Design & Fabrication of Suspension System of a Formula Student Race-Car

Ishan Vasoya[1] Akshay Patel[2] Akshit Vithlani[3] Chirag Valani[4]
[1,2,3,4]Student
[1,2,3,4]Department of Mechanical Engineering
[1,2,3,4]Chandubhai S. Patel Institute of Technology, Charusat University, Changa, Gujarat, India – 388120

Abstract— Vehicle dynamics is a widely researched and an important subject in the field of automotive and mechanical engineering. Especially in case of designing a race car, it plays a quintessential role. So, this paper is a study about how the mechanics behind the excellent performance of a race-car actually works. Then after the procedure of designing and manufacturing of a race-car suspension system is elaborated step by step.
Key words: Race-Car, Suspension System

I. Introduction

Suspension system is the term given to the system of springs, shock absorbers and linkages that connect a vehicle to its wheels. When a tire hits an obstruction, there is a reaction force and the suspension system tries to reduce this force. The size of this reaction force depends on the unsprung mass at each wheel assembly. In general, larger the ratio of sprung weight to unsprung weight, the less bumps, dips, and other surface imperfections affect the body and vehicle occupants such as small bridges. A large sprung weight to unsprung weight ratio can also impact vehicle control.
The role of suspension system is as follows:

– It supports the weight of the vehicle.
– Provides a smoother ride for the passengers.
– Protects the vehicle from damage.
– Keeps the wheels firmly pressed to the ground for better traction.
– It isolates the vehicle from road shocks.

II. Design

While designing suspension geometry we considered many suspension systems and a-arm designs and explored their pros and cons. First option was to use Trailing arm, no doubt it offers some benefits but it couldn't convenes us enough to use it considering the drawbacks: weight of trailing arm is more thus increasing unsprung to sprung mass ratio which is not desirable, complexity in adjusting suspension parameters, packaging and fabrication problems.

So we decided to use Independent Double Wishbone Suspension system. Independent system allows independent vertical movement of each wheel on same axle. Ease of drive, manufacturability, smooth handling during all acceleration conditions is what we kept in consideration. To provide the smooth riding as well as handling easiness, the Independent system was found to be good compared to other. Further classifying the system selected is Fully Independent. The review and research led us to select the Double Wishbone (A-Arm) Suspension system. Double wishbone system allows great control over vehicle dynamic parameters like Camber change rate, castor angle, scrub radius, steering axis inclination and others. It offers better ride quality and handling characteristics due to lower unsprung weight and the ability to address each wheel individually. The maintenance of Double wishbone is easy and more efficient than Trailing arm cause if load acting on a-arm exceeds the capacity in Trailing arm case it may cause damage to drive shaft also leading to expensive maintenance on other hand in case of Double Wishbone the tubes of a-arm will get damaged which are easy and economical to replace than drive shaft

A. Design Constrains

The main specification of the suspension system as required by the rules is that the car must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of 25.4 mm in jounce and 25.4 mm in rebound, with driver seated.

The suspension team chose the SLA suspension. The A-arm configuration meets the design requirements for structural integrity and suspension travel. The suspension arms are supported by a push rod assembly that resists the motion associated with the suspension travel, and supports the weight of the car in its neutral position. The rocker is affixed to the chassis, and transfers the force in the push rod to a shock absorber and spring assembly that both dampens, and supports the travel of the suspension system, and to a pushrod. The Push-Rod system enables us to have restrictive wheel movement and helps to keep the car at road level at desired clearance.

B. Preliminary Design of Parameters including Event Rules Wheelbase

1) Rule T2.3

"The car must have a wheelbase of at least 1525mm" [1]. The shortest wheelbase possible would be desirable due to the increased maneuverability it instills on a tight track like that experienced on autocross and endurance circuits at FSAE competition. So, a slightly longer wheelbase of 1573 mm was selected to introduce a margin for later design compatibility. The wheelbase affects load transfer in the longitudinal direction; however, it has a small effect on the kinematics of the suspension.

2) Vehicle Track

Rule T2.4: "The smaller track of the vehicle (front or rear) must be no less than 75% of the larger track" [1]. The front track width is generally larger than that of the rear, for a rear wheel drive racecar in order to increase rear traction during cornering exit by reducing the amount of body roll resisted by the rear tires relative to the front tire. A rear track width of 45inches was selected as a compromise between the benefits of reduced weight transfer from a wider track and the tighter travel path about a chicane a narrower track allows. Due to larger track width, there will be chance of buckle of A-arms. So, Ojaswat 17 car has a track width of 47inches at front and 45inches at rear.

3) Ride Height

The vehicle must be equipped with fully operational front and rear suspension systems including absorbers and a usable wheel travel of at least 50mm with driver seated (25mm jounce and 25mm rebound). An initial ride height of 45mm was chosen to provide sufficient ground clearance and prevent the bottom of the chassis from hitting the ground under full bump and maximum braking. Such lower ride height will be beneficial to keep the center of gravity of the car lower.

4) Anti-Pitch Geometry

No anti geometry was used in Ojaswat 16 due to compliances caused in side view geometry of arrangement and location of control arm pick-up points on chassis. 25% anti-dive geometry is chosen for Ojaswat 17 to control the pitching movement of the car during braking and accelerating.

5) Swing Arm Length

As the swing arm length is the largest determinant of the camber curve variation of this distance would be used to attain the desired camber gain. Based on the recommendations identified in [3], a medium FVSA length (1000mm –1800mm) was chosen as the best compromise of minimizing roll center movement, getting the required camber gain and limiting scrub radius. A longer swing arm at the front (relative to the rear) to reduce camber gain as castor geometry on the steering would introduce extra negative camber on the outside wheel. Swing arm lengths of 1600mm on the front and 1200mm on the rear were finally selected for the kinematic design.

6) King Pin Inclination

Positive kingpin angle causes the outside tire to take on positive camber when the front wheels are steered which is highly undesirable [2]. This undesirable effect can be countered by the negative camber gained during steering from castor angle. Also to reduce the scrub radius and overall trail so as to provide an easy steering, a positive KPI of 3.88^0 is adjusted.

7) Roll Centre Height

There were three options to choose the location of roll center. First one was below ground level second was on the ground level third was above the ground level. Each option had different type of effect on rolling behavior of vehicle.

If roll center were to be below the ground, the geometry would result in excessive rolling moment accompanied by jacking forces as jacking moment and rolling moment would be in same direction. This would result in less rolling stability and drivability.

If roll center were to be on ground, the jacking force would be zero and vehicle would be under rolling moment only but if the roll center would be above ground then jacking moment and rolling moment would be opposite meaning that jacking moment would negotiate effect of rolling moment up to certain extent depending up on roll center height. But if roll center were to be too high above ground then a large distance between ground and roll center would lead to high jacking forces.

So, for maximum rolling stability a roll center height somewhere between Center of Gravity and ground had to be selected. After many simulations a range of 40mm to 60mm seemed good.

Now, second part was to decide roll axis inclination. As the car is rear drive, it would have some intensive over-steering characteristics. So, to counter them, we decided to keep rear roll center higher than front roll center. Also, we kept in mind that a larger difference in front and rear roll center height would result un-drivable characteristics. Hence, 45mm roll center height was selected for front suspension and 60mm roll center height was selected for rear suspension.

C. Wheel Alignment

1) Camber

In this system, the upper control arm is kept shorter for inducing negative camber angle during the wheel jounce. Camber angle will control the handling characteristics of suspension system. Negative camber will help to maintain good tire contact patch while cornering. Also, maximum acceleration along a straight line is obtained when the camber angle is zero. Too much camber angle leads to bad handling. With the application of SLA (Short Long Arm) system the camber angle can be adjusted. Ojaswat 17 car has a static camber of -1^0 at front and -1^0 at rear.

2) Castor

Castor angle provided is used to optimize the steering characteristics during cornering. Proper Castor angle helps to achieve self-centering during the left or right steering conditions. Positive castor angle will help to give directional stability as well as provide easier driving. Ojaswat 17 car has a front castor of $+5^0$.

3) Toe

Toe is given to car wheels so that they remain straight forward when in straight driving conditions. As the car is rear wheel drive the front wheels are kept at toe in conditions so that during the running condition the wheels will turn straight providing maximum contact patch. This will aid to the straight driving stability. The required toe angle is a compromise between the rolling resistance and straight wheel stability. The toe can be adjusted per the requirement. Ojaswat 17 car has a static toe $+1^0$ at front and 0^0 at rear.

D. Design Data & Calculations

Parameter	Value
Weight of whole car with driver	300 kg
Weight of car without driver	230 kg
Roll gradient	1.2°/g
Roll Center Height for front(RCH)	45mm
Roll Center Height for rear(RCH)	60 mm
Horizontal distance of C.G from front(a)	867.15mm
Horizontal distance of C.G from rear(b)	707.85mm
Wheelbase	1573 mm
C.G. height	230mm
Total sprung weight	235 kg
Motion ratio	0.8
Track width for front	47’’
Track width for rear	45’’
Ride frequency front (W_F)	2.63 H_Z
Swing arm length (front)	1600mm
Swing arm length (rear)	1200mm
Ride frequency rear (W_R)	2.36 H_Z

Table 1: Vehicle Data

1) Calculations [2]

Hs is the vertical height of CoG of sprung weight. Hrm is the vertical distance between the CoG of sprung weight and the roll axis.

Zf and Zr are roll center height at front and rear respectively.

Hs is determined by Equation-

$$Hs = \frac{Wtotal*h - Wunspf*RLF - Wunspr*RLR}{Ws}$$

So, Hs = 231.1 mm

Also, $Hrm = Hs - (Zf + (Zr - Zf) * (1 - as))$

as = sprung mass weight distribution $\left(\frac{Wsprf}{Wspr}\right)$

So, Hrm = 177.8 mm

Required rolling stiffness of system for no anti-roll bar case,

$$Køreq = \frac{Wspr*Hrm}{R.G.}$$

$$= \frac{235*9.8*0.1778}{1.2}$$

= 341.22 Nm/deg roll

So, taking Køreq= 340 Nm/deg roll

Køf + Kør = 340

Taking Køf = x therefor Kør = 340 – x,

Calculating ride rate,

$$Krf = \frac{Køf*360}{Tf2*\pi} = 80.406* x \qquad (1)$$

$$Krr = \frac{Kør*360}{Tr2*\pi} = 87.712(340\text{-}x) \qquad (2)$$

Also,

$$Krf = 4\pi^2 * \omega f^2 * \left(\frac{Wsprf}{2}\right)$$

$$\text{So, } \omega f = 0.19696\sqrt{x}\ \text{Hz} \qquad (3)$$

$$Krr = 4\pi^2 * \omega r^2 * \left(\frac{Wsprr}{2}\right)$$

$$\text{So, } \omega r = \sqrt{(11.62154 - 0.034181x)}\ \text{Hz} \qquad (4)$$

We want to keep the difference between front and rear ride frequency about 10%. Hence,

ωr = 0.9 ωf

$$0.17739\sqrt{x} = \sqrt{(11.62154 - 0.034181x)}$$

x = Køf = 177.1468 Nm/deg

Kør = 162.8532 Nm/deg

From (3) and (4)

ωf = 2.62 Hz, ωr = 2.36 Hz

From (1) and (2),

Krf = 14243.664 N/m

Krr = 14284.18 N/m

Since, tire and spring can be considered in a series connection (for vertical stiffness),

$$Kwf = \frac{Krf*Kt}{Kt - Krf}$$

= 16705.181 N/m

$$Kwr = \frac{Krr*Kt}{Kt - Krr}$$

= 16760.943 N/m

Now calculating spring rate,

$$Ksf = \frac{Kwf}{MR^2} \text{ and } Ksr = \frac{Kwr}{MR^2}$$

Ksf = 26101.845 N/m and Ksr = 26188.973 N/m

Wheel Load transfer due to lateral acceleration:

$$\delta Wf = Ay * \frac{W}{Tf}\left(\frac{Hrm * Køf}{Kø} + \frac{b}{l}Zrf\right)$$

δWf = 46.9 Kg

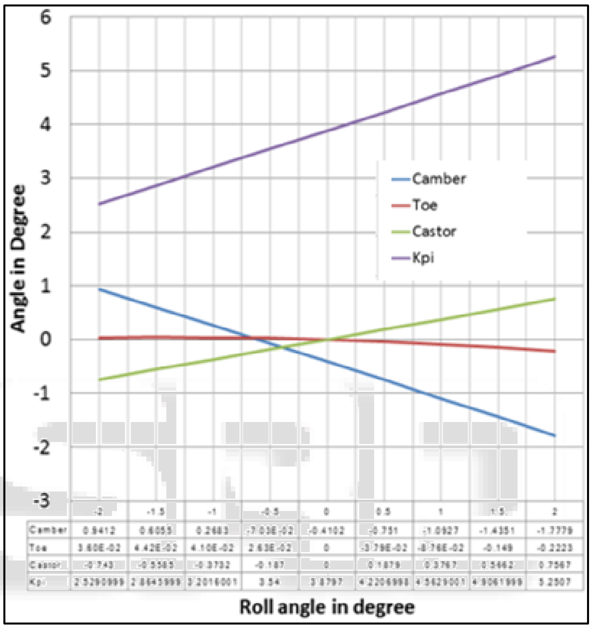
$$\delta Wr = Ay * \frac{W}{Tr}\left(\frac{Hrm * Kør}{Kø} + \frac{a}{l}Zrr\right)$$

δWr = 51.28 Kg

2) Wheel Loads

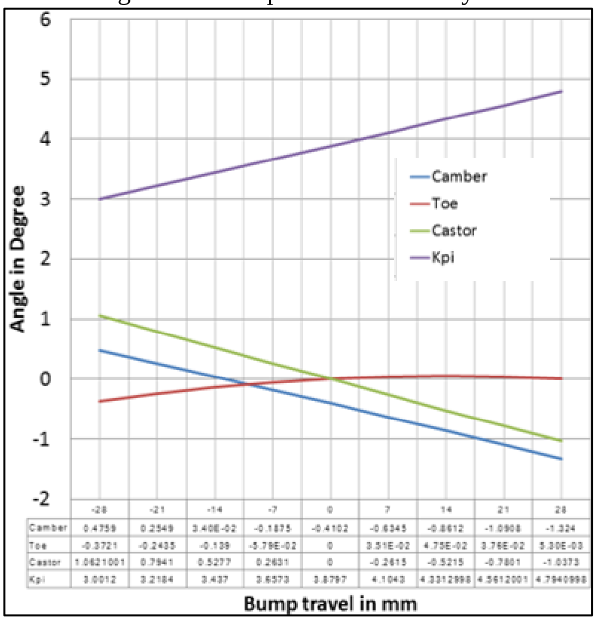
- Wfo = Wf + δWf = 114.4 Kg
- Wfi = Wf - δWf = 20.6 Kg
- Wro = Wr + δWr = 133.78 Kg
- Wri = Wr - δWr = 31.22 Kg

No anti-roll bars were included in this design to avoid extra layer of complexity this would add to the design. It was deemed more effective to spend time getting the other areas of the suspension correct. It is arguable that the effects of an anti-roll bar would be limited because of a FSAE vehicle's very low Centre of gravity and correspondingly small roll angles. Secondly anti-roll bars do not reduce the amount of lateral weight transfer; they are merely a tuning tool used to adjust handling by varying the portion of lateral weight transfer on the front relative to the rear.



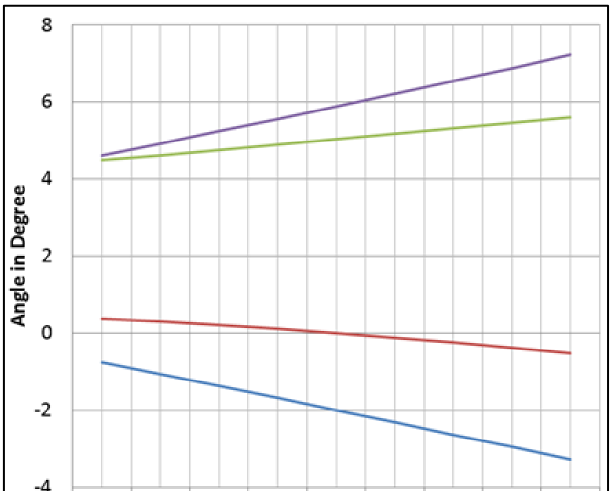
	-2	-1.5	-1	-0.5	0	0.5	1	1.5	2
Camber	0.9412	0.6053	0.2683	-7.03E-02	-0.4102	-0.751	-1.0927	-1.4351	-1.7779
Toe	3.60E-02	4.42E-02	4.10E-02	2.63E-02	0	-3.79E-02	-8.76E-02	-0.149	-0.2223
Castor	-0.743	-0.5583	-0.3732	-0.187	0	0.1879	0.3767	0.5662	0.7567
Kpi	2.5290999	2.8643999	3.2016001	3.54	3.8797	4.2206998	4.5629001	4.9061999	5.2507

Fig. 1: Rear Suspension Roll Analysis



	-28	-21	-14	-7	0	7	14	21	28
Camber	0.4759	0.2549	3.40E-02	-0.1875	-0.4102	-0.6345	-0.8612	-1.0908	-1.324
Toe	-0.3721	-0.2435	-0.139	-5.79E-02	0	3.51E-02	4.75E-02	3.76E-02	5.30E-03
Castor	1.0621001	0.7941	0.5277	0.2631	0	-0.2615	-0.5215	-0.7801	-1.0373
Kpi	3.0012	3.2184	3.437	3.6573	3.8797	4.1043	4.3312998	4.5612001	4.7940998

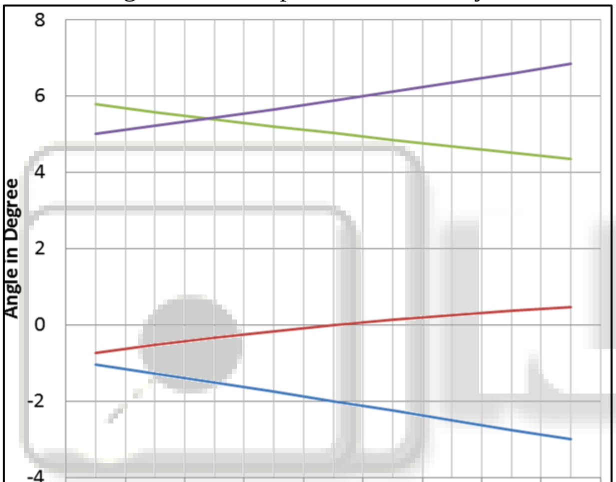
Fig. 2: Rear Suspension Bump Analysis



	-2	-1.5	-1	-0.5	0	0.5	1	1.5	2
Camber	-0.7517	-1.0608	-1.3719	-1.685	-2	-2.3167	-2.635	-2.9548	-3.276
Toe	0.3721	0.2928	0.2044	0.1069	0	-0.1164	-0.2424	-0.3782	-0.524
Castor	4.4864998	4.6170001	4.7502999	4.8862	5.0249	5.1662998	5.3102999	5.4569001	5.6061001
Kpi	4.6015	4.9166999	5.2351999	5.5569	5.8818002	6.2097998	6.5409002	6.8751998	7.2125001

Roll angle in degree

Fig. 3: Front Suspension Roll Analysis



	-28	-21	-14	-7	0	7	14	21	28
Camber	-1.0415	-1.2783	-1.5167	-1.7572	-2	-2.2456	-2.4945	-2.7469	-3.0034
Toe	-0.7365	-0.5256	-0.3329	-0.1579	0	0.1411	0.266	0.3749	0.4681
Castor	5.7768002	5.5826001	5.3924999	5.2066002	5.0249	4.8474999	4.6743002	4.5054002	4.3408999
Kpi	5.0028	5.2150002	5.4322	5.6543999	5.8818002	6.1143999	6.3526001	6.5963998	6.8460999

Bump travel in mm

Fig. 4:

E. Design of Attachment Parts for A-Arm

An attachment for A-Arm facilitates the function of assembling a-arm tubes with the upright and damper. It is to be kept in mind that the design should be in such a way that minimum stresses should be developed during dynamic conditions as the load of entire sprung mass is passed through it.

1) Version 01

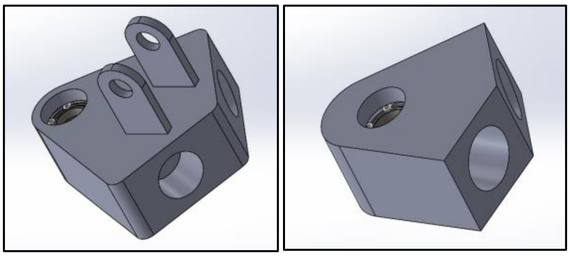


Fig. 5: Lower and Upper attachment parts respectively of version 01

This was the first design that came to our mind. The circular slots at front faces were provided to insert a-arm tubes which maintain the predefined angle between a-arm tubes during welding thus, eliminating the need of fixture, but after we realized that to accommodate this slot we had to increase the thickness of part more than necessary because whole tube would go inside the part which increased the thickness. The vertical mounting plates were welded on the attachment parts which were going to be used for lower a-arms to accommodate the push bar. In case of lower part circular slots at front would use the space below mounting plates of push bar means that we wouldn't have to increase the length to provide mounting plates meaning that, length of part can be decreased without worrying about space for mounting plates. A through hole of 16mm diameter was provided for the fitting of spherical bearing and two grooves at both side of bearing were provided for the circlips. The design gave good strength in fact it gave more strength than needed and it was very bulky and heavy design and material and machining cost were high too. In short the length of part could be kept small but the thickness was too much so, we decided to make changes in the design and optimize it.

2) Version 02

After making some changes in the previous design we came up with this design. Instead of circular slots we provided extruded cylindrical parts to maintain the predefined angles between the a-arms. This helped us to decrease the thickness as the tube would go upon the extruded cylinder which didn't affect the thickness of part rather than going inside the part increasing the thickness of part, keeping the thickness minimum such as circlip at both side of bearing could be provided. But by doing so the total length of part increased cause, in previous version the tubes would go below the mounting plates but that's not the case here. We had to provide extra length for extruded part and face supporting that extruded part. So this design was still heavy and raw material cost and machining cost were high so we decided to think about these problems and optimize it further.

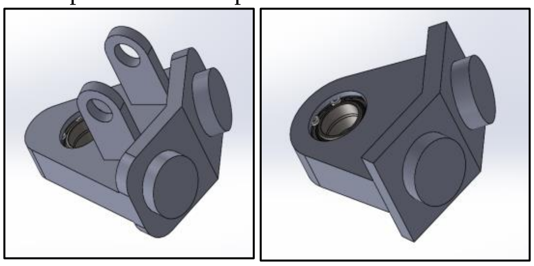


Fig. 6: Lower and Upper attachment parts respectively of version 02

3) Version 03

After analyzing previous designs we made some changes and came up with this design. Instead of having two front faces perpendicular to the axis of a-arms we gave only one face perpendicular to the center axis of part and we also cut down the extruded cylinder provided to maintain the angles. This rose the need of fixtures to maintain the angles between the tubes of a-arm and also the surface contact between the tube and face of part was very less and there were no other support for tubes which made us think that it may be risky considering the load which it may have to go.

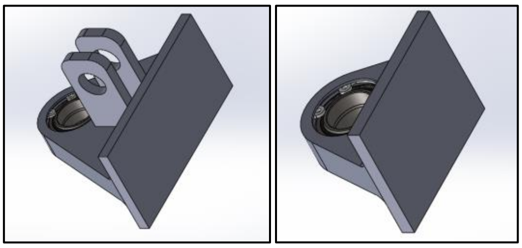


Fig. 7: Lower and Upper attachment parts respectively of version 03

There were no significant progress regarding weight reduction in part as well as material and machining cost of part as the thickness of part was still same as the previous design. So we decided to do more changes to the design.

4) Version 04

After going through above complicated designs we came up with this simple design which met our basic requirements. The front face provided perpendicular to the center axis of part was removed. Thickness at the front end where previously the face was provided was kept 5 mm and at rear end where ball bearing would be provided, the thickness was kept 10mm to accommodate space for the groves needed for circlip at both side of bearing. In this design the thickness of part where the tubes would be welded was just 5mm so came up with an idea to provide 5mm thick slot having 20mm depth at the end of the tube so that tube would fit on the part exactly and the surface which could be welded was greater than previous designs so we had no worry about the strength of welded joint of part and tube. Well, to maintain the predefined angles between a-arm tubes the necessity of fixture rose up again, but at that point we got it that fixtures were unavoidable if we wanted to optimize the previous design any further and to bring significant changes in existed design, so we accepted the use of fixtures without any more doubts. The design was almost optimized, the front face being removed lessened the required raw material and machining needed on raw material to get the final part was reduced too, thus reducing overall cost. But still there was still one thing that we weren't satisfied with: thickness at rear end was 10mm for that we had to get the block of 10mm width as raw material and after that machining phase we would have to do extra machining on whole part except rear end, so the raw material and machining cost would increase unnecessarily. In the end we thought that we had to do something about this also.

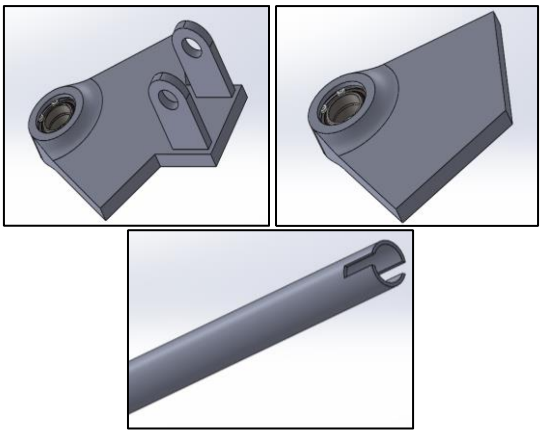


Fig. 8: Lower and Upper attachment parts and slotted tube respectively of version 03

5) Version 04

The long process of thinking finally led us to this design. This was the simplest design of all previous designs yet it fulfilled all of our basic requirements.

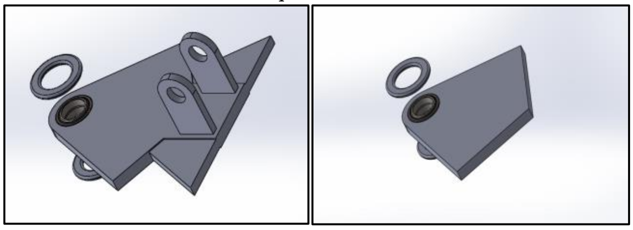


Fig. 9: Lower and Upper attachment parts respectively of version 04

In previous design we had provided 10mm thickness at rear end for the grooves but in design we removed this and gave uniform thickness of 5mm. The thickness of bearing was 5mm and outer diameter was 16mm same as the through hole given at the rear end, thus bearing was press fitted in hole and to secure the bearing further we brazed two washers with part at both side of bearing. The washers were of such size that it would only be in contact with collar allowing the ball of bearing to rotate without any obstruction. This setup of brazed washer at both side of bearing secured the bearing in hole of part that it would not come out of part if press fit were to fail. In lower parts, rather than increasing the whole length of part to avoid the collision of mounting plates of push bar and knuckle we extended a rectangular section upon which push bar mountings would be welded. This extended section was only provided at front lower parts because, push bar mounting plates and knuckle may collide only during cornering after certain angle but in rear the tire will not change its toe angle (our car is front steered) so there was no need to increase the weight of rear lower part by providing extended rectangular section. The part was cut from 5mm metal sheet by laser cutting and the hole for bearing was provided by same method. So raw material cost and machining cost was very less compared to all previous design and was most light weighted also.

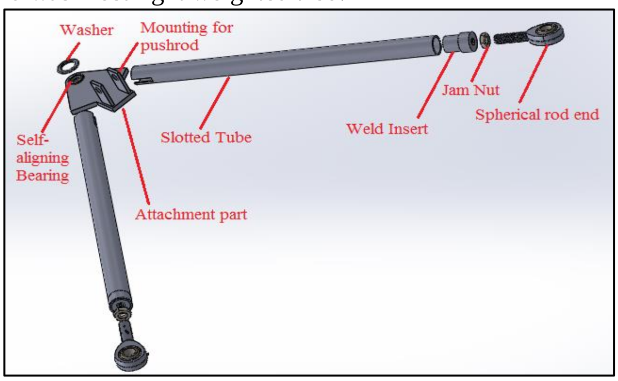


Fig. 10 A-Arm assembly

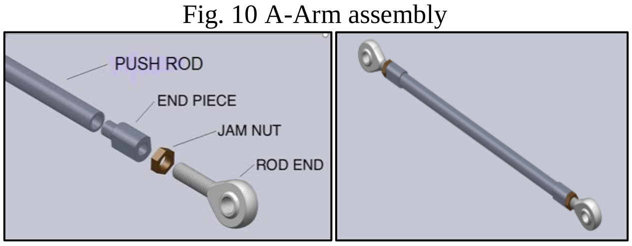


Fig. 11: Pushrod assembly

III. MATERIAL SELECTION & PROCUREMENT

A. Material Selection

1) A-Arms

Carbon fiber was our first priority for the wishbones as it is commonly used by the more experienced teams as it offers a great improvement in strength to weight ratio and stiffness. Carbon fiber exhibits yield strength approximately that of steel (dependent on the structure of the fiber) and density of approximately plastic (Dragon Plate 2011). However, the main disadvantage and the reason it was excluded for our vehicle was its high cost, with a 22mmx3mm (O.D. x thickness) tube costing was €76.13 per meter.

We were also thinking about Aluminum due to its extensive use in lightweight, high-strength structures like airplanes. Even though the yield strength to weight ratio of some high strength steels approaches that of aluminum, its much lower specific gravity of 2.7 g/cm^3 compared to steel's 7.85 g/cm^{3,} allows for stronger structures to be built, especially in bending as the material will be further away from the neutral axis (higher second moment of inertia)[4]. However, aluminum's disadvantages include a low modulus of elasticity (under identical loading an aluminum component will deflect three times as much), poor creep wear and poor resistance to fatigue. Aluminum however was rejected for the two main reasons of difficulty in welding (especially for the inexperienced welder as there is a fine line between no penetration and blowing holes) and because it would be 4-5 times expensive as steel for only a small weight.

The next option was Chrome Molybdenum (AISI 4130) high-strength steel. The content composition is in the range of 0.3% Carbon, 0.5% Mn, 0.3% Si, 1% Cr and 0.2% Mo. AISI 4130 alloy steel contains chromium and molybdenum as strengthening agents. It has low carbon content, and hence it can be welded easily. Cost of appropriate size AISI 4130 tubing ranges from €12.55 to €25.10 per meter (Go Gear 2010). But after doing simulation we got higher factor of safety which was not required and considering the cost AISI 4130 has high cost, costs for a-arm tubing is estimated at €34.62. Chrome Molybdenum was not used because of the reason: it was costly and had more strength than required so instead of using this we thought of using more economic material having adequate strength as per the requirement.

So, we decided to use IS 2068 which is a grade of mild steel. It can be welded easily and have adequate strength. It has mass density of 7.850g/cm^3 and yield strength of 240N/mm^2 and has cost of 1.27 € per meter. Material has enough strength needed and is comparatively economic than previous options. The tube used has 14mm outer diameter with 1.75mm wall thickness.

2) Push Rod

AISI 4130 was selected for push rod material. Dimension elected as 0.75" diameter and 2 mm thickness. Length of push rod for front suspension is 49.5cm and for rear suspension is 13cm.

IV. STRUCTURAL STATIC ANALYSIS

Part : Rocker Arm (Bell Crank)	
Load Applied	1400 N (pushrod) 2200 N (spring reaction)
Fixed Points	Fulcrum
Maximum Deformation	0.091 mm
Maximum Stress	166.1 MPa
Minimum FOS	2.3

Table 2: Boundary Conditions for Analysis

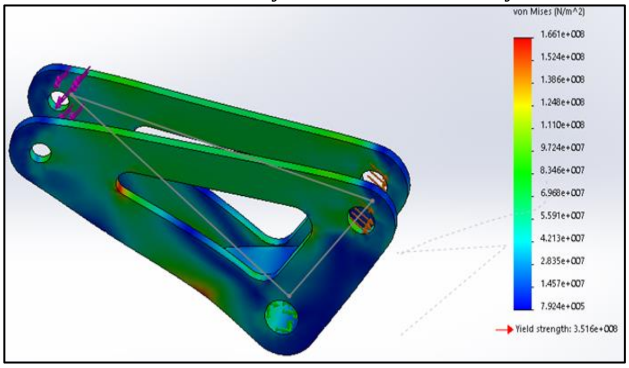


Fig. 12: Static FEA Analysis of Rocker Arm

REFERENCES

[1] www.formulastudent.de, "Formula Student Rules 2017".
[2] William F. Milliken, Douglas L. Milliken, "Race Car Vehicle Dynamics", SAE International, December 1994.
[3] Staniforth, "Competition Car Suspension: Design, Construction, Tuning", 3rd ed., Haynes Publishing, November 1999.
[4] https://www.engineersedge.com/

