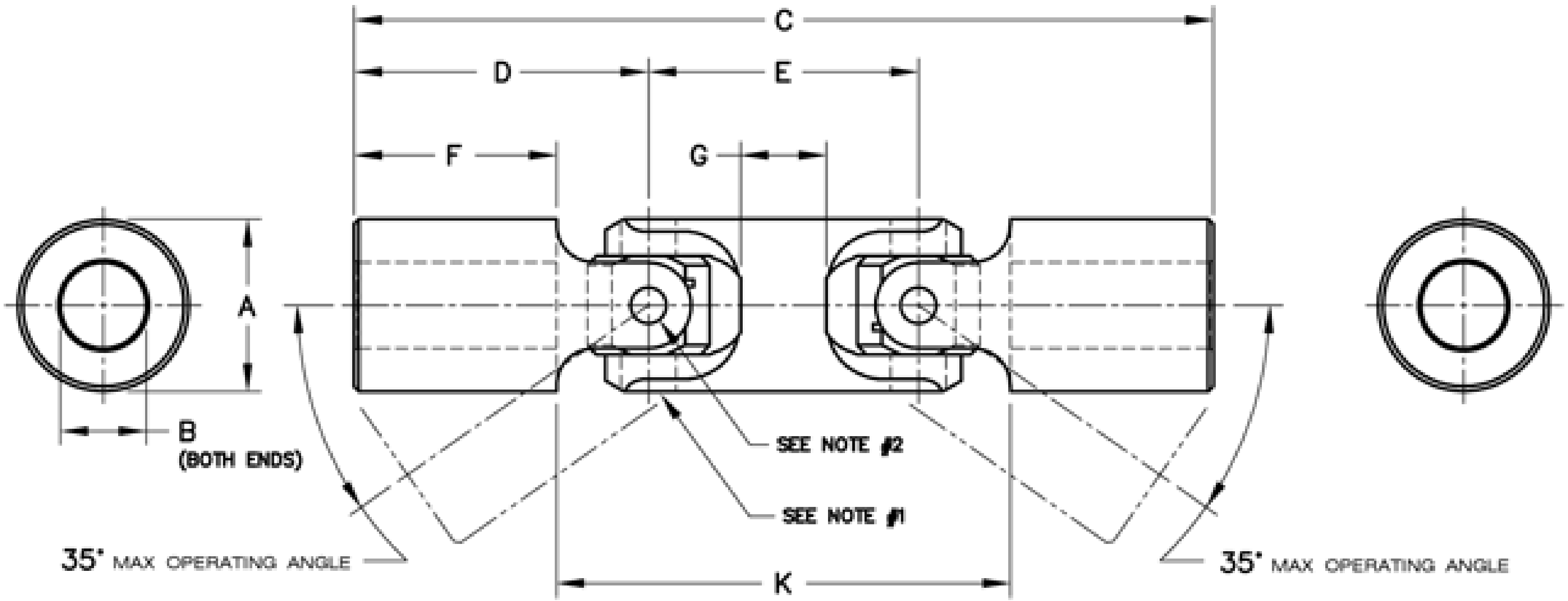


Drawing 5. Shaft

ILLINOIS INSTITUTE OF TECHNOLOGY



Drawing 6. Shaft-union



Drawing 7. Double Universal Joint

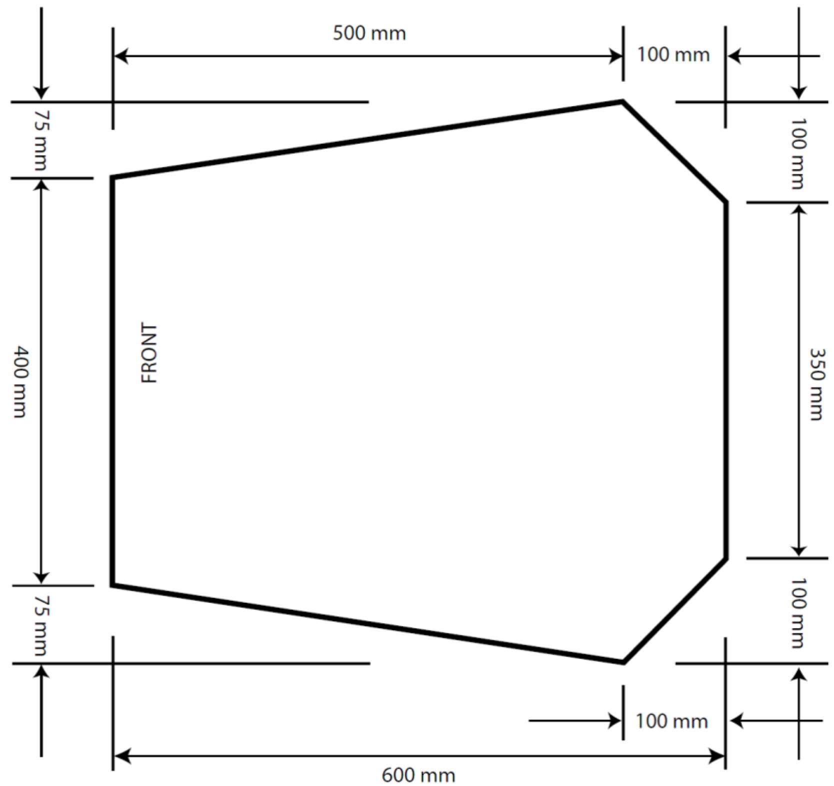
Catalog Number	SS643DB
Outside Diameter (A)	0.625± .020
Bore Diameter (B)	0.3125± .001
Total Length (C)	3.250± .030

ILLINOIS INSTITUTE OF TECHNOLOGY

End Fork Lengths (D)	1.125± .010
Centerline to Centerline Dimension (E)	1.000± .010
Hub Length (F)	0.813± .015
Center Section Hub Length (G)	0.375± .015
Shaft to Shaft Distance (K)	1.625
Static Torque Rating	900Inch Pounds
Approximate Weight	0.26Pounds

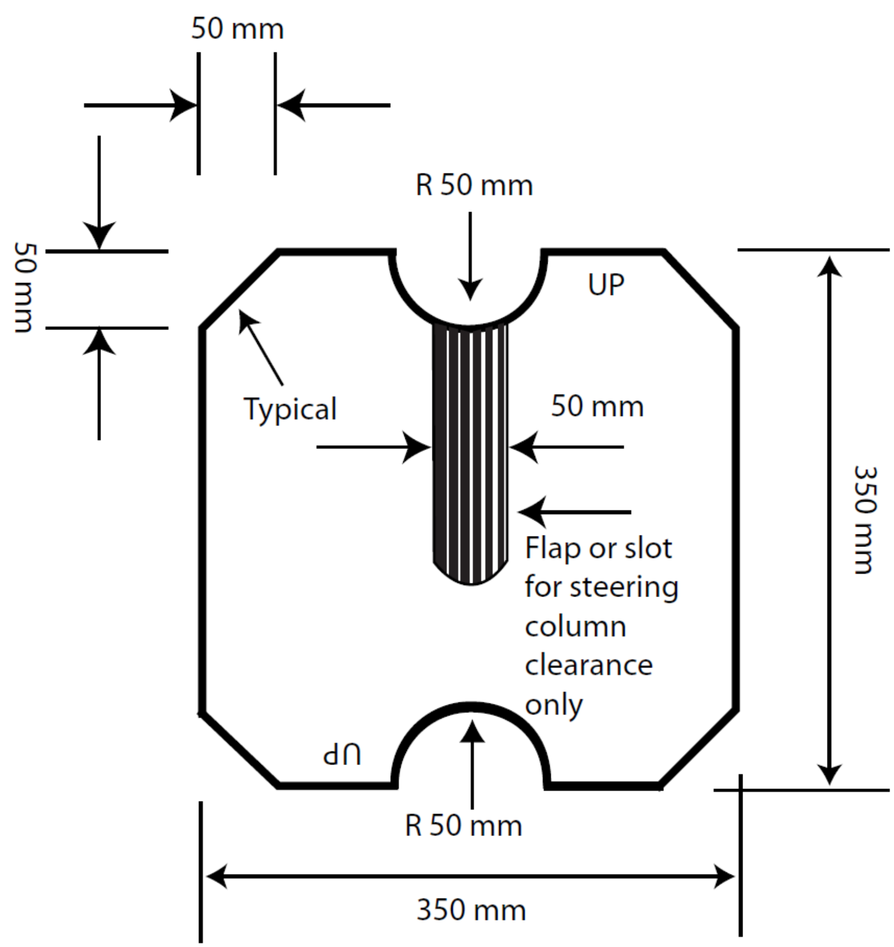
Table 8. Double Universal Joint measures

APPENDIX II: TEMPLATES



Drawing 8. Template 1

ILLINOIS INSTITUTE
OF TECHNOLOGY



Drawing 9. Template 2