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STEERING SYSTEM DESIGN OF THE SECOND GENERATION FORMULA SAE

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RESEARCH ARTICLE

ABSTRACT: The aim of this paper is to design the steering system of the formula racing car. This includes the designing of its main components in SolidWorks, its analysis by calculation and finite element simulation. Load acting on the wheels of formula car are calculated and input in the analysis wherever necessary. The cars steering rack is repositioned to avoid any collision with a-arms. The force acting on the bolts at tie rod, clevis and mounting bracket is found below its yield strength. The clevis attached at the end of the rack is subject to load and fatigue analysis in ANSYS and all results were found satisfactory. Similar analysis is done for rack arm at critical areas and its was found that region where the rack arm can withstand fluctuations ranging for almost whole life till failure.

Later the whole steering system was split into three major components (Steering column assembly, steering rack assembly, Tie rod assemblies) and each component is designed separately in SolidWorks and then assembled into whole one. SKF SAKAC 10 M ends are selected rod ends for steering arm assembly allowing rigid force transfer between rack arms and upright assembly as well as vertical motion of wheel assembly in operation. In the assembly 6 x M8 and 4 x M10 exist with the associated washers and nuts. The universal joint connecting the two steering column was machined according to Australian standards with splines at both ends. Two splined steering columns was machined in order to complete the steering column assembly. In order to stop the clevis colliding with the steering with the rack on full lock and to restrict the steering of the vehicle two locking collars were manufactured.

KEY WORDS: Steering assembly, rack and pinion, design, fatigue, ANSYS, SOLIDWORKS

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DIZAJN UPRAVLJAČKOG SISTEMA DRUGE GENERACIJE FORMULE SAE

REZIME: Cilj ovog rada je da osmisli sistem upravljanja vozila formule. To uključuje projektovanje njegovih glavnih komponenata u paketu SolidWorks, njegovu numeričku analizu i simulaciju metodom konačnih elemenata. Opterećenje koje deluje na točkove vozila formule se proračunava i unosi u analizu kad god je to potrebno. Letva upravljača vozila postavljena je tako da izbegne bilo kakav kontakt sa A vođicom. Sila koja deluje na zavrtanj na sponi, čauri i nosaču nalazi je niža od granice tečenja. Spona pričvršćena na kraj letve podvrgnuta je analizi opterećenja i zamora u ANSYS-u. Utvrđeno je da su svi rezultati zadovoljavaju postavljene kriterijume. Slična analiza je rađena za sponu u kritičnim presecima i rezultati su pokazali da u toj oblasti spona može podneti fluktuacije opterećenja tokom celog životnog ciklusa do otkaza.

Kasnije je ceo sistem upravljanja podeljen na tri glavne komponente (sklop stuba upravljača, sklop letve upravljača, sklopovi spona). Svaka komponenta je dizajnirana odvojeno u SolidWorks, a zatim su sastavljene u celinu. Zglobna SKF SAKAC 10 M je izabran za kraj upravljačke letve koji omogućuje kruti prenos sile između upravljačke letve i gornjeg sklopa, kao i vertikalno kretanje sklopa točkova u radu. U sklopu se nalazi još 6 x M8 i 4 x M10 sa pripadajućim navrtkama i podloškama. Zglob koji povezuje dva dela stuba upravljača je izrađen je prema australijskim standardima za navoje na oba kraja. Dva zaobljena dela stuba upravljača su dodatno obrađena. Kako bi se sprečilo zaključavanje dodata su dva elementa da spreče zaključavanje.

KLJUČNE REČI: Upravljački sklop, letva i zupčanik, dizajn, zamor, ANSYS, SOLIDWORKS

STEERING SYSTEM DESIGN OF THE SECOND GENERATION FORMULA SAE

Mobin Majeed, Greg Wheatle

INTRODUCTION

The purpose of this research paper is to design steering assembly of formula racing car. Aspects of the first generation car will be implemented into the second generation design with additional improvements to the vehicles maneuverability, handling and steering system safety. Steering system is being designed by carefully adhering to the rules of Formula SAE rule book and Australian standards. The steering system such as rack and pinion mechanism, steering wheel, steering column, tie rods as well as all necessary joints is fully designed in SOLIDWORKS and later analyzed in ANSYS. The major changes were to incorporate the current rack and pinion mechanism, move from its current position to avoid conflicts with driver's leg.

DESIGN APPROACH

The rules for the FSAE competition act as a constraint on the design of the steering system. According to the FSAE rule for Driver's leg and protection all moving components of the steering system including the steering rack and tie rods must be repositioned away from the driver's legs for driver safety[1]. All threaded fasteners utilized in the steering system must meet or exceed, SAE Grade 5, Metric Grade 8.8 and/or AN/MS specifications.

The Australian Standards that apply are listed in **Error! Reference source not found.** . Shaft sizes were determined using AS1402-2003. Standards 1665 and 1551 were referred to decide weld standards required to weld the mounting bracket with the frame of the car. Bolts used to mount the rack to the mounting bracket conform to product grades A and B in AS1110.1 (also listed in FSAE fastener regulations in the previous section). The mounting bracket and steering column of shaft were manufactured using 4140 choromoly with machinability specifications in accordance with AS1444-2007.

Table 1: Australian standard for design[2][3]

	Australian Standards
•	Australian Design Rule 10/02 — Steering Column
•	Australian Welding Code of Practice
•	AS1551-2011 - Welding of steel structures
•	AS1866 - Aluminium and Aluminium alloys
•	AS1665 - Welding of Aluminium structures
•	AS4024.1401 – Ergonomic Principles
•	AS1554.1 - Structural Steel Welding
•	AS1110.1 – Bolts
•	AS1444 – Wrought Allow Steels

The steering system was being designed parallel with other components of a car, so a discussion with other teams need to be had in order determine additional design constraints. Steering system tie rod connects directly to the upright to direct the front wheel, so the dimensions and solid works models of the upright geometry was needed to be obtained and analyze to understand the design constraints implemented by the upright team. Approximate dimensions for the JCUM driver was also needed to be attained in order to reposition the rack and pinion mechanism, steering column and tie rods without constricting the cockpit space of the driver[4][5].

DESIGN PROCESS

The design trend for the steering systems in competition remain relatively constant since long time[6]. All designs have a rack and pinion mechanism with complex packaging due to its high precision[3]. Most of the designed cited have their tie rod in line with the rack gear that reduces the bending stress[7]. A complete steering system design provided by our team leader is illustrated in the following **Figure 1**.

Figure 1: *a) Rack and pinion mechanism b) Formula car*

The second generation had mitigated the steering problems associated with the first generation formula car. The three major concepts are

- Minimizing overall effort of driver into the steering wheel[8]
- The driving space and steering system must be ergonomically feasible for the driver[9]
- Reduce stress on mounts and associated bushes
- Better Ackerman steering effect
- Smooth maneuverability

Three designs of steering wheel with rack and pinion connections are proposed. In the first design (**Figure 2**a) the universal joint is completely removed from the system for direct energy transfer. But the greater angled steering is ergonomically not feasible and cumbersome for driving. Second design (**Figure 2**b) consists of a single universal joint to improve high angled steering. The assembly is positioned to make two shaft lengths nearly equal for better distributing the forces. While the third design proposes two universal joint in the steering system (**Figure 2**c).

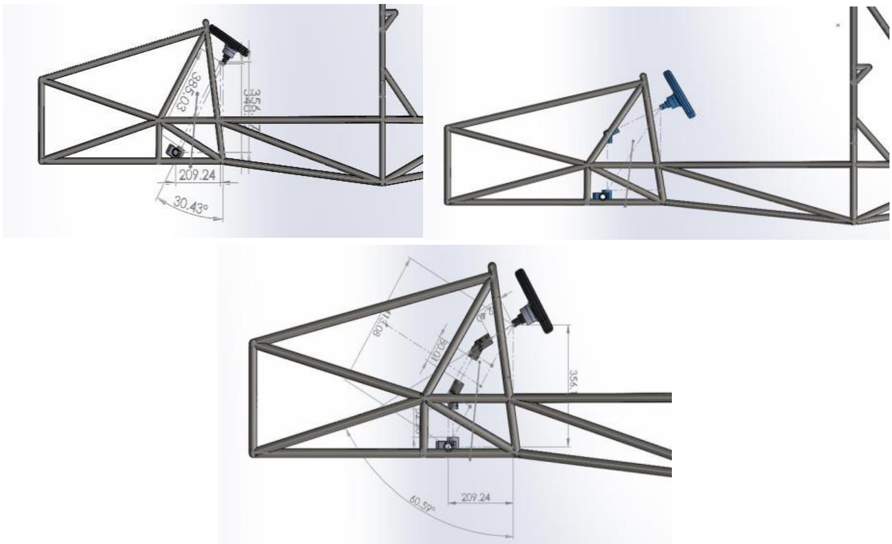

Figure 2:*a) System without universal joint b) with universal joint c) with two universal joint*

Furthermore, the preliminary design aims to position the steering rack such that the tie rods would run in line with the rack gear of the steering rack to the uprights[1]. But as the vehicle a-arms are also attached to the upright, this could result in collision especially in bumpy terrain[10]. So, the steering rack was repositioned further forward along the bottom strut of car frame as depicted in figure below (**Figure 3**).

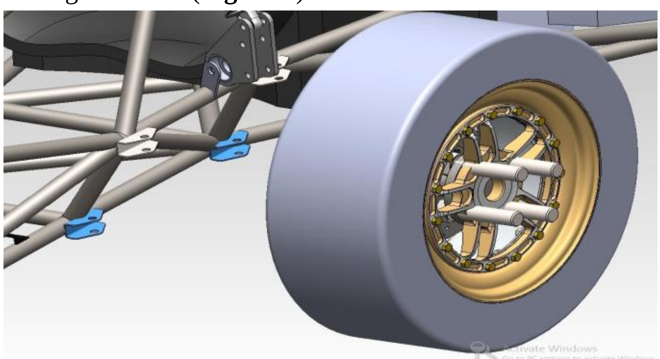

Figure 3: *Repositioning of steering rack*

DESIGN ANALYSIS

Analyzing and resolving the loads, it was found that three major force components were found into the y, x and z axis. The Y- axis force was being negated into the calculations as it is vertical force and assumed to be distributed. The z axis force of 783.7 N is approximated as 800 N for simplicity of analysis. Resolving the x axis and z axis force along the clevis steering and mounting bracket, the outcome is 800 N along horizontal and 461 N at an angle of 30^{o} as illustrated in **Figure 4**a below.

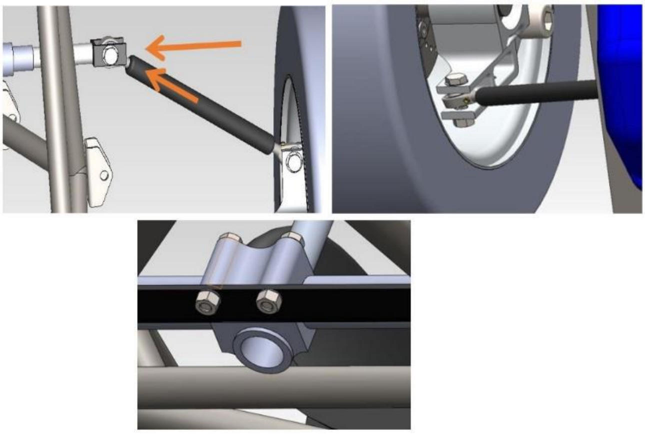

Figure 4: *a) Tie rod and mounting bracket b) M10 bolt c) M8 bolt at rack mounting bracket*

Bolts in the steering arms and rack assemblies were force transfer points, therefore they were subjected to these critical loads. It is found that M10 bolt (**Figure 4**a) in steering arm assembly was subjected to 923.76 N force and both M8 bolts at the rack mounting bracket had received half of this magnitude (figure 4c). However, this applied force is well below the strength of these bolts (22kN and 45kN for M8 and M10 bolts respectively).

FINITE ELEMENT ANALYSIS OF STEERING COMPONENTS

Two 5 mm fillet weld fix the rack mount to the 24.5 mm chromoly frame members. These welds were completed using TIG, argon shielding gas and ER80S-D2 filler metal. The situation is replicated in the ANSYS model. Two temporary 8 mm rods were loaded with the critical cornering load of y:461N and x:800N. Two solids at either end were created to simulate the 5 mm fillet weld. This is depicted in the **Figure 5** below. A mesh refinement of 3 was applied at the weld ends and bolting holes.

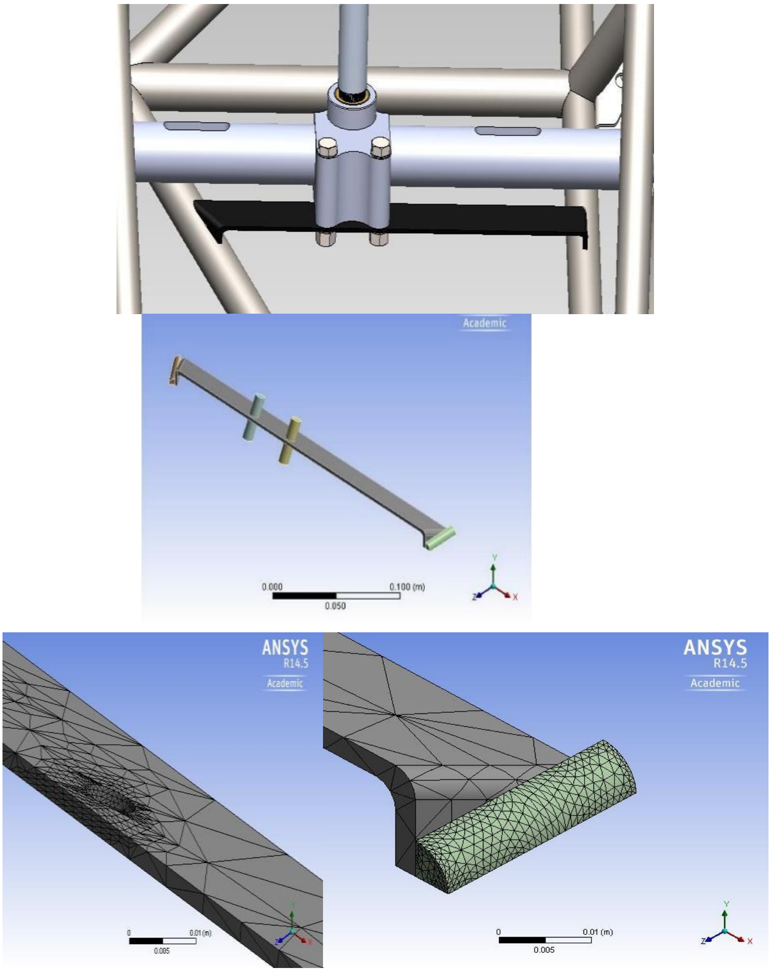

Figure 5:

Figure 5: *Rack mounting bracket welds SOLIDWORKS and ANSYS model*

Hand calculations have estimated a von misses stress equivalent of 5.586 M Pa. However, from the ANSYS analysis it was found that the stress range is between 29.55KPa and 69.97MPa (**Figure 6**). These stresses are well below the yield strength of the ER80S-D2 filler metal at 492MPa. The maximum stress location occurred in the bolt holes with a stress of 314.77MPa. Again, this is within the limits of the yield strength of 4130 chromoly steel. These results imply the selected welds are suitable for fixing the mounting bracket to the frame.

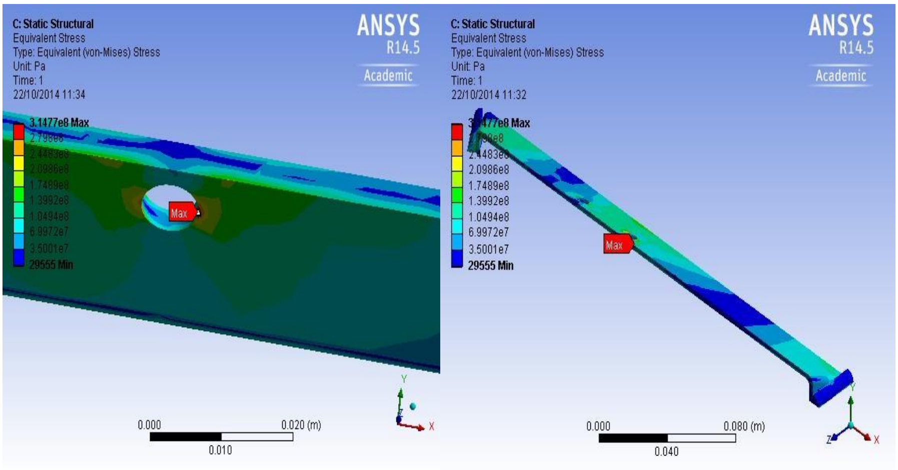

Figure 6: *ANSYS simulation analysis of mounting bracket.*

RACK ARM CLEVIS ANALYSIS

The resultant of x = 800N and Y=461N is 931 N acting on the temporary pin of clevis pin hole with the clevis is being constrained with a fixed support. The clevis design consisted of two 10 mm pin holes with the pin hole plates having a thickness of 3 mm and a 20 mm geometry depth. Its ANSYS analysis is illustrated in **Figure 7** below. The mesh is being refined by a factor of 3 at the critical zones.

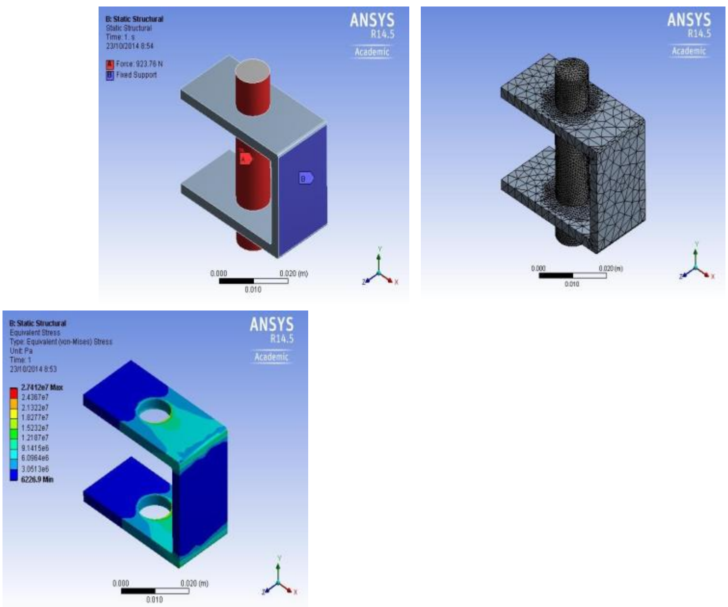

Figure 7: *a) Load and resistance case of clevis (left) b) Mesh case of clevis(middle) c) Static structure analysis(right)*

The result of the equivalent von misses stress analysis of clevis with a 931.76 N load is shown in **Figure 7**c.

Additionally, a fatigue analysis was completed for the clevis using an endurance modifying factor of 0.571. A fully reversed load was assumed and the Goodman stress theory was used as the fatigue criteria. These conditions were used for all fatigue analysis done on the clevis. An image of the life and safety factor for the clevis are shown below is **Figure 8**a and **Figure 8**b.

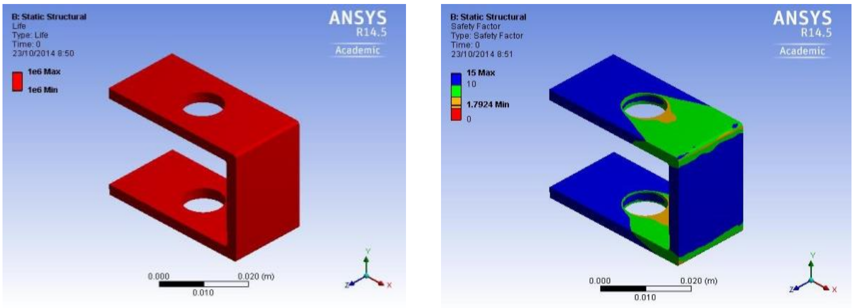

Figure 8: *a) Resultant life plot of clevis(left) b) Resultant safety factor plot of clevis(right)*

A review of the compiled data revealed that the safety factor was in excess of 1 for the entirety of the clevis with a minimum fatigue safety factor of 1.79. Hand calculations reinforce ANSYS data as a fatigue safety factor of 3.48 was calculated for the clevis pin hole. The life plot confirms that the clevis is rated for infinite life under the current loading conditions. As such, the clevis when loaded with the worst possible load case with the contact patch being located in line with the rack arm under cornering loads is rated for infinite life.

RACK ARM ANALYSIS

The steering rack was analyzed as a whole due to the complex geometry of the steering rack and mesh refinement is only applied to areas of interests. The main area of interest for the steering rack arm was the region where the fully extended rack arm meets the steering rack housing and a mesh refinement of 3 was applied in this region as shown in **Figure 9**a.

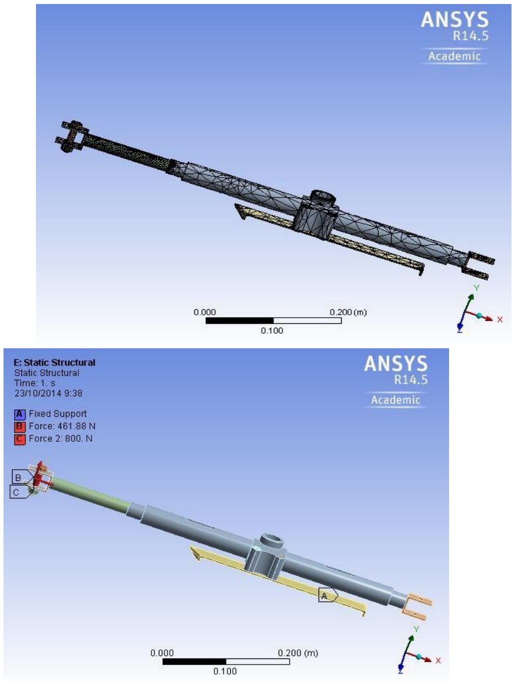

Figure 9: *Steering rack a) Meshing(left) b) Load and support setup of rack arm(right)*

Similar to the rack arm clevis, the rack arm was analyzed during the worst possible case when the contact patch is in line with the rack arm under corning loads. As such, the clevis bolt on the fully extended rack arm was loaded and the bracket was used to secure the assembly using the fixed support as depicted in **Figure 9**b. The contact region between bolt and washer found to have a maximum von-misses stress of 459 MPa illustrated in **Figure 10**.

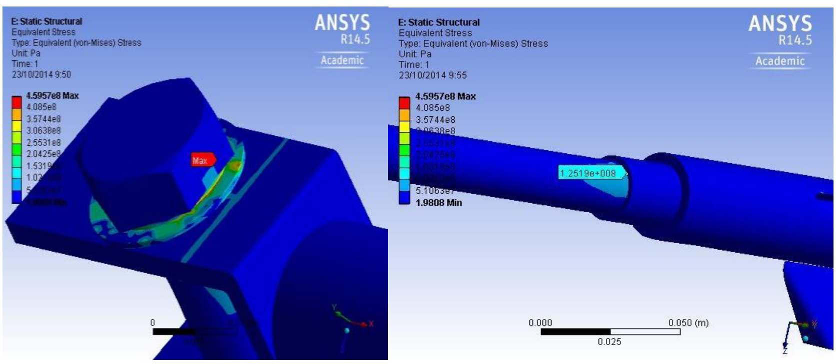

Figure 10: *Maximum von- misses stress at contact regions*

The equivalent stress at region where the fully extended rack arm meets the steering rack housing have found to have a stress of 125 MPa (**Figure 11**) in comparison of 118 MPa calculated by hand.

Rack arm is also subjected to fully reversed fatigue analysis with the Goodman safety theory as a safety criterion and endurance modifying factor of 0.651. **Figure 11** illustrates the infinite life behavior of rack arm.

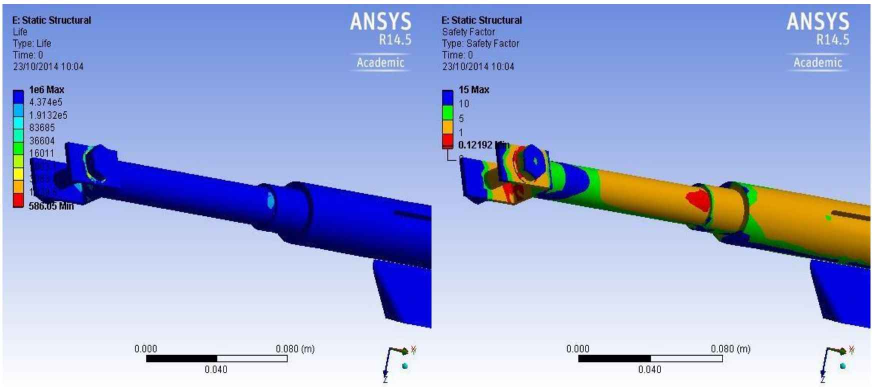

Figure 11: *a) Infinite life plot(left) b) safety factor plot(right)*

As shown in **Figure 11** a, region where the rack arm meets the rack housing can withstand fluctuations ranging 80, 000 to 90, 000 till failure. However, it is highly unlikely that it is subjected to this much fluctuations in its full lifetime.

FINAL DESIGN

The final design can be broken in to three main design components, and is typical to most kart style motor sport steering system.

- Steering column assembly
- Steering rack assembly
- Tie rod assemblies

These assemblies link together to form the final steering system. This is illustrated in the **Figure 12** below

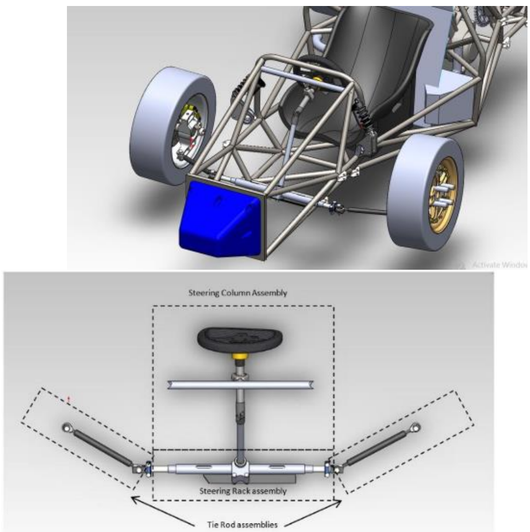

Figure 12:*a) Steering system isometric view(left) b) Steering system main component breakdown(right)*

The design is a standard universal joint, rack and pin steering system where by rotational forces are turned into linear motion from a pinion gear and rack. These forces are then transferred from the tie rods into wheel uprights where turning begins. The steering column is simply supported by a rotational rod end under the front hoop. The column is then spline attached into the rack which is supported at an angle of 21.6^o. A clevis at either end links the tie rods through 10mm rods ends. The following **Figure 13** illustrates this final design.

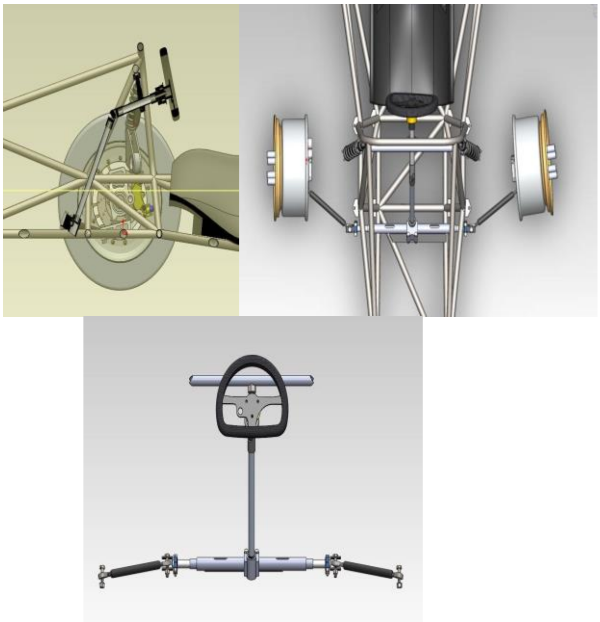

Figure 13: *Steering assembly a) Mid planar view (left) b) Top view with wheel assembly (middle) c) Drivers view(right)*

ASSEMBLY PARTS OF STEERING SYSTEM

SKF SAKAC 10 M ends are selected as rod ends for the steering arm assembly, allowing rigid force transfer between rack arms and upright assembly as well as vertical motion of wheel assembly in operation. In addition, there are attached grease nipples for ease of lubrication(**Figure 14**). In this build a fine 1.25mm pitch thread was selected where the minimum cross sectional area was larger at 61.22mm^2.

Figure 14: *Rod ends a) SKF 10 mm(left) b) SKF 25 mm(right)*

Similarly, an SKF SAKAC 25M rod end was selected as the steering column support permitting a rigid x and y axis platform for the 25mm diameter 162mm steering column to rotate about. Again this rod end uses a 1.25mm fine pitch thread with the thread diameter of 16mm (M16).

The designed rack a 360^o rotation of pinion corresponds to 35 mm x displacement of the rack arm. The rack is fabricated with machined aluminum, having arms of 20 mm solid alloy rod and clevises are bolted flush to the ends (**Figure 15**). The rack is mounted using two Class 8.8 M8 Bolts and 4 X 8mm mild steel washers. The splined pinion insert is ¾ inch 19mm 48 splined.

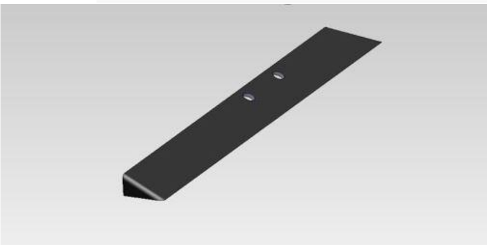

Figure 15: *a) Rack and pinion mechanism(left) b) Mounting bracket(right)*

The mounting bracket (**Figure 15**) for the steering rack was manufactured at JCU Engineering workshop with 3mm plate of 4130 Chromoly steel. The bracket is being welded onto the bottom frame using two 5 mm welds along the outside of bracket contacts. The rack is mounted on the brackets using two class 8.8 M8 bolts and 8mm washers.

M 8 AND M 10 BOLTS AND WASHERS

All bolts in the steering assembly are class 8.8 as per FSAE rules. In the assembly 6 x M8 and 4 x M10 exist with the associated washers and nuts. As per rules a minimum of two threads protrude from the nut trailing end. Information on these bolts and location is tabulated(***Table 1***) and shown(**Figure 16**)below.

Table 1: *Information on bolts*

Nominal Size	Pitch mm	Stress Area mm^2	Min. Tensile Stress MPa	Ult. tensile load kN	Min. Breaking Shear load kN
M8	1.25	36.60	800	29.28	22
M10	1.5	58.00	800	46.60	45

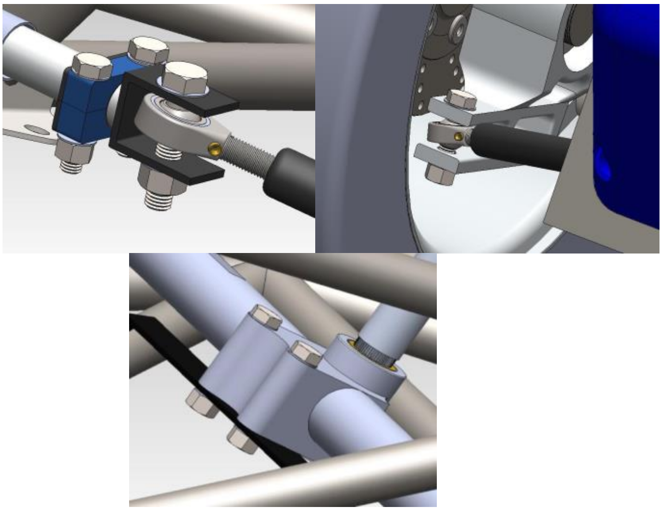

Figure 16: *Location of bolts*

The chosen universal is a ‘no slop’ compact unit with an operating angle operating angle of 35 degrees, which is the angle at which the assembly has been designed for (**Figure 17**a). This blank universal was machined with the following properties from PMD racing products Australia – ¾” – 48 spline x ¾” -48 spline at respective ends. This is having an outer diameter of 28mm and Inner major diameter 19mm with a 25 mm bore of each internal spline.

Figure 17: *a) Universal joint(left) b) 325 mm steering column(right) c) 162 mm steering column(middle)*

Two splined steering columns was machined in order to complete the steering column assembly (**Figure 17**a & b). A 162mm length column and 325mm column was machined from chromyl with a ¾” 48 spline 9mm splined end with a spline length of 25mm. The diameter of these shafts is 25mm with an inside diameter of 18mm.

LOCKING COLLARS

In order to stop the clevis colliding with the steering with the rack on full lock and to restrict the steering of the vehicle two locking collars were manufactured. The design consisted of two 12mm wide top and bottom pieces which when assembled allowed a 20mm rod to be inserted. This is where the rack arm will connect. Two M8 bolts were holding the two parts together and provided locking forces to the rack arm. To reduce any possible damage to the arm, a thin 1mm copper sleeve was inserted into the hole made from copper sheeting. This had also distribute clamping forces. An assembled and exploded view of the locking collars is depicted in **Figure 18** below

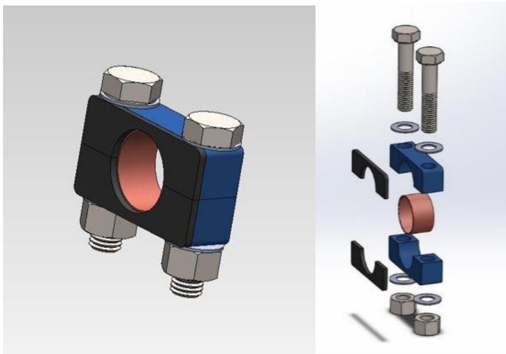

***Figure 18**: Assembled(left) and exploded view(right) of locking collars*

TECHNICAL SPECIFICATIONS

Ultimately the final design had to meet the requirements of the initial design constraints and perform these actions within the FSAE rules. The top plane view below illustrates some of the geometrical relations between the steering rack and tie rod assemblies(**Figure 19**).

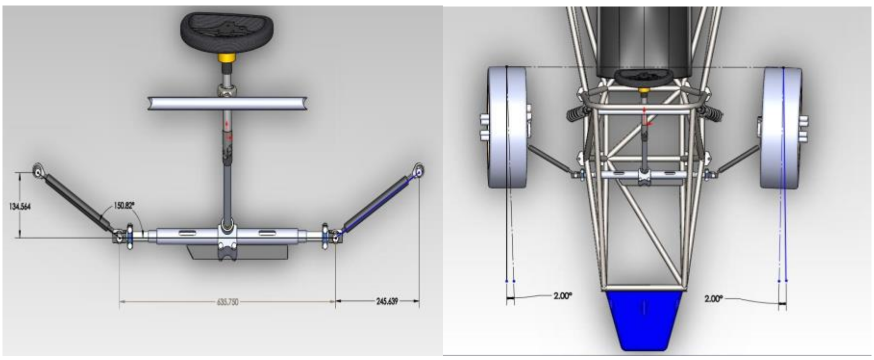

***Figure 19:** a) Top view geometric relations of steering system(left) b) Toe in reference dimensions(right)*

Note that this shows the geometric relations when the rack is in a neutral position, with the Toe. The current configuration has a U-joint operating angle of 34.5^{o}. Which is in the range of the joint specification. Furthermore, the steering column support bar has been designed so that minor corrections can be made to this angle. By simply removing the 162mm steering column and rotating the support rod end, the steering wheel can shift 12mm vertically, relieving the angle further.

ERGONOMIC FACTORS

- Leg clearance from frame base = 54.710mm
- Steering wheel angle approx. = 16^{o}
- Universal Joint clearance = 335.8 mm
- Column support clearance = 368.7mm

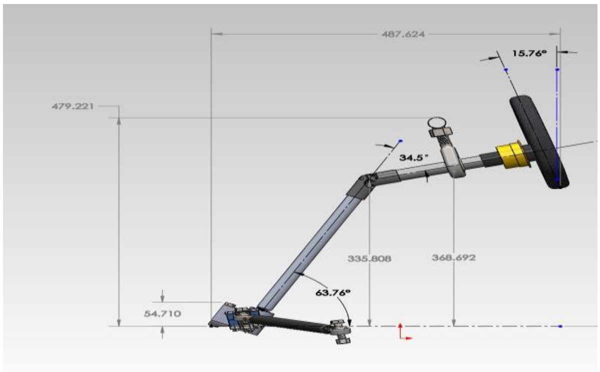

Figure 20: *Side view geometric relations*

Leaving the universal joint angle at 34.5^{o} (shown in **Figure 20**) was desired as suppressing the angle further would require shifting the rack further forward in the frame. Doing so would reduce the space allowed for the complex foot pedal/ control assemblies that are required in the vehicle.

CONCLUSION

Using the proposed constraints, an appropriate steering system was devised as an improvement in regards to the previous generation. The universal joint operating angle was relieved to a suitable degree as specified in the chosen joint. A major design component was to mitigate the lag in steering induced by the previous locking pin system in the steering column assembly. This was investigated, and appropriate splined components were designed where necessary, however the inclusion of splined parts incurs higher manufacturing costs.

FEA analysis concluded that all critical components are well within operating constraints of the critical load case proposed under fatigue, and static analysis. The final design was commended with an appropriate turning radius, using the existing rack. Collaboration with the suspension, uprights, A-arm/wishbone assembly teams is needed to effectively coordinate a solution.

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